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About Authors

● **Ernest G. Boden** *(M '25) received his technical education at the Manchester School of Technology, England. After early training in the engineering department and factory of Crossley Motors, Ltd., he served as assistant chief jig and tool designer of the National Aircraft Factory, Stockport, and assistant chief of engine design, Sizaire Berwick, London, before coming to the United States. In this country he has been in the inspection department of Rolls-Royce, chassis and quality engineer of Chandler and, for the past eleven years, with Timken Roller Bearing Co., where he is now experimental engineer. Mr. Boden developed the original laboratory tests on extreme-pressure lubricants which resulted in the Timken Lubricant Tester.*

● **Kenneth Campbell** *(M '34) has had a diversified interesting career. He was born in 1899 at the Lick Observatory, Calif., of which his father, the late Dr. W. W. Campbell was director. In 1914 as a helper on the Crocker Solar Eclipse Expedition he went to Kiev, Russia. During the World War he was an ambulance driver on the Piave River, Italy, and served as a cadet and 2nd lieutenant in the Royal Air Force, England. After receipt of his A.B. and S.B. degrees from Harvard College ('21) and Harvard Engineering School ('23) he worked in the Bethlehem, Pa., plant of the Bethlehem Steel Co. in a variety of operating and maintenance assignments, including blast-furnace foreman, until 1928. He then served on the staff of Sanderson & Porter, industrial and powerplant engineers until 1933. For the past five years he has conducted applied research on aircraft-engine development problems for the Wright Aeronautical Corp., where he is project engineer in charge of design in special fields including superchargers, induction systems, valve gears, instrumentation, cylinder cooling and cowling.*

● **H. K. Cummings** *(M '27), following his graduation from Worcester Tech in 1910, taught physics for five years and did graduate work in physics and mathematics at Clark University and the University of Chicago. He was lieutenant in the Ord-*

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Prentis at S·A·E Annual Dinner

Says Reform Must Spring from Hearts of Men

"CUSTOMS that were common in business a generation ago, are today taboo; any reform in personal or economic morals that is to be sound and permanent, must spring primarily from the hearts and minds of men. Restrictive and regulatory laws can only make permanent the advances that the collective moral sense of the community develops and approves."

In these words H. W. Prentis, Jr., president, Armstrong Cork Co., emphasized a main thesis in his talk on "Patriotism and Sound Industrial Practices" at the SAE Annual Dinner on Nov. 14 at the Hotel Commodore, New York.

Greatest Annual Dinner

Nearly 1200 members and guests filled every available seat at this most dynamic gathering of its kind in many years. B. C. Forbes, leading business commentator, was toastmaster, and Grover Whalen, president, New York World's Fair, spoke briefly in welcoming the SAE World International Automotive Engineering Congress to the Fair at which May 24 has been set aside as SAE Day. President C. W. Spicer introduced 1939 nominee-for-president W. J. Davidson, who expressed sincere appreciation of the honor given him and paid tribute to the high standards of leadership set up by SAE presidents of the past. Paul Whiteman and his orchestra furnished the entertainment.

Calling vigorously upon fellow-business men to set their own house in order, Mr. Prentis said that "business must come into the court of public opinion with clean hands if it is to render its full patriotic duty to the nation in this crucial period." He spoke in some detail of the Platform for American Industry, formulated by the Committee on Industrial Practices of the National Association of Manufacturers of which he is chairman – a code of ethics for business executives which, he said, has been endorsed by the Congress of American Business, has been acclaimed by the press, has been welcomed in right-wing government circles and which "indicates a growing sense of social stewardship in industry."

"Human freedom," Mr. Prentis urged, "rests upon the tripod of free enterprise, representative democracy and religious freedom. Destroy any one of the three legs of this tripod," he warned, "and the whole structure of freedom will crash. The history of past ages, the events of the last two decades and recent happenings in Europe, all bear incontrovertible testimony to the truth of this fact," he said.

"Whether sincere or insincere, logical or foolish, the traducers of American industry must not be permitted to destroy the faith of the American people in our system of private, free enterprise," he declared.

For too long, he pointed out, the American people, its business and professional men, its intelligent farmers and wage earners have taken for granted the "ineffable blessings of our political, economic, and religious liberty." For a hundred years or more, we have been so busy exploiting our natural resources, inventing new machines and tools, contriving new organization procedures, piling up physical conveniences, comforts and luxuries that only the acquisition of wealth could provide, "that we have failed lamentably to pay much attention to the maintenance of those liberties that we have smugly assumed were ours by inalienable rights – and would therefore automatically continue indefinitely."

"But, in the meantime, a tempest of propaganda has piled the flood waters of ignorance and class hatred high against the dikes of freedom," he said.

Whether the business man realizes it or not, Mr. Prentis said, every executive will actually be doing his part to perpetuate our system of representative democracy and religious liberty by bringing his individual practice to an unassailable ethical plane. Neither democracy nor religious liberty can long endure in the absence of the other, he said.

Business Rises to Opportunity

Citing the catch phrases that have played an immense part in forming public opinion, Mr. Prentis said that the government's census shows that most minors are working on farms, in small family enterprises, in retail distribution, domestic service, and on newspaper routes. Yet manufacturers are generally accused of employing the bulk of child labor in this country. He also showed that the nation's economic power is not highly concentrated, but that 81% of the country's entire economic activity in 1933 was carried on by partnerships, individuals, and by small and medium-sized companies.

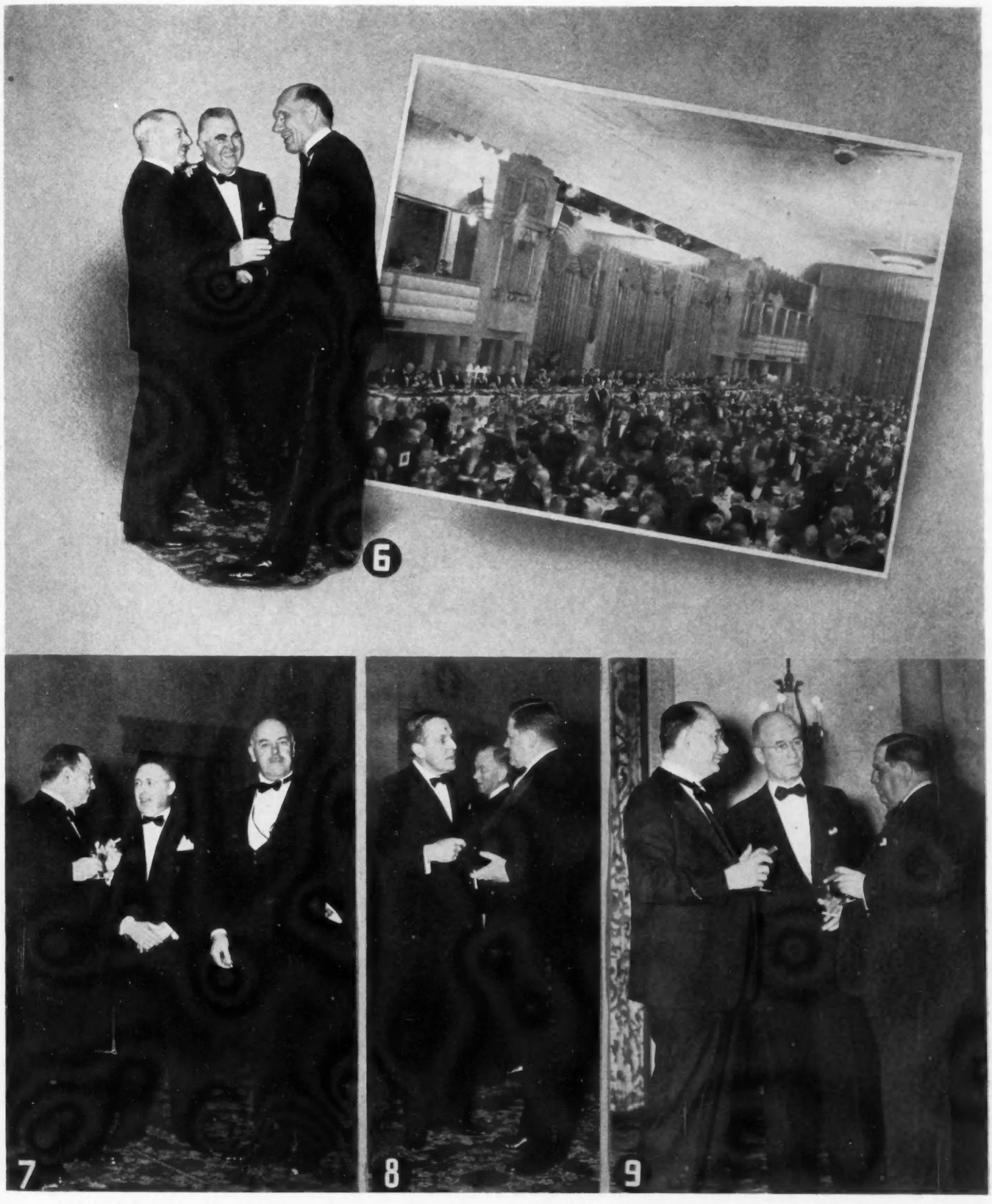
"I am confident that American business men will rise to their opportunity today," he concluded.

Notables Were Numerous at the Reception for



1. W. S. Knudsen, president, General Motors Corp., and Pyke Johnson, vice-president, Automobile Manufacturers Association.
2. H. W. Prentis, Jr., speaker of the evening, and Toastmaster B. C. Forbes.
3. Ralph R. Teetor, past-president, SAE, and Dr. George W. Lewis, National Advisory Committee for Aeronautics.
4. Byron C. Foy, president, De Soto, and Roland Campbell, president, Reo.
5. W. J. Davidson, General Motors Corp., nominee for 1939 SAE presidency, and John A. C. Warner, SAE secretary and general manager.

Speakers' Table Guests at SAE Annual Dinner



6. Alfred Reeves, vice-president and general manager, Automobile Manufacturers Association, Fred M. Zeder, vice-chairman of the board, Chrysler Corp., and C. F. Kettering, vice-president, General Motors Corp.

7. B. C. Forbes, toastmaster, Nicholas Dreystadt, general manager, Cadillac, and B. E. Hutchinson, vice-president and finance committee chairman, Chrysler Corp.

8. Paul G. Hoffman, president, Studebaker Corp., and K. T. Keller, president, Chrysler Corp. In background, Ward M. Canaday, board chairman, Willys-Overland.

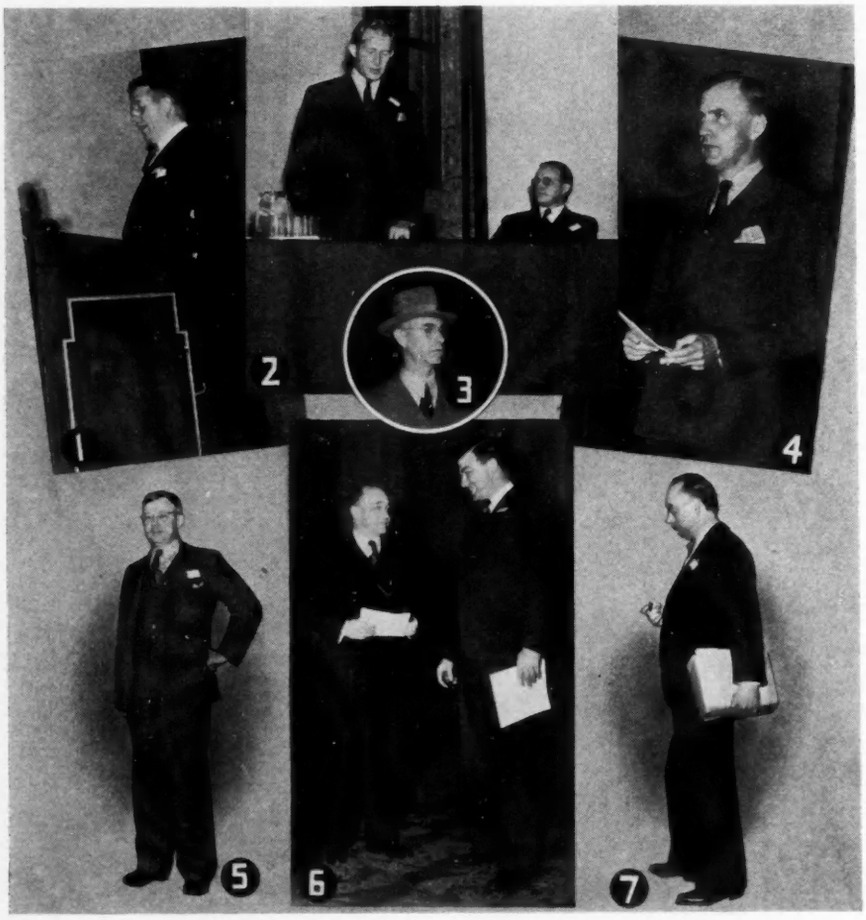
9. M. C. Horine, chairman, Metropolitan Section, SAE, Charles F. Horner, president, National Aeronautic Association, and George W. Mason, president, Nash-Kelvinator Corp.

Photographs by Nation-Wide News Service

News of the Society

Fruitful Debates Mark SAE 1938 Transportation Meeting

ACCENT was on discussion at the SAE 1938 Transportation Engineering Meeting, held at the Hotel New Yorker, New York, Nov. 14-16. Everything conspired to promote it. Six straight-from-the-shoulder papers either stimulated or provoked it; four one-paper sessions and one two-paper session allowed plenty of time for it; and frank opinions by equipment manufacturers and fleet operators who did not pull their punches brought to the discussion a new value and interest. The meeting was sponsored by the Transportation & Maintenance and the Truck, Bus & Railcar Activities of the Society, with the cooperation of the Metropolitan Section.



1. F. S. Baster spoke on the opening day
2. TB&R Vice-President H. E. Simi and T&M Vice-President F. L. Faulkner opened the meeting with greetings to members and guests
3. Carl J. Bock, nominee for 1939 TB&R vice-president
4. W. K. Creson, chairman of the TB&R meetings committee
5. O. A. Axelson, chairman of the T&M meetings committee
6. F. K. Glynn (*left*) presided at the opening session. H. O. Mathews (*right*) nominee for 1939 T&M vice-president
7. M. C. Horine read a paper at the opening session

Photographs by Nation-Wide News Service

Debate on the first day centered around the desirability of establishing a new basis for tire sizes, and on ways to get higher output and better economy from truck engines. Advantages and disadvantages of diesel-electric, hydraulic, and gear-box drives for buses were aligned in the next session. How far it is desirable to go in the use of instruments for maintenance work was the moot question in the third session. On the last day the "rough-and-tumble" engineering efforts of a number of fleet operators to solve special problems were reported, and three ingenious developments in truck springs were compared critically.

Two Vice-Presidents Open Meeting

An attendance of more than 200 greeted F. L. Faulkner, vice-president of the T&M Activity, and H. E. Simi, vice-president of the TB&R Activity, as they officially opened the meeting on Monday afternoon with brief talks. Session Chairman F. K. Glynn directed the wealth of discussion generated by the two papers read.

A bmep of 125 lb per sq in. is within the "realm of practical possibility" with 70-octane fuel, believes F. S. Baster of The White Motor Co., if the suggestions advanced in his paper, "Why Not 125 BMEP in an Ell-Head Truck?" are followed.

The goal of exceedingly high output is not selected solely for the purpose of being able to propel a larger payload faster, but more particularly for the attendant reduction in specific fuel consumption, especially at road-load power, Mr. Baster explained early in his paper.

Distribution, combustion, and cooling are still the fundamentals in engine design, he pointed out. To improve distribution he suggested a manifold with a single division zone located close to the carburetor. This arrangement, he told his audience, will serve the dual purpose of having a minimum of condensate to distribute, and any of which is formed subsequently must go into the cylinder for which it was intended rather than permitting a choice of cylinders by the attending velocity changes.

It is doubtful if the inventor of the hydraulic lifter himself originally could have conceived all of the advantages of this device, Mr. Baster contended.

An unusual feature of Mr. Baster's presentation was his use of large clean-cut charts to illustrate his talk, instead of the conventional lantern slides. Every detail of these charts was clearly discernible to those in the back rows.

Additional advantages of hydraulic lifters, to supplement those given in Mr. Baster's paper, were enumerated by Carl Voorhies of Wilcox-Rich, in written discussion. That the ramp required for adjustable lifters is greater than that for hydraulic lifters, and that hydraulic lifters permit the use of any valve material without consideration to coefficient of expansion, are two of the advantages mentioned.

The work reported in Mr. Baster's paper parallels our experience on distortions, recalled Lewis P. Kalb of Continental Motors Corp. To overcome mechanical distortion, he explained, we employed the judicious use of cast iron; then temperature distortion caught up with us. Mr. Baster did not mention one important method of avoiding mechanical distortion, that is, use a few more cylinder-head bolts, he pointed out. "By saving on bolts and washers, you may be storing up trouble for yourself, especially when bmep's get up to where Mr. Baster says they will go," he concluded.

Tire Sizes Attacked and Defended

Intervals between tire sizes for six-tire trucks vary from 1050 to 6600 lb in capacity, Merrill C. Horine, Mack Mfg. Corp., demonstrated in his paper, "Tire Sizes – Not More but Better,"

(*Continued on page 23*)



Conference Before West Coast Meeting

General Meeting Chairman Ellis W. Templin, *left*, discusses the meeting with W. E. Powelson, chairman of the Southern California Section; Charles F. Becker, representing Northern California Section Chairman G. L. Neely; J. Verne Savage, chairman of the Oregon Section; and Harley W. Drake, chairman of the Northwest Section.

Western T&M Men Concentrate On Slashing Operating Costs

IF western fleet owners suddenly find that their expenses are taking a nose dive, they may well be able to trace the economies to knowledge gained at the SAE West Coast Transportation & Maintenance Meeting held in Los Angeles, Nov. 3-5, to the theme "Lower Overall Costs for Motor-Vehicle Operations."

Cost control, overhead, depreciation – accounting subjects of immense interest to alert fleet operators, shared the program with technical papers on methods of reducing expenses in shop operations and through preventive maintenance. Army maintenance practices were clearly explained and representatives of two Pacific Coast Universities told what these schools are doing to advance automotive transportation.

At the Banquet Col. R. T. Eddy, regional director, Interstate Commerce Commission, gave a brief talk on "Regulation of Highway Carriers."

From the opening session on Thursday morning until the completion of the inspection trips on Saturday, the meeting was a credit to its sponsors – the four SAE Pacific Coast Sections; and to Ellis W. Templin, its general chairman, and his hard working committees.

Total attendance at all sessions neared 1200 and included representatives of 50 leading truck and bus fleets that operate more than 2000 vehicles.

Introduced by Session Chairman Robert N. Reinhard, Golden State Dairies, Watt L. Moreland, president and general manager of the Moreland Motor Truck Co., presented the first paper of the meeting, "The West's Contribution to Low-Cost Motor-Vehicle Operation."

The West and the motor vehicle have grown up together, he declared. With less railroad mileage per thousand cultivated acres than in the East, this section has utilized motor trucks and buses to link the more distant centers of population and the thousands of communities which have no other means of transportation – many of which owe their existence to the automotive industry.

The need for high standards of performance and the difference of operating conditions between the East and the West, he said, stimulated the ingenuity of western engineers and at one time it was stated that 25 per cent of all the nation's patent applications for motor-vehicle improvements were applied for by citizens of one of the western states, California.

Among the West's contributions to reliable and economical transportation mentioned by Mr. Moreland, were: the first copper-lead bearings; the first successful engine to use distillate for fuel instead of gasoline; the double counter-shaft multiple-speed transmission for use in heavy motor trucks; auxiliary transmissions; air brakes; hydraulic brakes; multiple axles; and the modern drop-frame, low-center-of-gravity, safety passenger bus. Many of these developments were western inventions, others were results of western operators' demands upon eastern manufacturers, he added.

Following the presentation of the paper Stanley Whitworth, vice-president in charge of production, Studebaker Pacific Corp., presented Mr. Moreland with a plaque commemorating his 30 years of SAE membership.

An unannounced feature of the morning session was a series of slides on micromatic honing and super finish presented and explained by C. H. Eden of the Foster Machine Co., Elkhart, Ind. Mr. Eden's comments were based upon a paper by D. A. Wallace, president, Chrysler Sales Corp.

J. Verne Savage, chairman of the Oregon Section, presided at the luncheon session. He introduced Harold Ager, chairman of the SAE Student Branch at Oregon State College, who told "What Oregon State College is Doing That is Important to Pacific Coast Motor-Vehicle Operators." He touched upon the general and specific engineering courses that are preparing young men for work in the transportation industry; the University's laboratory facilities; and its Engineering Experiment Station. He told of SAE Student Branch activities, mentioning specifically a safety campaign which it organized and the students' participation in a technical paper contest sponsored by the Oregon Section. In concluding he listed a number of Oregon State graduates affiliated with the transportation industry.

Chairman Savage, Harley W. Drake, Farley E. Beckett and Charles F. Lienesch joined in discussion, championing further development of student activities by the Society.

"We must give as much attention to 'humanics' as we do to mechanics, if we are to reduce our costs of operation," Harley W. Drake, chairman of the Northwest Section declared as he opened the afternoon session at which he presided. He defined "humanics" as the understanding of humanity, and stressed the necessity of having foremen and supervisors as well as executives trained in "humanics."

The United States has a transportation system valued at $60,000,000,000 of which one-half is represented by highways and trucks, C. G. Anthony, speaker of the session, told his hearers when introduced by Mr. Drake.

Mr. Anthony is general manager of the Pacific Freight Lines. He spoke extemporaneously on the problem of reducing maintenance costs, stressing the importance of the subject by noting that in many businesses delivery expenses total from 20 to 40% of the gross income.

He reported that his company operates 600 pieces of equipment hauling general merchandise on regular schedules over regular routes between fixed terminals. Maintenance costs, he said, constitute about 11% of the total operating costs, and those of a subsidiary engaged exclusively in the transportation of liquid petroleum products in large 6-wheel, 10-ton trucks and trailers approximate 9% of total operating costs. These, he believes, are among the lowest maintenance costs in the country.

He emphasized the importance of having expert drivers. In his company's fleet, he said, the operator reports each night on the condition of the truck, and if repair work is required it is done that night. Mr. Anthony believes in a preventive maintenance program, but warns that it is subject to abuse and may increase expenses, not reduce them.

If maintenance is properly done and inspection intelligently performed, he stated, a general overhaul should not appear very frequently in a well managed shop. Such a job receives careful consideration by an executive in his company, and in many cases it is found to be more profitable to do away with the old equipment and replace it. Sometimes, he added, just the most necessary repairs are made and the equipment placed in less rigorous service.

Measuring Stick Needed

It is Mr. Anthony's belief that expenses can be kept to a minimum only if there is some dependable measuring stick used for measuring the efficiency of an operation, and that this can best be set up through accounting.

Pacific Freight Lines, he said, keeps records for each piece of equipment on the cost of maintenance parts, maintenance labor, outside repairs, repairs caused by accidents, tire repairs, body-repair labor and parts, battery maintenance and garage overhead per mile operated. The same expense items are also shown in percentage of gross revenue for each month and the total for the year to date. These operating ratios, he contends, constitute the

yardstick to be applied in measuring the efficiency of the maintenance department.

Answering a question on comparative maintenance costs of diesel and gasoline motor equipment, put by Joseph Gutman, the speaker stated that while diesel maintenance costs were considerably higher than those of gasoline motors they have been reduced so that they now run 4% less than for gasoline engines. This reduction, he said, has been due mainly to developments which have been made in diesels. Parts have been redesigned, he explained, and certain metallurgical changes have been made; all resulting in profound reduction in maintenance costs. He noted that his company has abandoned gasoline motor equipment.

At the close of his paper, Mr. Anthony was presented with an Indian headdress and a plaque dubbing him "Big Chief Dieselizer" for his work in pioneering the use of diesel engines for highway transportation.

Following the afternoon session there was a demonstration of the United States Army mobile shops and mobile mounted anti-aircraft gun units brought from Fort McArthur.

The Army Dinner followed the demonstration. Robert E. Rowley, engineer, general plant division, Department of Water & Power, Los Angeles, presided, introducing the speaker, Lt. Col. E. C. McGuire, assistant chief of staff for supply, Ninth Corps Area.

Colonel McGuire explained that, in general, motor maintenance in the army is based upon the unit replacement system which is divided into five echelons of repair. The first echelon covers the vehicle operator's maintenance, such as cleaning, lubricating, servicing, and the tightening of bolts and screws.

The second constitutes the field operator's maintenance, which includes periodical and systematical inspection by an officer or his assistant

(*Continued on page 22*)

Colwell Reports Strides In Valve Engineering

● No. California

"Material, design, and expense are the three factors which determine valve efficiency," so stated A. T. Colwell, vice-president in charge of engineering, Thompson Products, Inc., in his talk at the Nov. 1 meeting of the Northern California Section. His topic was "Modern Valve Engineering for Long Performance – Aircraft and Automotive."

Mr. Colwell stressed the importance of material, over design and even cost. Assisted with slides, he gave a background of valve development from the first iron valves, which would not operate 50 hr in aircraft service, up to the present time when sodium-cooled, special steel valves operate from 5000 to 8000 hr. He recalled fond memories to the old-time fliers, when a side slip with an OX-5 motor meant warped valves and perhaps a forced landing.

The valve itself is not the complete answer to the problem since forced feed lubrication of the entire mechanism, as used by the Wright and Pratt & Whitney corporations, has been the greatest single aid to satisfactory valve operation, the speaker declared. A creeping valve, one that rotates slightly, was pointed out as being highly desirable as it insures even heat distribution and gives the valve a chance to lap its own seat. A washer or ball bearing insert between the spring and its seat on the block has been found to be effective in imparting the creep.

In summing up the developments of the valve field to date, Mr. Colwell mentioned: stellite seats, which are pressed into the block and reduce considerably the requirement for grinding; good cooling, as accomplished by sodium-cooled valves; better design of the gas flow through the ports due to the head shape of the valves; since the adoption of the Bradley type modified tulip valve design there has not been a single report of valve breakage.

The speaker then showed the application of the airplane design to the automotive industry and stated that with the higher output being taken from engines in this service, their requirements are rapidly approaching that of the aircraft engine. There are several fleet operators at this time on the Pacific Coast using sodium-cooled valves and special valve inserts as used in aircraft engines. Their experience has been that the increased cost of this equipment is justified, Mr. Colwell stated.

Interest in new developments was aroused when Mr. Colwell predicted that within a few months there would be several new improvements in engine design that have been given little publicity heretofore, such as: dry-type cylinder liners; piston rings of entirely new design; chemical precoating of cylinders, rings and pistons to insure rapid and safe break-in; a constantly variable transmission; superchargers for trucks and buses that can be connected and disconnected according to the engine load; and a turbo-supercharger of a practical design for aircraft.

Mr. Colwell stated, however, that this is not the end of the new developments and challenged those present by stating that anyone who could solve one of the following problems could write his own meal ticket:

1. A substitute for stellite inserts.
2. A better valve material than silcrome steel that will sell for a lower price.
3. A harder guide material than now used which could be easily machined.
4. A more efficient coolant than sodium. Sodium potassium compounds have been found to be more efficient but more difficult to seal in the valve stem.
5. Elimination of sticking valves by the development of an oil which will form no lacquer.
6. An automatic valve lifter which is mechanically rather than hydraulically operated.

The technical session was followed by a sound moving picture shown by W. C. Jordan, northern California distributor for Thompson Products, Inc., which showed the manufacture of engine parts in the Thompson factory as well as the road testing of these developments on the race tracks throughout the past two decades.

As a comedy feature, Chairman G. L. Neely had arranged for the construction of an immense valve which was presented to Mr. Colwell by J. F. Putnam as a typical example of the valve complaints received in California, the land of large engines and stories. The mystery of the valve trouble was finally solved when Mr. Putnam, with the adeptness of a professional magician, removed from the hollow stem several articles such as a bag of salt, a rubber rat, miniature garden tools and a can opener. The valve was accepted by Mr. Colwell as a remembrance of his enjoyable trip to the Coast and he promised to have it on display at his office in Cleveland.

Chrysler Superfinish Described by Roensch

● Indiana

Some 215 Indiana Section members and guests heard Max M. Roensch, powerplant development engineer, Chrysler Corp., describe the Chrysler "Superfinish" at the Nov. 10 meeting. He defined superfinish as "an extremely fine surface finish produced upon flat, internal, external, round, concave, convex and other types of surfaces by a combination of short motions, light abrasive pressures, slow abrasive cutting speeds, hard abrasive stones and a lubricant of proper viscosity that eliminate the scratches and surface defects created by previous mechanical operations, without creating scratches and surface defects in the superfinish surface that is produced on these surfaces."

Mr. Roensch spoke also of the profilometer developed by Dr. E. J. Abbott, president of Physicists Research Co., with which a superfinished surface may be measured in microinches. This instrument, he said, is portable and can be handled by almost anyone. One of these instruments was later demonstrated by a representative of the Foster Machine Co., Elkhart, Ind., which supplies the synthetic abrasive stones used in the superfinish process.

Slides were shown along with Mr. Roensch's talk which illustrated the superfinish process and compared parts finished by this and other methods.

Section Holds Meeting On University Campus

● Milwaukee

The University of Wisconsin was the site of the Milwaukee Section's combination aircraft and student-activity meeting, Nov. 4.

After registration at the University the group inspected the United States Forest Products Laboratory which is headquarters for National Forest Service Research. There they witnessed the work of the laboratory in developing protective coverings for wood, making paper from wood pulp, making plastics from wood and resin, proper veneering of wood, and research in effort to conserve natural resources.

The group then proceeded to the plant of the Madison-Kipp Corp., where a program of die-casting demonstrations took place, including several examples of precision casting.

Dinner was served at the University, at which President Dykstra welcomed the Section members to the campus. The scheduled speaker, William Littlewood, vice-president in charge of engineering, American Airlines, could not attend because of illness. In his place Capt. M. G. Beard, chief test pilot of American Airlines, and Mr. Kirchner, Mr. Littlewood's assistant, discussed air transport engineering problems. The speakers were introduced by Section Chairman John Hilt who also presented P. C. Ritchie who spoke on the Section's Student Activity.

Uses of Synthetic Materials Depicted

● So. New England

Synthetic rubber and plastics – two comparatively new products which are finding numerous applications in the automotive industry – were the up-to-the-minute topics which attracted more than 150 members of the Southern New England Section to its Oct. 19 meeting. Two du Pont experts, C. D. Locke and Stuart P. Miller, were the speakers. Their talks were preceded by the sound film, "Wonder World of Chemistry," which graphically explained the development of such synthetic products as cellophane, Neoprene, rayon and other materials developed in du Pont laboratories.

Explaining that research for a true synthetic rubber was begun in 1925 and that a fundamental process was discovered by Dr. Newlands in his research with acetylene gas, Mr. Locke, who is a chemist in the fabrics division of the du Pont company, opened his paper, "Experience and Usefulness of a Synthetic Rubber – Known as Neoprene." He revealed that the basic constituents of Neoprene are coal, limestone, salt and water, and that the price of Neoprene has been reduced from an early cost of $1.05 per lb to the present cost of 65c per lb through research and modern manufacturing methods.

The outstanding property of Neoprene, the author stated, is that it resists mineral products, oils, gasoline and acids. It is processed similarly to rubber and its cost today is somewhat more than rubber in varying compositions, he continued. If Neoprene and rubber are submerged in oil, rubber will disintegrate and lose its tensile strength where Neoprene will regain some of its tensile strength after it is exposed to air, he stated, adding that when it is exposed to sunlight Neoprene will last six months, whereas rubber will last only one month. He also declared that Neoprene-treated fabrics will stand the leakage of gases six times better than rubber.

Although Neoprene is not as good as rubber for electrical insulation, it is often used because it will resist heat better up to 300 F and is somewhat flame resistant; burning only while in direct contact with flame, according to the author. Mr. Locke also stated that Neoprene has vibration-damping qualities which are superior to rubber and that it will take a permanent set comparable to rubber. Other advantages listed are the fact that the thermal conductivity of Neoprene is 28% better than rubber; that it is not corrosive to aluminum alloys used in the aircraft industry; that its resistance to abrasion in the dry state is similar to, but no better than rubber.

The greatest use of Neoprene is for hose liners, he said, adding that rubber is used to cover these hoses because of the lower cost. On automobiles it is used as a cover on the rubber insulation, as diaphragms for fuel pumps, and as a water seal in water pumps. The aircraft industry uses Neoprene to make pontoons watertight, as a cement for sealing gasoline tanks and under rivets and in the filling of corners in construction.

Recent Developments in Plastics

In the second paper, "Plastics and Recent Development," Mr. Miller described two types of plastics. The thermal-setting type of plastic, he said, would harden at room temperature and the thermal plastic has to be chilled in order to get a semi-hard product. He likewise explained the difference between compression molding, where the molds have to be heated first then cooled, and injection molding where the compounds are injected under high pressure and cooled and hardened almost immediately.

Mr. Miller spoke particularly of the plastic which goes under the trade name of Lucite. This material, he said, has very good reflective qualities and has a conductivity of light similar to quartz in that it will bend light around a corner. Its electrostatic qualities are such that it is sometimes a disadvantage on round pieces, whereas in square or rectangular pieces the electrostatic quality is dissipated through the four corners, he reported. It was also noted that the material is readily machined; is water white; softens at 200 F; has a peculiar quality of being self cleaning out in the weather; has an index of refraction similar to light; can be made in almost any color, and its absorption of water and discoloration by sunlight is nil. This material has a 25% shrinkage in castings, Mr. Miller stated, adding that consequently it can be made only in flat panes or round bars and afterward reprocessed into desired shapes.

He explained that Lucite can be polished but is subject to abrasion; that it is about four-tenths as heavy as glass; has a tensile strength of 200 lb per sq in.; is non-inflammable at 200 F; will burn in direct flame and after flame is removed; is a very poor conductor of heat, and will not pass the ultra-violet rays of sunlight.

It is used in the automobile industry for dash board lenses, ornaments and on such lights as tail lights and indicator lights, the speaker mentioned in closing his paper.

Following the two papers was a discussion period, after which samples of Neoprene and Lucite were examined.

Air Conditioning, Economics, and the U. S. Air Force October Meeting Topics

● **Detroit**

THE complex requirements for thorough air conditioning of motor vehicles were outlined in two papers presented to the Detroit Section in October. These technical discussions were part of the opening month of SAE activities in Detroit which also included a peek at the national economic outlook and at national air defense.

The season's opener, addressed by John H. Van Deventer, editor of *The Iron Age*, on Oct. 3, dealt with the "Economic Outlook." A demand within the next few months from those European countries, which recently evaded temporarily the threat of war but which inevitably must face a final show-down, will keep makers of American production machinery working overtime, Mr. Van Deventer predicted. "While this may not directly stimulate the export of the products of the automobile industry in like proportion, it is bound to stimulate the home demand," Mr. Van Deventer said, adding, "for when you shake even one corner of America's closely-knit industrial fabric, the tremor passes over all of it."

Turning directly to American industrial activity, Mr. Van Deventer declared that in his belief the steel industry still furnishes the best barometer of industrial activity. "When steel production is good," he pointed out, "business is good. On July 6 of this year, steel production in the United States was 23% of capacity. Now steel production is running in excess of 48% of capacity. I still hold the opinion I expressed early in August of this year that before October is ended we shall see a steel rate close to 60%. That sustained increase week after week is encouraging in itself, but to my mind the most encouraging feature of this steady gain is the fact that what is called 'miscellaneous uses' of steel have contributed most largely to it."

Speakers at the air-conditioning session, Oct. 17, offered suggestions for further research on this subject of coming commercial importance. Regarding "Temperature Sensations in Automobile Bodies," Allen C. Staley, Chrysler Corp., analyzed physiological requirements and physical limitations to insuring passenger comfort from a temperature viewpoint. His approach to the subject was paralleled by that of Prof. F. J. Linsenmeyer, University of Detroit, who stated, quoting the American Society of Heating and Ventilating Engineers, that the term air-conditioning must include: "Controlled heating and cooling; humidification and dehumidification; ventilation; distribution of air; and purification of air."

Much valuable and interesting data concerning actual passenger requirements were included by Mr. Staley in his discussion. Upon study of zero-day conditions and the consequent problem of frosted windshields, he concluded that 30% relative humidity inside the car, with air temperature at 60 F, would be a satisfactory condition. Further, he added that since a car with five passengers will give off about 500 Btu per hr in latent heat, with about 3500 grains of water, something like 45 cu ft of fresh air per min is needed (with outer air at 50% relative humidity, assumed) to prevent condensation. In a similar analysis of summer cooling, he declared that a satisfactory heating or cooling plant for an automobile requires an unusual degree of flexibility and quick response to load changes, involving a control system.

An adequate and modern review of present car heaters of the exhaust, water and steam types was included by Prof. Linsenmeyer, who also touched on the newer fuel combustion type heater. Of air-cleaning he said that dust contents in Detroit is upward of 60,000 lb per sq mile per month deposited. Auto cooling and ventilating systems will draw this dust into the car, along with the air that has (at street line) 2 to 7% carbon monoxide. These facts, he said, indicate need for air cleaners and purifiers on completely air-conditioned cars.

The full schedule of meetings for the month was concluded Oct. 25 by a joint Junior-Student-Aeronautic session. J. M. Crawford, chief engineer of the Chevrolet Motor Division of General Motors Corp., was the first speaker. Introduced by A. C. Hazard, chairman of Junior-Student activity, Mr. Crawford delivered an inspirational address on some of the more important personal factors of professional success, outlining for the embryo engineers the "Requirement of Modern Engineering."

A panoramic picture of national defense, with particular emphasis on the participation of aircraft in plans of the United States Army, was presented in technical form by Major George B. Tourtellot, U. S. Army Air Corps. The officer, assigned as an instructor to the 107th Observation Squadron, Michigan National Guard, has a background of 21 years in Army service. Primary object of our Air Force, he said, will be defensive. All our present plans are based on efforts to halt invaders. First it will expend its efforts in counter air force activity in an attempt to prevent any enemy air force from getting established bases or operating areas. If an enemy were to accomplish this much, Major Tourtellot declared, their next effort would be to land troops. In such an event, our Air Force would concentrate on their supply ships and seaplane bases. The American Air Force on which we rely for this defensive and protective program is much smaller than the probable coalition which might be expected to dare attack, he declared, but would have the advantage of having bases already operating; whereas the enemy would be seriously harassed in its attempt to set up useable bases. Major Tourtellot emphasized the declaration that the nation's air power "is not potential; it is actual." He added that the War Plans Section estimates that it will take six months at least to supplement our air force in an emergency.

More Frequent Oil Changes Predicted

● **New England**

More frequent oil changes are indicated by trends in car design, according to Paul V. Keyser, director of laboratories, Socony-Vacuum Oil Co., Inc., who presented his paper, "Lubrication – What, Where, When and How," before the Oct. 25 meeting of the New England Section. The tendency is toward specifying oils of lower viscosity and this, he said, will necessitate changing oil more often, despite car manufacturers' claims that less frequent changes are required for cars of today. He also remarked that oil filters do not do a thorough job, because they only stop heavy substances.

Mr. Keyser discussed all phases of automotive lubrication, and his presentation stimulated lively discussion which carried the meeting long past its scheduled closing time.

This was the New England Section's first meeting of the 1938-1939 season. Chairman John W. Lane presided and introduced the other new officers. Attendance was close to the peak recorded last year, indicating great interest in the Section's activities.

'Tailor-Made' Fuels Predicted by Frey

● **Cleveland**

We can be assured that fuels of the future will be "tailor-made" and designed to give a certain type of performance, Julian J. Frey, Ethyl Gasoline Corp., declared in his paper, "Behind the Scenes of Today's and Tomorrow's Fuels," presented at the Oct. 10 meeting of the Cleveland Section.

Mr. Frey bases this conclusion on the fact that refining has made such great strides in recent years. He noted that it has progressed through various stages, including simple distillation, cracking, hydrogenation, and polymerization, and that the period of accepting any product which may be produced in large quantities from any crude that may happen to be available is passing.

Progress in refining has been paralleled by improvements in engine design which have made fuels which were not considered satisfactory a few years ago quite adequate for ordinary performance, he said. Mr. Frey also noted that laboratory work and practical experience in aircraft transport operations show that major improvements in both power output and fuel economy are obtainable from higher octane fuels than generally are distributed at this time. Although these fuels are considered special in today's market, he continued, they are being used in million-gallon quantities in aircraft service this year. These operations show that major developments are possible in both fuels and in the engines which burn them that will produce worth-while improvements in the power-weight ratio and in fuel economy, he added.

It is expected that development in the near future will be towards fuels above 100-octane rating, and that engine improvements will probably include the use of superchargers, he concluded.

R. E. Bolz, winner of the student paper contest, which the section sponsored last season, was awarded a $50 scholarship at the Case School of Applied Science, where he is a student, and a student membership in the SAE. The presentation was made by Hoy Stevens, chairman of the Section's student activity committee. W. T. Gloor, W. B. Keller, and D. W. Steel, also students at Case, received student memberships in the Society as honorable mention awards for the papers which they submitted.

Describes Maintenance System of 500-Bus Fleet

● **St. Louis**

How the St. Louis Public Service Co. maintains their 500 motorcoaches which travel 16,000,000 miles per year over the 270 miles of bus routes that comprise its system, was explained to the St. Louis Section, Oct. 26, in a paper by its chairman, Adam Ebinger, who is superintendent of the company's garages. J. E. Jury, chairman of the Section last year, read Mr. Ebinger's paper.

More than 75 members and guests attended the meeting which included an inspection trip through the shops of the St. Louis Public Service Co.

Explaining that experience has shown that a complete system of unit maintenance is the background of any preventive maintenance plan, Mr. Ebinger gave a detailed report of the method followed by his company.

"For each new piece of equipment," he said, "each type of unit is studied and examined in order to arrive at an accurate estimate of the mileage expected of it. During this initial period, when it is undergoing close scrutiny, the unit is frequently removed from the coach merely for the purpose of examining its condition and for verification of the life estimate previously established. When the life expectancy of a unit has been determined finally, all units of the same type are expected to operate economically and satisfactorily during this mileage period. Frequently, as various methods of improved control are discovered and adopted, the life expectancy period is moved into higher and higher brackets.

"Our operating background has established that this method is fundamentally sound, and there are relatively few units removed because of failure. After the life expectancy has been determined, it becomes a routine matter for the record office to control the removal of these units from the coaches and, by judicious timing, to insure an even flow into the automotive shop.

"A most important feature of this system is that it is possible to plan the shop work several weeks – even months – in advance so that shop output can be budgeted. This spreads maintenance costs more evenly over longer periods of time."

Adam Ebinger, vice-chairman of the St. Louis Section, advanced to Section chairman to fill the vacancy which occurred when W. Rex Brashear resigned as chairman because of moving to New York. The new vice-chairman elected by the Section is Joseph Sunnen of the Sunnen Products Co.

Traces Evolution of Distillation Processes

● **Canadian**

From the wines of ancient Egypt to the gasolines of today, was the course taken by Dr. Gordon McIntyre, manager of the technical service department, Imperial Oil, Ltd., in tracing the evolution of the process of distillation in his paper, "Gasoline," presented before the Canadian Section, Oct. 19.

Dr. McIntyre pointed out that there are possibly a thousand different ingredients in gasoline and that molecular structure, form, size and association of these ingredients determine performance characteristics. It is through the alteration of these by the division of some and the coalescence of others that some of the desirable mutations are effected in gasoline, he added in briefly referring to the cracking and catalytic processes.

In reply to one of the many questions advanced during the discussion period, Dr. McIntyre dealt with the development and functioning of the gum-solvent oil used in some gasolines, and presented a short film covering the subject. The sound picture, "Designed For Power," was also shown.

Automobile Engines Not Efficient Noise Makers

● **Baltimore**

When it comes to making noise the modern automobile engine is less than 0.000214% efficient, R. F. Norris, president of Norris Industries, Inc., told members and guests at the Oct. 6 meeting of the Baltimore Section. As a consequence, he said, we cannot claim that reducing the noise of an engine materially increases its efficiency. However, he continued, a great deal can be done to suppress the noise and in this way add to the comfort and health of the passengers.

Mr. Norris explained that a loud noise produces in the human system a physiological sensation, known as the fear reaction. This draws the muscles of the solar plexus tightly together and glandular disturbances and secretions occur which unbalance the system. He likewise pointed out that when driving a car in a noisy environment the energy necessary to overcome the noise must be supplied by the operator, making him tire much more readily. It was also noted that noise interferes with the power of concentration causing a person to make errors of judgment which he would not make in a quiet place.

"From the standpoint of the engine and the insulation and isolation of the body, I would say that the present day automobile is sufficiently quiet," Mr. Norris remarked, after describing how various units have been silenced. However, he continued, engineers apparently have either overlooked or disregarded one noise, and that is wind rustle. At high speeds, he said, the air rush creates eddies which can make a great deal of noise. They may beat on the outer surface of the car with enough force to cause a vibration of the metal surfaces which transmit a high-pitched wind hiss into the car or, if the windows are open, eddies will be formed around the edges of the window which produce a truly deafening sound. Mr. Norris explained that this has been aggravated to a great extent by the manufacturers' adoption of ventilating systems using deflecting vanes on the front windows.

If the public's demand for quiet becomes great enough, he concluded, the automobile manufacturer will develop methods of introducing sufficient air into the car for ventilation and cooling without generating the terrific noises which they at present seem to disregard entirely.

Rabezzana Addresses First Meeting of Year

● **Pittsburgh**

The Pittsburgh Section opened its 1938-1939 season at a dinner meeting, Oct. 19. More than 70 were on hand to hear Hector Rabezzana, chief engineer of the AC Spark Plug Division of General Motors Corp., present his paper on "Spark Plugs and Their Relation to Motor Efficiency." Participating in the discussion were B. H. Eaton, Ralph Baggaley, Jr., Charles R. Noll, A. R. Platt and many others. A slide film on ignition problems followed the discussion.

E. F. Lowe, assistant general manager of the Society, was a guest of the Section and was called upon to say a few words at the dinner which preceded the technical session. He was introduced by Section Chairman Robert N. Austen.

Student Branch Plans For Coming Activities

● **Oregon State**

Plans for the SAE skit for the Engineers' Bust; the student-paper contest, sponsored by the Oregon Section, and expansion of membership, were high spots in the business meeting of the SAE Student Branch at Oregon State College.

After business matters were taken care of, Chairman Harold Ager introduced Earl Marks, secretary of the Oregon Section. Mr. Marks compared old types of ignition system with new systems. He called particular attention to three changes in coil design, namely: putting the primary winding on the outside; a new magnetic core; and removing resistance to regulate current. He also described automatic spark control, listed its advantages, and emphasized the importance of correct timing.

(*News of Society continued on page* 16)

THE S.A.E. Annual Meeting

DETROIT JANUARY 9-13

BOOK-CADILLAC HOTEL

HIGHLIGHTS

on the 5-Day Program
of 20 Technical Sessions

Sound engineering papers embracing practically every SAE Activity . . . . . A day devoted to National Defense . . . . . A Special Student Session . . . . . All, coupled with one of the best SAE Dinner programs ever planned, give a thumb-nail description of the 1939 Annual Meeting.

Key men in the Nation's Armed Forces, including the Quartermaster General, the Chief of Ordnance, the Chief of Air Corps, and the Assistant Chief of the Navy's Bureau of Aeronautics, will be guest speakers on National Defense Day, a vital innovation.

The SAE Dinner will be part of the Defense Day Program — falling on Wednesday this year.

Civil as well as military aeronautical problems will be attacked in the five Aircraft and Aircraft-Engine Sessions. The F&L program will include such topics as detonation and fuel distribution. Tires will be highlighted at the passenger car sessions. T&M men will devote one session to safety and another to maintenance. The Diesel Activity has scheduled papers on instrumentation and piston temperatures. Thirty topics, in all, will be discussed.

Of particular interest will be the Engineering Exhibit. More than 30 manufacturers will have displays.

THE ANNUAL MEETING

Dinner

CLIMAXES NATIONAL DEFENSE DAY

WEDNESDAY EVENING JANUARY 11

Dinner restricted to SAE Members. Make reservations through SAE Headquarters at:

29 West 39th St., New York, N.Y.

Price $3.50 per place

1939 Changes Analyzed At Annual Show Meeting

● Chicago

Functional streamlined styling and compelling eye appeal give a flashing appearance to 1939 cars; but equally impressive, and more important to car performance, are the mechanical improvements in the new cars. This was the message Joseph Geschelin, Detroit technical editor, Chilton Co., Philadelphia, brought to more than 200 members and guests of the Chicago Section at its Annual Auto Show Meeting and Dinner.

Spotlighting of these 1939 mechanical changes was admirably done by word and slides by speaker Geschelin at this annual event held Armistice Day evening, at the Famous Saddle & Sirloin Club adjoining the International Amphitheatre where Chicago's Auto Show was staged.

The eye of the observer sees in the gleaming new cars arresting triple radiator grilles, more daring profiles, inset fender headlights, and advanced body streamlining with greatly increased glass areas in windows and windshields, the speaker pointed out, but more basic in character, he said, are several engineering change trends of importance, which are to be seen for the first time in this year's cars. These changes he grouped in three divisions: (1) improvements in riding comfort through basic changes in suspension; (2) basic changes in clutch design; and (3) basic changes in transmissions and transmission control.

In rear spring design, one maker incorporates a "variable rate" spring which gives the same ride whether the load is one passenger or six. Another form of suspension incorporates a fifth shock absorber in the rear stabilizer and a special friction element in the front shock-absorber arms, he noted. This new development is said to give complete freedom from road shock irregularities due to washboard road surfaces.

In clutch design, the trend toward simplification is evident, Mr. Geschelin noted. One design, he said, embodies a single diaphragm spring, which eliminates many of the conventional parts. Another maker has developed a similar clutch design which has only nine parts. The conventional clutch is redesigned in several cases with the view to eliminate parts and provide greater simplicity.

Remote control transmissions feature the biggest engineering changes, according to the speaker who noted that most cars offer, as standard or optional equipment, a hand control operated under the steering wheel. Most transmissions have been completely redesigned and fitted with synchronizers which provide for the use of hand shifts, he said. New automatic overdrives of the type always under control of the driver are provided as optional equipment on a number of makes, it was stated.

With Ford's adoption of hydraulic brakes, this type brake is now practically universal, Mr. Geschelin pointed out. Powerplants, he said, have been improved in one case by a special finishing process applied to major motor parts. New piston rings with specially treated surfaces are incorporated in many new powerplants. The result, he reported, is said to be increased life of parts and lower oil consumption.

Following his talk, Mr. Geschelin responded to a number of questions pertaining to status of the fluid flywheel, diesel engine application in passenger cars, and the factors which limit the application of diesel trucks to specialized zones of activity.

Students to Compete In Speaking Contest

● Ohio State

The SAE Student Branch at Ohio State University is planning a competitive speaking contest. Members will participate and prizes of $10 and $5 will be awarded to those placing first and second.

Officers for the first quarter were elected at the Oct. 20 meeting of the Branch. J. A. Scerba was named chairman; W. E. Harpst, vice-chairman; L. A. Dever, secretary; and J. M. Heldack, treasurer.

Meetings of the group are scheduled for each Tuesday.

Joint Smoker at N.Y.U.

A joint smoker of the New York University student branches of the SAE, American Society of Mechanical Engineers, ASME (Aero), and the Institute of the Aeronautical Sciences, Nov. 2, was addressed by Robert Johnson, field engineer, Wright Aeronautical Corp. His topic was "Relations Between Engine and Aircraft Manufacturers." About 50 students attended.

SAE *Coming* EVENTS

Annual Meeting
and Engineering Display
Jan. 9-13, 1939 Detroit, Mich.
Book-Cadillac Hotel

World Automotive Engineering Congress
May 22-June 8, 1939

Baltimore – Dec. 1

Longfellow Hotel; dinner 6:30 P.M. Factors Influencing Engineering Organization – James M. Crawford, chief engineer, Chevrolet Motor Division, General Motors Corp.

Buffalo – Dec. 13

Hotel Statler; dinner 6:30 P.M.

Canadian – Dec. 14

Royal York Hotel, Toronto; dinner 6:30 P.M. New Firing Lines for Engineers – C. W. Spicer, vice-president, Spicer Manufacturing Corp., and president of the SAE. Truth Through Torture – John A. C. Warner, secretary and general manager of the SAE.

Chicago – Dec. 6

Medinah Club; dinner 6:30 P.M. Keeping Up to Date on Motor Fuels. A symposium by T. H. Risk, automotive engineer, Pure Oil Co.; T. B. Rendel, director, Engineering Research Laboratories, Shell Petroleum Corp.; W. G. Ainsley, in charge of engineering research, Sinclair Refining Co.; D. P. Barnard, associate director of research, Standard Oil Co. of Indiana; and Gustav Egloff, director of research, Universal Oil Products Co.

Cleveland – Dec. 12

Cleveland Club; dinner 6:30 P.M. Engineering Developments for 1939 – Austin M. Wolf, automotive consultant.

Detroit – Dec. 5

Hotel Statler; dinner 6:30 P.M. What About the Engine? – Alex Taub, power unit engineer, Vauxhall Motors, Ltd.

Indiana – Dec. 8

Antlers Hotel, Indianapolis; dinner 6:30 P.M. An Analysis of Current Automobile Design – Joseph Geschelin, Detroit technical editor, Chilton Co.

Kansas City – Dec. 9

Hotel Kansas Citian; dinner 6:30 P.M. The Automatic Transmission – H. N. Metzel, development engineer, Olds Motor Works Division, General Motors Corp.

Metropolitan – Dec. 1

Hotel New Yorker, New York City; dinner 6:30 P.M. High Output Aircraft Engines – H. C. Tsien, Massachusetts Institute of Technology, and member National Resources Commission in China. Discussion by Maj.-Gen. H. H. Arnold, Chief of the United States Army Air Corps.

Milwaukee – Dec. 9

Milwaukee Athletic Club; dinner 6:30 P.M. Engineering Problems Pertaining to Cold Weather Lubrication – M. J. Zucrow, consulting engineer.

New England – Dec. 13

Engineers Club, Boston, Mass.; dinner 6:30 P.M. The Motor Clinic – R. J. Coveney, Ethyl Gasoline Corp.

Northern California – Dec. 6

Hotel Bellone, San Francisco; dinner 6:30 P.M. Two-Cycle Diesel Engines – F. G. Shoemaker, chief engineer, Detroit Diesel Engine Division, General Motors Corp.

Northwest – Dec. 9

Hotel Roosevelt, Seattle, Wash.; dinner 6:30 P.M. New Emphasis in Industrial Management – N. J. Aiken, placement director, Washington State College.

Oregon – Dec. 9

Lloyd's Golf Club, Portland. Pertinent Automotive Maintenance Problems of the Forestry Service – T. P. Flynn, senior equipment engineer, U. S. Forestry Service. Oil Filter Testing Procedure – G. B. Hall, test engineer, U. S. Forestry Service.

Philadelphia – Dec. 14

Engineers Club; dinner 6:30 P.M.

Pittsburgh – Dec. 13

Hotel Webster Hall; dinner 6:30 P.M. What's Ahead for the Automotive Industry – Norman G. Shidle, executive editor, SAE JOURNAL.

St. Louis – No Meeting

Southern California – Dec. 13

Mona Lisa Cafe, Los Angeles; dinner 6:30 P.M. Automotive Two-Cycle Diesel Engines – F. G. Shoemaker, chief engineer, Detroit Diesel Engine Division, General Motors Corp.

Southern New England – Dec. 13

New Departure Club, Bristol, Conn.; dinner 6:30 P.M. Bearing Materials, Manufacturing Practices and Failures – C. F. Smart, metallurgist, Pontiac Motor Division, General Motors Corp.

Syracuse – No Meeting

Washington – Dec. 13

Cosmos Club; dinner 6:30 P.M. Modern Transportation – Walter A. Olen, general manager, Four-Wheel Drive Auto Co.

Your Society

SAE Relations with Outside Agencies

By Philip H. Smith

EVER since the founding of your Society, there has been very close cooperation with outside agencies and organizations. Link after link has been forged to meet specific needs as they arose, so that today, with connections far and wide, it can be said truthfully that the SAE sun never sets on the automotive world.

When the founding fathers of the Society launched the first cooperative effort it was with no preconceived idea as to the ultimate form of association with the outside world. When there was need to standardize steels, for instance, they went to work with the steel industry. And ever since when there has been something needing to be done, the mutuality of interest has been stressed, and another link has been added.

The extent of relationships with the outside world can be seen by referring to the ROSTER or the HANDBOOK. Under the heading, "sectional committees under American Standards Association procedure" there are 46 cooperative activities, while some two score are classified under "SAE representatives on other organizations and committees." To this listing should be added the work of the Engineering Relations Committee, the Ordnance Advisory Committee and several others, because they, too, cooperate closely with other organizations.

Aside from the fact that cooperation characterizes all these activities, and thus creates a similarity, there is a great difference between groups in other respects. There is wide variation in the degree of participation by the Society, in the duration of the association, and in the methods of operation. Need is always the arbiter. It gives rise to the association, determines the manner in which the Society will contribute, and makes the link active or quiescent. The number of cooperative activities at any given moment, therefore, reflects the needs of the moment.

Some of the associations have been of long duration. Your Society has worked with the Automobile Manufacturers Association almost from the first day, and now is engaged in joint work on safety matters involving headlighting, brakes, and the like, in cooperation with the Automotive Safety Foundation. Long standing cooperative enterprise has been carried on with such agencies as the Bureau of Public Roads, the Bureau of Standards, the American Standards Association, and the American Society for Testing Materials. Your Society is now working jointly with the ASA on the standardization of screw threads, ball bearings, safety glass and motor-vehicle inspection, while with the ASTM, it works on such problems as steels, castings, petroleum, rubber, and testing procedure.

One of the most outstanding examples of joint endeavor is the Cooperative Fuels Research program carried on with the American Petroleum Institute, the Automobile Manufacturers Association and the Bureau of Standards. The work of this joint association has been discussed in detail in a previous article so its fruit must be well understood. But it is outstanding in revealing what can be and is accomplished by means of active linkage.

The growth of work with outside organizations parallels the growth of the automotive industry and its limits cannot be foreseen unless one determines what lies beyond the horizon. That there would come to be joint work in the field of radio hardly could have been prophesied 10 years ago, yet today, cooperative work with the Radio Manufacturers Association occupies a place on the Society's program. The problem giving rise to this work is the interference with reception on police radio cars and the need is to discover fundamental causes and correct the situation. From the automobile industry's standpoint this is not a pressing matter because motorists do not use low wave bands, but announcements that commercial television will become a reality very shortly may make it imperative to find the answers.

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This is the *seventh* article of a series on "Your Society."

This series aims to bring alive every phase of Society functioning as related to the individual member.

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Cooperative enterprise spreads far beyond the confines of the nation's borders to give international scope to contact with outside agencies. There has been intermittent work in international standardization over many years, in so far as the means to cooperation have been available. At the moment, international standards work is expanding along two channels – through direct representation with foreign organizations when the needs are strictly automotive in character, and through the American Standards Association which is a member of the International Standards Association.

Of course, the expansion of these outside relationships has increased the work of the Society and it has placed a heavy burden upon the Council, because it is the duty of this governing body to determine the validity of proposals for joint work, to decide upon the extent of Society's participation, and to oversee the Society's activity once work is undertaken. In recent years it became physically impossible for the Council to handle adequately the far flung relationships with the existing machinery of organization and thus, in 1935, the Engineering Relations Committee was organized to assist with the burden. This Committee has the responsibility for investigating all proposals involving work with outside agencies and for reporting its findings to the Council. In this respect it functions as a fact-finding body.

Engineering Relations Committee

The Engineering Relations Committee holds much the same position in the Society's structure as does the Grading Committee which investigates applicants for membership and grades them properly to take detail work off the shoulders of the Council. Both Committees came into being to fill needs created by the growth of the Society and both make a distinct contribution towards efficient operation. But this Committee in addition to being a fact-finding body, is a coordinating group in that it acts as a sort of clearing house for all the diverse activities of the Society which contact with the outside. It does not originate any technical material, but often brings forward material developed by the technical committees for better availability.

There was a time when almost all the Society's work with other organizations had to do with the technical side of design, production and maintenance of automotive products. However, as the automotive industry has matured, more and more attention has been given to the use of motor vehicles. Governmental agencies, educators, and safety groups, as well as many other outside agencies have become greatly concerned with their operation upon the highways. Many of the problems, most vital to the automotive industry tomorrow, are originating, therefore, quite outside the industry and with organizations which have not figured earlier as ones with which the Society habitually cooperated. The Engineering Relations Committee brought to your Society adequate organization with which to serve these newly concerned groups. The Society is empowered to discover and to give out engineering fact, but it has never had the authority to become involved in matters of opinion.

It is very logical for such agencies to turn to your Society for cooperation. Its prestige alone is sufficient to warrant this. An official whose duty it is to work for highway safety can assume with every reason that the Society can provide factual data to help in the solution of headlighting and braking problems, for instance. Yet no officer, no committee, nor any single member has the right to commit the Society to an opinion.

There has been cooperation with outside agencies at all times in the past, but the growth of the Society and the expansion in number of such agencies has made proper liaison a complex matter. There has been a two-way need. There has been call for a centralized agency with responsibility for coordinating the work of the many technical committees and making it readily available for outside agencies – some person or group having the time to do a good knitting-together job. There has also been need for a similar responsibility to keep in touch with what goes on outside the Society and bring the information back to headquarters so that the Council, committees and members benefited from it to the full extent. In short, the pressure was for more flow of material and information back and forth, and it was the realization of this fact among others that put the Engineering Relations Committee to work.

One of the duties of this Committee is to help the Council see to it that no unauthorized expressions of opinion are uttered in the name of the Society. There is too much prestige at stake to have it thrown carelessly behind any movement that any single member happens to advocate.

This Committee has made already a profound contribution to highway safety by facilitating – on request – the provision to officials of the findings of technical committees upon which regulation might be based. By supplying data on brakes, headlamps and the like, which are permissible facts, regulations have tended to be sane rather than punitive. This is of great value to every member of the Society because it helps to keep free the broad fields in which engineering progress can be made.

In the work of this Committee we have an example of three things which characterize every SAE association with outside agencies – every link enables the Society to exercise its force in the common good and to the extent that it does so its prestige is enhanced; every association strengthens further the prestige of the Society by focusing attention upon its dominant position, and, finally, these many links give the individual member a larger stake in the engineering world and afford him wider participation in organization work without at the same time exacting from him more dues, more outlay of time and effort.

New York and San Francisco Fairs Set 1939 SAE World Congress Days

BRILLIANT added attractions for those who attend the 1939 World Automotive Engineering Congress of the SAE will be visits to the New York World's Fair during the opening days of the Congress, and to the Golden Gate Exposition, during the closing week at San Francisco. Both of these huge expositions will open a few weeks before the Congress convenes.

May 24 has been set aside by New York Fair officials as "Society of Automotive Engineers' Day," and June 7 has been designated as "SAE World Automotive Engineering Congress Day" at the San Francisco Exposition.

The SAE World Congress opens in New York on May 22 for a five-day technical program and the big Congress Dinner. Continuing across the continent the Congress will visit Indianapolis for the annual 500-mile race and a gala dinner on May 29 and 30. The next three days will be spent in Detroit where the program will provide for visits to automotive plants, technical sessions and a dinner. The Congress will then board streamlined diesel trains for the three-day technical session in San Francisco. The West Coast Dinner, June 8, will conclude this great meeting.

More than 70 technical papers and hundreds of prepared discussions are being scheduled for the Congress, which is expected to be one of the outstanding events in automotive history.

New Members Qualified

These applicants who have qualified for admission to the Society have been welcomed into membership between Oct. 15, 1938, and Nov. 15, 1938.

The various grades of membership are indicated by: (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate Member; (SM) Service Member; (FM) Foreign Member.

Canadian Section

SMITH, JOHN GODFREY (A) president, Federal Wire & Cable Co., Ltd., Guelph, Ont. (mail) 265 Suffolk St.

Chicago Section

TOBIN, THOMAS T. (M) chief engineer, E. Edelmann & Co., 2332 Logan Blvd., Chicago (mail) 2743 North Campbell Ave.

Cleveland Section

GOODAIRE, EDWIN W. (A) owner, Ring-Free Oil Co. of Cleveland, 2775 Pittsburgh Ave., Cleveland.

MCCREA, CHARLES H. (M) general manager, National Malleable & Steel Castings Co., 10600 Quincy Ave., Cleveland.

Dayton Section

HARRISON, JOHN F. (M) mechanical engineer, Huber Mfg. Co., Marion, Ohio (mail) 139 Merchant Ave.

Detroit Section

BEYSTER, HENRY E. (M) Commissioner, Public Works, City of Detroit, Room 104, City Hall, Detroit.

HUBBARD, JAMES ALFRED (A) secretary, M. D. Hubbard Spring Co., 217 East Central, Pontiac, Mich.

Metropolitan Section

CROOK, FRANK (A) salesman, Raybestos Division, Raybestos-Manhattan, Inc., Bridgeport, Conn. (mail) Jennings Road 980, R.F.D. 6, Fairfield, Conn.

O'NEILL, JAMES GEO., JR. (M) staff engineer, Sinclair Refining Co., 630 Fifth Ave., New York.

PETERSON, H. A. (M) assistant to vice-president, American Gas & Electric Service Corp., 30 Church St., New York.

REMO, JOHN G. (A) salesman, A. Jacoby & Sons, Inc., 8612 18th Ave., Brooklyn, N. Y. (mail) 680 81st St.

Northwest Section

DAFOE, LESLIE R. (J) draftsman, Kenworth Motor Truck Corp., Seattle, Wash. (mail) 3818 43rd North East.

PITZER, ERNEST VICTOR (A) salesman, General Parts Co., 1714 Broadway, Seattle.

SHELL OIL CO. (Aff.) 1219 West Lake North, Seattle. Rep: Davis, E. A., division sales.

Southern California Section

CRAIG, BURNIE MCDONALD (J) designer, draftsman, California Institute of Technology, Astrophysics Dept., 1201 East California St., Pasadena, Calif. (mail) Suite 17, Arcade Bldg.

SMITH, W. S. (A) assistant refining engineer, Wilshire Oil Co., Inc., 1500 Carmineta, Norwalk, Calif.

Syracuse Section

ANDERSON, JOHN W. (M) manager, engineering, American Locomotive Co., 100 Orchard St., Auburn, N. Y.

Washington Section

EWBANK, WALTER JAMES (J) research engineer, Briggs Clarifier Co., 3262 K St., North West, Washington, D. C. (mail) 3044 N St., North West.

Outside of Section Territory

HENNEY, JOHN W. (M) president, Henney Motor Co., Freeport, Ill.

PHILLIPS PETROLEUM CO. (Aff.) Bartlesville, Okla. Reps: Alden, R. C., director of research; Hansen, C. S., lubrication engineer, sales dept.; Henry, R. W., chief chemist, Okmulgee Refinery; Martin, C. J., manager, automotive maintenance dept.; Oberfell, G. G., vice-president, charge of research; Parker, Billy, manager, aviation sales div.

WARD, FRED R. (M) superintendent of equipment, Mistletoe Express Service, Inc., 10 North West Fifth St., Oklahoma City, Okla.

Foreign

ANDREWS, HAROLD (F M) managing director, Harold Andrews Grinding Co., Ltd., Bristol Road, Birmingham 29, England.

BURTON, ALEXANDER T. (M) engineering test pilot, North American Aviation, Inc., Inglewood, Calif. (mail) No. 27 Maintenance Unit, Royal Air Force, Shawbury, Shrewsbury, Salop, England.

ORNSTEIN, WILHELM (F M) engineer, technical div., Lilpop, Rau & Loewenstein, 65 Bema St., Warsaw, Poland (mail) A1, Jerozolimskie 18-4.

TOLLENAERE, PIERRE L. E. (A) staff technical inspector, Ets. Daniel Doyen, S.A., 31-32 Blvd. du Midi, Brussels, Belgium (mail) 57 Ave. Michel-Ange.

Applications Received

The applications for membership received between Oct. 15, 1938, and Nov. 15, 1938, are listed herewith. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

Baltimore Section

FASIC, GEORGE G., manager of warehouse, Bowers Battery Mfg. Co., Baltimore.

THOMPSON, JOHN T., standards engineer, The Glenn L. Martin Co., Baltimore.

Buffalo Section

GEORGE, WALLACE LEE, sales engineer, Marlin-Rockwell Corp., Jamestown, N. Y.

Canadian Section

BELL, W. THOMAS A., lubrication sales manager, Imperial Oil Ltd., Toronto, Ont.

EBERTS, HERMANN L., mechanical supt., Montreal Tramways Co., Montreal.

Chicago Section

AASLAND, BYRON T., designer, David Bradley Mfg. Works, Bradley, Ill.

BOYD, LANDON B., vice-president and general manager, De Luxe Products Corp., La Porte, Ind.

STREBINGER, RICHARD, Bendix Products Corp., South Bend, Ind.

Cleveland Section

FEISS, GEORGE JAMES, president and general manager, Superior Foundry Co., Cleveland.

PINSENSCHAUM, EDWIN C., engineer, Diesel Division, Timken Roller Bearing Co., Canton, O.

Dayton Section

BLANCHARD, W. J., president, Engineering Projects, Inc., Dayton, Ohio.

ZINN, ROBERT W., draftsman and assistant in engineering department, Armstrong Furnace Co., Columbus, O.

Detroit Section

ANKLAM, CHARLES W., chief engineer, C. M. Hall Lamp Co., Detroit.

BLOMQUIST, ANFRED E., sales manager, McCord Radiator & Mfg. Co., Detroit.

BURGESS, MAGNUS M., president, Sheller Mfg. Corp., Detroit.

GRAHAM, CHARLES RICHARD, engineer, Graham-Paige Motors Corp., Detroit.

HORVATH, LEWIS M., JR., chassis units layout, Chevrolet Motor Division, General Motors Corp., Detroit.

HUME, JAMES W., vice-president, Harley C. Loney Co., Detroit.

KALMER, STEPHEN, research engineer, AC Spark Plug Division, General Motors Corp., Flint, Mich.

KILDOW, WILLIAM H., JR., manager, Industrial sales, Shell Petroleum Corp., Detroit.

LEEDER, ERNEST ROBERT, resident manager, Fisher Body, Pontiac Division, Pontiac, Mich.

PLACE, CHARLES EDWARD S., sales engineer, National Machine Products, Detroit.

RANDALL, FOREST C., Chevrolet Motor Division, General Motors Corp., Flint, Mich.

REED, CLAIR S., assistant factory manager, Reynolds Spring Co., Jackson, Mich.

SMITH, TORRANCE B., service engineer, American Brass Co., Detroit.

VOIGT, J. S., chief engineer, Graham-Paige Motors Corp., Detroit.

Indiana Section

BAKER, CHARLES HENRY, JR., Warner Gear Division, Borg-Warner Corp., Muncie, Ind.

BRADY, THOMAS ROBERT, factory manager, American Bantam Motor Car Co., Indianapolis.

Metropolitan Section

BRYANT, ELWOOD D., test engineer, Wright Aeronautical Corp., Paterson, N. J.

GILLE, HUGO EDWARD, junior engineer, Ethyl Gasoline Corp., New York.

HAMILTON, WILLIAM S. H., equipment electrical engineer, New York Central Railroad, New York.

HEALEY, CHARLES D., maintenance supervisor, Tide Water Associated Oil Co., New York.

VARRICCHIO, FREDERICK O., foreman, Hudson Laundry, New York.

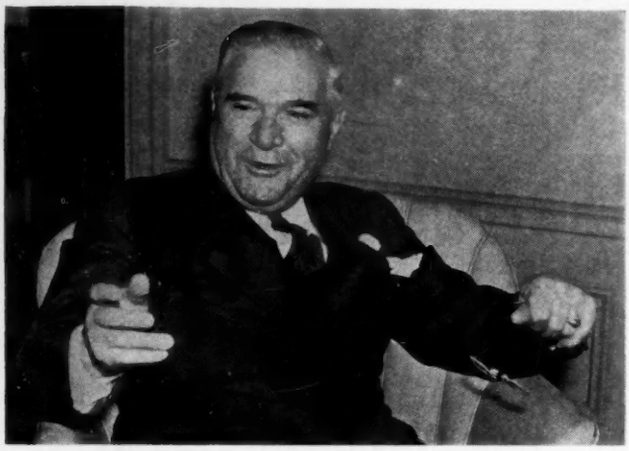
WILLS, JACKSON T., experimental test engineer, Wright Aeronautical Corp., Paterson, N. J.

Milwaukee Section

KAMLOOKHINE, IGOR, engineer, Allis-Chalmers Mfg. Co., Milwaukee.

(*Concluded on page 26*)

About SAE Members:



JULIUS P. HEIL was elected Governor of Wisconsin in the general election last month. This victory climaxed his first venture into politics. Mr. Heil is president-treasurer of the Heil Co., Milwaukee, which he founded, and he has been a member of the SAE since 1921.

Acme

JOHN M. COLLINS, since 1933 treasurer of the Moore Drop Forging Co., Springfield, Mass., has been elected president of the company. He will continue to hold the office of treasurer.

FREDERICK KUHLEN has been named chairman of the Department of Mechanical Engineering of the College of the City of New York.

JOHN A. HERLIHY has been appointed vice-president, operations, of United Air Lines, with headquarters in Chicago. He previously was director of engineering, United Air Lines Transport Corp.

KENNETH B. CHAMP recently resigned as general sales manager of Mid-West Abrasive Co., Detroit, to become president of Smith Brothers Mfg. Co., Findlay, Ohio, manufacturers of automotive replacement parts and accessories.

SAE Speakers

IGOR SIKORSKY, vice-president in charge of engineering, Sikorsky Aircraft Division of United Aircraft Mfg. Corp., before the Lilienthal Society for Aviation Research, Berlin, Oct. 12. . . . **ALFRED P. SLOAN, JR.,** chairman of General Motors Corp., before the Boston Chamber of Commerce, Nov. 3. . . . **F. C. CRAWFORD,** president, Thompson Products, Inc., before the Investment Bankers' Association, White Sulphur Springs, Oct. 27. . . . **PAUL G. HOFFMAN,** president of Studebaker Corp. and the Automotive Safety Foundation, before the American Trucking Associations, Inc., in Detroit early last month. . . . **EDWARD C. HOENICKE,** vice-president, Eaton-Erb Foundry Co., at Nov. 7 meeting of Metropolitan New York-New Jersey chapter of American Foundrymen's Association. . . . **T. A. BOYD,** head, fuel department, Research Laboratories Division, General Motors Corp., at the Detroit Forum of the American Engineering Foundation, Nov. 11. . . . **COL. L. K. SILLCOX,** first vice-president, New York Air Brake Co., at the Lehigh University Conference on the Social Significance of Engineering, Oct. 28. . . . **LEO HUFF,** manager of the Pure Oil Co.'s motor transport department and chairman of the American Petroleum Institute's central committee on automotive transportation, directed discussion on that topic at the API annual meeting in Chicago last month. . . . **ARTHUR W. STEVENS,** of Boston, recently participated in the program of the Massachusetts Safety Council.

GEORGE D. LAUGHTER has been advanced from sales engineer to eastern district sales manager of the Cimatool Co., Dayton, Ohio. His headquarters are in Newark, N. J.

SEIJI KONISHI has been named Omori factory manager of Tokyo Jidosha Kogyo Co. He was assistant manager of the company's Kawasaki factory. . . . **KANENORI NOMAGUCHI** has joined that organization as engineer of the standardization department. Previously he was designer with Tokyo Motor Vehicle Industry Co., Inc.

HOWARD EDWARD SOMES, who has been consulting engineer with the Budd Wheel Co., Detroit, has been made chief engineer of Budd Induction Heating, Inc., of the same city.

The Ethyl Gasoline Corp. has transferred **HUDSON W. KELLOGG,** from Detroit, where he was manager of the testing laboratory, to Yonkers, N. Y., as manager of the gasoline testing laboratory.

HOWARD D. MATTHEWS has joined the Minneapolis-Honeywell Regulator Co., Minneapolis, as consulting engineer on thermostatic bimetal. Before this change he was directing engineer of the General Plate Co., Attleboro, Mass.

ROY SMITH, a graduate of New York University, is junior draftsman with George Wittbold, Inc., New York.

AUGUST KELLER is vice-president and secretary of the Motor State Products Co., Ypsilanti, Mich.

LAURENCE H. POMEROY has established an office as consulting engineer in Middlesex, England. He formerly was general manager, engineering division, De Havilland Aircraft Co., Ltd., of the same city.

Visitors from Abroad

Foreign Members of the Society who visited SAE Headquarters while in New York to attend the National Automobile Show, the SAE Annual Dinner and the SAE National Transportation Meeting, include: **DIPL. ING. ZSIGMOND HOLLOS,** designer of military equipment, from Paris; **ING. GUIDO SORIA,** Fiat Co., from Turin, Italy; and **FREDERICK J. BUSWELL,** The Bristol Tramways & Carriage Co., Ltd., from Bristol, England.

FRANK G. GARDNER, formerly engineering department, Scintilla Division of Bendix Corp., is aviation representative for Kendall Refining Co., Bradford, Pa.

FRED J. FISHER, formerly with Scott Bros., Inc., has joined the General Truck Sales & Service Corp., Philadelphia.

LESLIE F. LITTLE has been named works superintendent of Vickers-Armstrongs, Ltd., Chertsey, England. He previously was technical manager.

The Border Cities Aero Club, Roseland, Ont., has appointed **ROBERT D. BYERS** its manager. He also instructs in flying.

CHARLES J. LEASENFELD has been advanced from student engineer to experimental engineer by the Mack Mfg. Co., Allentown, Pa.

JAMES N. MEZEY is service salesman, motor clinic division of Gulf Oil Co., New York. He was automotive engineer with Tide Water Associated Oil Co.

M. E. CHANDLER is vice-president in charge of engineering of the recently formed Chandler-Evans Corp. of Detroit, manufacturers of aircraft carburetors, fuel pumps, and aircraft accessories.

VINCENT BENDIX has been named president of Bendix Airport, Inc., recently formed to develop Bendix Field, Bendix, N. J., to make it a center for private flying and research in the east.

CHESTER L. WENTZ has been advanced from sales engineer to Detroit division manager by Penola, Inc., a subsidiary of Standard Oil Co. of N. J.

HARRY A. TOULMIN, JR., has been elected a director of the Hydraulic Press Mfg. Co.

MAJOR EDWIN E. ALDRIN, aeronautical consultant, is vice-chairman of a recently organized committee known as Air Youth of America, which has been established to study development of the aeronautical education program for youth. Other SAE members on the committee are **MAJOR LESTER D. GARDNER** and **GROVER LOENING.**

The United States Steel Corp. of Del. has announced the appointment of **GEORGE WHEELER WOLF** as president of its export subsidiary, the United States Steel Products Co., effective Jan. 1, 1939. Mr. Wolf, at present in charge of operations of the General Motors Overseas Operations, has been with that company for 13 years, ten of which were spent abroad. He was successively managing director of General Motors Argentina; General Motors Poland, General Motors B.m.b.H., Germany; and of General Motors Peninsular at Barcelona, Spain. He returned to this country in January, 1936, as vice-president of the General Motors Export Division, in charge of production and engineering.



G. W. Wolf
From Motors to Steel

ARCH WARNER, until recently Michigan district manager of the Logan Gear Co., is now affiliated with Mechanics Universal Joint Division, Borg-Warner Corp. Mr. Warner will be working out of the Division's offices in Detroit. Prior to going with the Logan Gear Co. he was affiliated with the Universal Products Co. for approximately 10 years.

WILLIAM L. BATT, president of SKF Industries, Inc., was guest speaker at the Annual Luncheon Meeting of the American Standards Association, held in New York, Nov. 30.

CHESTER A. CRAIG, formerly vice-president, Ryan Sales Engineering Co., Detroit, has joined the sales staff of the Kawneer Co., Niles, Mich.

CHARLES WAYNE MAYNARD is 2nd lieutenant, A.C.R., U. S. Army Air Corps, Randolph Field, Texas.

Several SAE members have been named officers of the Detroit Industrial Safety Council. . . . **WILLIAM S. KNUDSEN,** president of General Motors Corp., has been re-elected president. . . . **HUGH DEAN,** general manufacturing manager of Chevrolet, was named vice-chairman of the Council's board of directors. . . . Vice-presidents of the board of trustees include: **WALTER O. BRIGGS,** president of Briggs Mfg. Co.; **K. T. KELLER,** president of Chrysler Corp.; and **ALVAN MACAULEY,** president of Packard Motor Car Co.

W. S. JOHNSTON is president and general manager of Johnston Cadillac, Inc., Trenton, N. J.

FOSTER N. PERRY has been named sales manager of the eastern division of American Bosch Corp. (formerly the United American Bosch Corp.). He was manager of the New York branch.

E. R. VIBERG, formerly chief engineer and works manager, Canadian Car & Foundry Co., Ltd., Montreal, has been advanced to assistant vice-president.

The American Stock Gear Co., Chicago, has added **N. A. KLIPPER** to its sales staff. He was engineer with the Metal Specialties Mfg. Co.

ROBERT E. BUSEY, formerly project engineer, Research Laboratories Division of GMC, has joined the experimental department of White Motor Co., Cleveland.

DAVID C. SPAULDING, JR., has been named plant engineer of the Standard Oil Co. (Ohio); promoted from the engineering inspection department.

H. L. KOHLER has left the Sealed Power Corp., Muskegon, Mich., where he was research engineer, to become assistant professor of mechanical engineering at the University of Michigan.

The Modesto (Calif.) Junior College, technical institute, has added **CHESTER F. CARTER** to its staff as instructor and technician. He was field representative for the John Bean Mfg. Co.

RALPH LADD is in training for sales work with the Ahlberg Bearing Co., St. Louis, Mo. He graduated from Tri-State College of Engineering with the class of '37.

E. T. FOOTE, vice-president, Globe-Union, Inc., has been elected second vice-president of the National Battery Manufacturers Association. SAE members named to the board of directors are: **F. C. KROEGER,** general manager, Delco-Remy Division of GMC; **J. H. McDUFFEE,** vice-president, Electric Auto-Lite Co.; **B. F. MORRIS,** vice-president, manager of the Emark Battery Division, Thomas A. Edison, Inc.; and **S. W. ROLPH,** vice-president and general manager, Willard Storage Battery Co.

H. W. McQUAID, Republic Steel Corp., received the Sauveur Achievement Award of the American Society of Metals at the National



H. W. McQuaid
Honored

Metal Congress. . . . **DR. GUSTAV EGLOFF,** director of research, Universal Oil Products Co., has been nominated for the presidency of the American Chemical Society. . . . **PAUL G. HOFFMAN,** president of Studebaker, recently was elected president of the newly formed Indiana Traffic Safety Council. . . Last month Mr. Hoffman served on a committee cooperating with the Purdue Research Foundation in planning the annual industrial research conference at Purdue University. . . . **W. H. FUNSTON,** president of the Firestone Tire & Rubber Co. of Canada, Ltd., has been named to the board of directors of the Canadian Chamber of Commerce. Mr. Funston also is president of the Hamilton Chamber of Commerce. . . . **ROBERT T. HENDRICKSON,** president of the Hendrickson Motor Truck Co., is chairman of the commercial vehicle section of the Chicago Automotive Trades Association. . . . **JAMES I. SIMPSON,** vice-president and general manager, Dunlop Tire & Rubber Goods Co., Ltd., has been elected president of the Rubber Association of Canada. . . . **L. C. WELCH,** assistant general manager, Standard Oil Co. of Indiana, is vice-chairman of the lubricating oil committee of the American Petroleum Institute's division of marketing. . . . **LEE M. CLEGG,** executive vice-president, Thompson Products, Inc., and **DR. ZAY JEFFRIES** have been elected to the corporation of the Case School of Applied Sciences. Mr. Clegg will also serve on the board of trustees. . . . **JOHN A. HASSEY,** operating superintendent, division A, Cities Service Oil Co., has been elected chairman of the private carrier division of the American Trucking Associations, Inc.

SAE Graduates

W. A. HASBANY, who was vice-chairman of the SAE Student Branch at General Motors Institute, has been made assistant contact engineer, Olds Motor Works. . . . **DAVID W. GRIMM,** Polytechnic College of Engineering, is junior detailer, Vega Airplane Co., Burbank, Calif. . . . **HAROLD E. SCHLENER,** Pratt Institute, is inspector of equipment, Emby Photo & Film Machine Corp., New York. He also is taking night courses at Brooklyn Polytechnic Institute. . . . **CARL D. LYNN,** Ohio State University, is employed in the Atlas Works of the Socony-Vacuum Oil Co., Buffalo, N. Y. . . . **J. M. MACKENZIE,** University of Michigan, is a teaching fellow at the University of Iowa. . . . **JOHN G. McQUAID,** another Michigan graduate, is with the Chrysler Corp. as student engineer. He is also attending the Chrysler Institute of Technology.

Obituaries

John E. Ruth

John E. Ruth, Baltimore district sales manager for the American Hammered Piston Ring Division of the Koppers Co., died on Oct. 28. He had been an associate member of the Society since 1918.

His connection with the automotive industry started in 1908 when he did carburetor and ignition work in Baltimore. Later he had part in the technical construction work at the Ford branch in that city, involving plant layout, reinforced concrete work and the installation of machinery. He was in charge of technical service and commercial sales for the Winton Branch in Baltimore during 1917 and 1918. The next year he joined the Bartlett Hayward Co., and was made an engineer of its American Hammered Piston Ring Division in 1922. Leaving American Hammered in 1927, he did sales and advertising work with the Warwood Tool Co., the Ramsey Accessories Mfg. Co., and the Eastern Valve Co., before returning to the American Hammered Piston Ring Co. in 1934.

Mr. Ruth was born in Baltimore in 1888.

S. P. Worthington

S. P. Worthington, president of the Tiona Petroleum Co., died on Oct. 26. He was 46 years of age.

Since 1907, when he left high school to join his father in the Terabentine Co., sellers of petroleum solvents, he had been associated with the petroleum industry.

He learned petroleum chemistry through the "school of hard knocks" while working in the laboratory of the Union Petroleum Co., which he joined after one year in his father's company. He was advanced to chief chemist of Union, and later did special work in connection with the sales department and for some years purchased petroleum products for the company. When the Sinclair Refining Co. purchased the Union Petroleum Co. shortly after the World War, Mr. Worthington rejoined the Terabentine Co. as its president. In 1923 he formed the Tiona Petroleum Co., serving as its secretary until 1929 when he was made president. He continued his association with the Terabentine Co.

While purchasing petroleum products for the Union Petroleum Co., Mr. Worthington organized the first association of oil testers in the Pennsylvania fields and served as its president for one year. This association continued until A.S.T.M. methods were adopted.

Mr. Worthington became a member of the SAE in 1921.

K. J. Howell

K. J. Howell, a member of the Society since 1921, died on Sept. 24. He was a graduate of the Sheffield Technical School at Yale University and had been employed in engineering work by Mercer Motor Co., Rolls-Royce of America, Inc., the Studebaker Corp., and the General Motors Export Co. He was active in the American Automobile Association as Contest Board representative. Mr. Howell was born in Chicago in 1900.

V. G. Barford

V. G. Barford, chief designer, motor works, J. I. Thornycroft & Co., Ltd., Basingstoke, England, died on Sept. 6. He had been affiliated with that company since 1902, advancing from draftsman to chief designer. Earlier he did drafting work for companies manufacturing oil, gas and steam engines.

Mr. Barford was active in standardization work and had represented the Society of Motor Manufacturers and Traders and his company on committees of the British Standards Association.

He was born in 1872 at Northampton, England, and attended the Grantham Science Classes. Mr. Barford became a foreign member of the SAE in 1928.

West Coast T&M Meeting

(*Continued from page* 12)

to insure adjustment and correction of the first echelon work.

The third echelon comprises unit replacement, such as a rear end, transmission, or in certain cases an entire engine, with a similar serviceable unit.

The fourth echelon comprises unit repair and requires expert mechanics working in a shop equipped to rebuild units that have been replaced on the vehicle and turned in for general overhauling.

The fifth echelon contemplates reconstruction. This embraces, in addition to repair and manufacture, the reclamation and salvage of motor vehicles, unit assemblies and sub-assemblies in order to replenish stock from non-economic repairable units.

Colonel McGuire stated that all echelons of motor maintenance merge one into the other, particularly the first four which are often performed in a shop equipped for heavy maintenance. The echelons apply to track-laying vehicles as well as wheeled vehicles and are adopted in the motor maintenance of transportation units assigned to the Civilian Conservation Corps. The speaker also described the work of the mobile repair shops which had been demonstrated in the afternoon.

W. W. Churchill, superintendent of equipment, Washington Motor Coach Co., was prevented from presiding as chairman of the Friday morning session on bus operation by unexpected business matters. His place was taken by Mr. Drake who introduced the three speakers.

Value of Cost Control System

Reporting on the cost control system utilized by the Los Angeles Railway Co., of which he is superintendent of automotive equipment, C. B. Lindsey explained that from their records it is possible to not only check the cost per mile of any of the 24 types of coaches operated, but also to determine the cost per mile of the seven major units which go to make up a motorcoach; including such units as the body, brakes and engine. This, he said, indicates to the mechanical department where an increased cost per mile is coming from, making it possible to check the trouble in a very short time in most instances. A quarterly cost-comparison report, he added, is carefully checked for anything unusual.

The title of Mr. Lindsey's paper was "Cost Control in City Bus Operation," and it included, besides notes on records systems, reference to fuel costs, fuel combustion, tests of appliances, experiments with butane, and comments on gassing tests.

S. B. Shaw, automotive engineer, Pacific Gas & Electric Co., was the second speaker of the session, presenting a paper on "Overhead."

In transportation cost accounting systems, the speaker said, overhead generally includes the following items of cost: Investment charges (including interest and depreciation) on lands, structures, garage and shop equipment, and motor vehicles; taxes and insurance on these items; their upkeep; and supervision expense, including clerical, stores, and tool expenses.

Mr. Shaw analyzed each of these overhead items and showed comparisons of per-mile overhead charges based upon the number of miles operated per year, the number of trucks in the fleet, and the original costs of the trucks and other equipment.

The final paper of the session was presented by T. C. Howe, superintendent of equipment, North Coast Transportation Co. and entitled "Bus Maintenance Problems and Practices." Mr. Howe reviewed the methods used by his company to meet the service and maintenance problems presented by the various parts of the modern bus.

Participating in the discussion were Roland Moore, Mr. Meiser, E. C. Wood, and Carl Abell, who discussed the merits of automatic spark control, as well as the accelerometer, the exhaust-gas analyzer and other instruments.

At the luncheon which followed, C. J. Vogt, associate professor of mechanical engineering, University of California, presented a paper by himself and F. A. Ryder, of the University, on "Automotive Research at the University of California." He was introduced by L. M. K. Boelter, professor of mechanical engineering at the University.

The paper briefly described research activities of students, graduate students and members of the faculty on such subjects as fuel, spark plugs, crankcase oil, diesel injection systems, compressibility of oils, cetane numbers, motor-vehicle highway illumination, clearance and marker lamps, transmission of light in fog, and equipment for testing motor-vehicle operators.

Two papers were read at the afternoon session on Mixed Fleet Operation, which was presided over by William H. Brown, truck master, Southern California Edison Co.

"Depreciation," a paper prepared by T. H. Mullen, engineer of transportation, and R. G. Booth, assistant engineer of transportation, general plant division, Los Angeles Department of Water & Power, was read by Mr. Booth. The second paper was presented by A. T. Colwell, vice-president in charge of engineering, Thompson Products, Inc., and had to do with "What the Parts Manufacturer Can Do to Reduce the Cost of Operation."

Declaring that types of equipment, working conditions, topography, and climate have as much effect upon the life expectancy of a unit as they do on the cost of its mechanical repairs, Mr. Booth explained that the operator's aim is to establish some method of figuring depreciation which could apply to a fleet as a whole and arrive reasonably near the average.

Some of the methods in use for establishing a reserve fund for the replacement of existing equipment in a fleet were listed by the authors as based upon hours of actual service, mileage, fixed percentage yearly, varying percentage yearly, and a fixed percentage of declining balance. The first two of these methods, he noted, are based entirely on the actual usage of the equipment with the obsolescence feature ignored, while the latter three presuppose a decreased market value regardless of the amount of usage.

Candid Shots at West Coast Meeting



1. T. C. Howe, speaker at the Bus Operators' Session
2. Maj. E. C. Wood, banquet chairman, and S. B. Shaw, who spoke on "Overhead"
3. A. T. Colwell speaking at the Mixed Fleet Operators' Session
4. R. G. Booth presenting a paper on "Depreciation"
5. Robert E. Rowley, *standing*, introduces Lt. Col. E. C. McGuire, *seated*, speaker at the Army Session
6. Stanley Whitworth presents Watt L. Moreland with a plaque commemorating his 30 years of SAE membership
7. Dinner Speaker Col. R. T. Eddy talks on "Regulation of Highway Carriers"

Photographs by R. DeWitt Miller

No matter what method is used for determining it, he stated, depreciation is an outstanding item of cost. In a business offering trucks for hire, he continued, a too liberal rate for depreciation may eventually mean the difference between red ink and black on the balance sheet.

Factors affecting depreciation and obsolescence were discussed as well as the effect of continuous maintenance on depreciation and the determination of the economic period for the retirement of units.

Mr. Booth strongly emphasized that depreciation of automotive equipment is a problem which should come under the jurisdiction of the fleet superintendent and not the accounting department. "Surely," he said, "the man who is vested with the authority of determining the point when a unit should be retired is capable of estimating the procedure for evaluating that unit during its life."

Robert N. Reinhard, Golden State Dairies, told of the method of determining depreciation used by his company and of other methods used in his field. He pointed out that in the milk industry delivery trucks are subjected to severe usage due to numerous stops and starts in daily operation, and that because of this the average life before rehabilitation will not exceed four or five years. Others entering the discussion of the Mullen-Booth paper were S. B. Shaw, F. C. Patton, and J. F. Dixon.

"The alert parts manufacturer makes it his business to know the types of truck operations which cause trouble and the parts involved, and quickly has a superior part designed and available," Mr. Colwell said in his paper. "The policy of using these superior parts in all fleet equipment is gaining momentum because the longer service and freedom from overhaul and tie-up actually reduce the cost per mile of operation, even though the parts are more expensive," he added.

Other things that parts manufacturers can do to help operators cut costs, mentioned by Mr. Colwell, are: to make parts more available, to have field engineers to cooperate with operators in reducing maintenance costs and correcting troubles; to assist the fleet operator in planning and establishing a system of records; to have a more complete catalog with descriptive information; to furnish adequate instructions with parts; to supply a technical manual covering his parts; and to standardize parts so that they would be interchangeable.

In concluding, Mr. Colwell stated, "There is no part made today that cannot be made worse and sold cheaper. But the real reduction in operating cost is being made with more expensive parts that give longer service."

Presiding at the Banquet Friday evening was Major E. C. Wood, Pacific Gas & Electric Co., who introduced the representative group at the speakers' table. After a short talk by Frank Bull, Associated Oil Co., who announces important football games, the chairman called upon Pete Dunker who introduced the speaker of the evening, Col. R. T. Eddy, regional director, Interstate Commerce Commission.

Colonel Eddy pointed out that a peculiar condition exists in California because comparatively few carriers cross the state boundary, and questions have arisen regarding the I.C.C.'s jurisdiction. These are now before the committee. It is the speaker's belief that practically all trucks in the state are engaged in interstate commerce. For example, he said, although a carrier comes only from the harbor to Los Angeles, it is carrying interstate commerce in picking up and transporting goods from ships.

Saturday morning saw the convening engineers en route on the various inspection trips arranged by Southern California men. Carl Abell led a group to the Ethyl Tune-Up Clinic; F. C. Patton was guide at the Los Angeles Railway bus shops; and Frank Pim was in charge at the National Automotive and Diesel School.

National Transportation Meeting

(*Continued from page* 10)

in a comparison ranging from 5.50-20 to 13.50-20 in size.

These intervals are too great, Mr. Horine believes, at least above the 7.50 size. For the cause for this condition he goes back to the establishment of the present system of tire sizing.

"Instead of starting with a series of capacities that would be most useful from a standpoint of the weights to be sustained, the establishers of the present standards started with an arbitrary series of sizes and then determined by experiment what loads they would carry economically," he contended. "The tire industry had a golden opportunity some 15 years ago when the low-pressure series was established," he pointed out. "Then was the time when a study of load, body, and chassis weights, of legislative axle-weight restrictions and registration fees could have established a scale of practicable gross vehicle weights for which tire sizes could have been devised. Instead, the old arbitrary sizing was retained, employing decimals to designate the cross-sections and substituting rim diameter for peripheral diameter."

To remedy the situation, Mr. Horine suggests five additional sizes within the range discussed. If 22-in. rims were abandoned, he explains, eight present sizes would be omitted, seven new sizes avoided, and the total result would be five fewer sizes than at present. In any event, he concluded, the elimination of high-pressure sizes before long will decrease the total number of sizes to fourteen. Mr. Horine further recommended that tires be designated by rim size and capacity, for example 22-5000, instead of the present 10.50-22.

Establishment of tire sizes in 1925 was not done on an arbitrary basis, as stated by Mr. Horine, contended Dr. B. J. Lemon, U. S. Rubber Products, Inc. To understand the basis for their choice it is necessary to know the problems that had to be solved at that time, he explained. Trucks then were rated only on tons; there were no maximum ratings and very little legislation. Then, he continued, we had to try to establish a series of balloon tires to fit into the high-pressure line and rims; we had to consider also inflations and dual spacings. The rubber industry at the present time is trying to simplify sizes; to throw out high-pressure tires; and to get motor manufacturers to do it.

Our difficulty is not so much in finding the right size of tire, as in finding vehicles with the right size on them, contributed Mr. Faulkner.

Agreeing with Mr. Horine that the range of tire sizes is erratic and that some benefit could come from a change, James E. Hale, of Firestone Tire & Rubber Co., expressed doubt as to whether the advantages from a technical and economic standpoint would be enough to justify the "terrifically painful job of putting it through." He also mentioned the desirability of policing the automotive industry so that nobody demands a size that he shouldn't have, and of legislation that does not permit overloading of tires, such as has been passed in a foreign country.

Although Mr. Horine has opened up a subject where improvements can be made, complete revision of tire sizes will not be necessary, believes Adolf Gelpke of Autocar Co. We all know that tires are overloaded, he pointed out, and a certain amount of overload is approved openly by the industry. Referring to Mr. Horine's suggestion to eliminate some large sizes because of limitations of excessive brake-drum heat, Mr. Gelpke called attention to the fact that many of the vehicles using these large sizes are so slow-moving that brake-drum heat does not enter into the picture. Amplifying Mr. Gelpke's point, K. D. Smith, B. F. Goodrich Co., contended that Mr. Horine more or less overlooked the fact that load-carrying capacity is governed specifically by speed. Tables that give tire capacities based on speed soon will be available, he predicted.

The tire capacities given are all maximum values, pointed out P. A. Kerns of Goodyear Tire & Rubber Co. If the operator has excess capacity, he can drop his inflation pressure, he suggested. If the truck and bus companies keep on demanding wrong or obsolete sizes, we must keep on making them, he explained.

A plea to truck and bus manufacturers to consult the tire manufacturers before specifying sizes was voiced by B. W. Elgin of Firestone Tire & Rubber Co. He expressed the willingness of his company to cooperate in any revision of sizes that promises a return of something better or more useful.

No tire company encourages overload or underinflation, was Mr. Hale's reply to the statements of several discussers to the contrary. There has to be a commercial basis for a tire company to consider complaints, he explained, and that basis is the load and inflation rating.

In summary Mr. Horine conceded that Dr. Lemon and Mr. Hale had given cogent reasons why present tire sizes are as they are. One question still unanswered, he said, is why the size intervals are made dimensional instead of by increments of capacity. Replying to Mr. Hale's question on how the industry could be policed, Mr. Horine recommended the Tire & Rim Association as well qualified for the job. Referring to Mr. Kerns' recommendation to drop inflation pressure if there is excess tire capacity, Mr. Horine challenged his audience to name one truck operator who has excess tire capacity.

Improvements Asked in Electric Drives

During the past 12 years approximately 3000 electric-drive buses have been placed in service in this country, reported G. W. Wilson of General Electric Co. His paper, "Diesel-Electric Bus Drive," was the only one read at the second session of which C. O. Guernsey, J. G. Brill Co., served as chairman.

Of these buses, Mr. Wilson said, about 250 are used with automotive diesel engines and were placed in service during the past 18 months. Although conceding that the latest electric drive is still somewhat more expensive and heavier than gear-box drives, he pointed out that, by constant improvements, the electric drive has been made 35% lighter, 25% less expensive, and 3 to 8% more efficient than its predecessor.

"Experience has proved the necessity of having some form of flexible drive between the diesel engine and the driving wheels of a bus to reduce vibration and noise in the passenger compartment and to damp the torque pulsations to the driving mechanism," Mr. Wilson contended.

An ideal motor-bus powerplant combination, he explained, would consist of a diesel engine with a fully automatic, infinitely variable ratio transmission to permit complete mechanical isolation of the power unit from the passenger compartment. Electric drive has all of these desirable characteristics and, for many years, has been used on motor buses with gasoline engines, he contended.

Problem No. 1 in the design, application, and operation of these diesel-electrics has been the control of excessive heat, Mr. Wilson declared. Mr. Wilson's paper was illustrated by slides showing details, typical applications, wiring diagrams, and performance characteristics of the equipment.

O. D. Treiber of Hercules Motors Corp.,

brought up the problem of getting the torque curves of the diesel engine and the electrical apparatus to coincide in diesel-electric drives for best efficiency. He commended the electrical manufacturers for their cooperation in the progress already made, and showed by blackboard sketch the desirability of having the diesel torque curve continue upward instead of falling off at very low speeds.

After announcing that he was an "ardent exponent of the electrical system to give infinitely variable transmission ratios," Col. G. A. Green of Yellow Truck and Coach challenged several of Mr. Wilson's statements.

Mentioning first "the necessity of having some form of flexible drive between the diesel engine and driving wheels of a bus to reduce vibration and noise . . . . and to damp torque pulsations," which Mr. Wilson had said that experience has proved, Col. Green commented that "experience has proved no such thing," calling attention to the large number of diesel applications equipped with normal gear boxes.

Referring to Mr. Wilson's remarks to the effect that the low-speed torque of the diesels used in their diesel-electric drives was so low that they probably could not be run on a conventional gear train at all, Col. Green replied: "In so far as General Motors 2-cycle diesels are concerned, Mr. Wilson has been sadly misinformed," explaining that the torque of these diesels was equal to comparative gasoline engines.

The hydraulic transmission that is now being marketed by his company, Col. Green contended, has exactly the same characteristics as the electric transmission. In comparative proving-ground tests we found that the gasoline-electric drive took 20% more fuel than the hydraulic-gasoline drive, he announced: Col. Green concluded with a plea to the electric-drive manufacturers to concentrate on making the units cheaper, lighter, and more efficient so that they could be applied more widely.

Replying immediately to Col. Green, Mr. Wilson stuck to his guns with respect to the necessity for some flexible drive between the diesel engine and driving wheels of a bus. "I didn't know when I wrote the paper, and I still don't know of a diesel engine operating in a bus without such a flexible drive," he maintained. On the subject of fuel consumption, he recalled examples where the fuel consumption in street service had been no more than with mechanical-drive vehicles. He concluded with a confident prediction that both weight and cost of the electric drive equipment would be reduced in the near future.

One of the important benefits of the hydraulic transmissions may be to stimulate the electric manufacturers in their development of electric drives, contributed Merrill C. Horine, Mack Mfg. Corp.

"The diesel-electric drive is a great deterrer of abuse," contended Martin Schreiber of Public Service Coordinated Transport, explaining how difficult it was to abuse one even with four careless drivers a day. By using them we get rid of many annoyances, such as carburetors, spark plugs, and ignition systems, he continued. Another point in favor of the diesel-electric drive, he added, is that there is no depreciation in the efficiency of the drive after 8 to 12 years of service.

To conclude the session Chairman Guernsey summarized the situation as he saw it. Electric or hydraulic drives, he pointed out, have definite advantages in buses used in cities, especially where stops are frequent, as they eliminate frequent shifting and declutching. Also, he continued, the motor can be tuned more sharply than when a wide speed range is necessary. Cost, weight, and efficiency all affect the choice of drive selected, he concluded.

More Instruments in Maintenance Work?

The manufacturing branch of the automobile and truck industries puts to work every known measuring device and is constantly devising new means for determining, definitely and accurately, various factors affecting engine manufacture and design, asserted W. A. Roberts of the Rappuhn Corp., at the Tuesday afternoon session devoted to maintenance. Mr. Roberts was introduced by T. L. Preble of Tidewater Associated Oil Co., who served as chairman. Co-author of the paper with Mr. Roberts is W. H. Yenni of Joseph Weidenhoff, Inc. Its title was, "Test Procedure and Instrumentation for Maintenance of Gasoline Engines."

The result of this precision work, Mr. Roberts continued, is a wonderfully rugged and efficient engine that eventually finds its way into the hands of a mechanic who must be trained to use instruments intelligently.

By means of instrumentation, he explained, we are able to measure the performance of the various parts of the engine and, by comparing these measurements with known standards, we can tell if they are capable of continuing to operate normally.

A detailed amplification of the program made up the remainder of Mr. Roberts' talk. To explain graphically the methods used to teach mechanics how to use the various instruments, he displayed several large colored charts, one of which depicted the basic wiring diagram for any car.

"It is about time that we, as the users of automotive equipment, should have learned how to operate these vehicles," challenged Capt. O. A. Axelson of Columbia Gas and Electric Corp., in written discussion. The reason that we do not know how, he explained, is that we don't know what efficiency of operation really means – we have no yardstick except our own judgment (which unquestionably is usually biased) to go by.

"The determination of *when* a vehicle is operating at maximum efficiency and the determination of *when* and *what* repairs should be made are engineering problems, and standards for these determinations must be set up by engineers," Capt. Axelson contended; "our mechanics, good as they are, never will do it, and we can't expect them to do it."

Whether or not the vacuum gage is a dependable instrument for determination of valve, compression, and timing conditions looks to Capt. Axelson like "practically yes, but theoretically no," as the mechanics tell him, "yes" and the engineers, "no." An accurate answer should be obtained by scientific test, Capt. Axelson believes.

The problem of instrumentation in a scattered fleet was introduced by R. M. Cregar of Public Service Electric & Gas Co. With small fleets of around 30 vehicles at widely scattered points, he pointed out, we cannot afford to set up a complete laboratory. In reply Mr. Roberts indicated that four instruments costing $200 would be enough for an initial check and suggested that more detailed work could be done by outside garages.

Evidence of the efficacy of coil-testing equipment was presented by R. M. Critchfield of Delco-Remy. Before it came into use, he reported, 50% of the coils returned to us were O.K. Since this equipment has been available, however, the O.K. percentage has dropped to 15.

The instrument manufacturers have not done a good job in telling us how to use their equipment, opined J. Willard Lord of Atlantic Refining Co. In answer to Mr. Cregar's concern about the cost of the instruments, Mr. Lord reminded him that a few miles per gallon per year increase will buy a lot of instruments in an operation of any size.

Supplementing Capt. Axelson's remarks on the use of the vacuum gage or tachometer for timing, W. C. Condit of Sun Oil Co. reported that he has not seen it work consistently and cannot see any reason why it should work – how it can be sensitive enough.

Although admitting his inability to explain why, Mr. Roberts stated that he knew mechanics who had used them successfully for years. It's the best method we've had so far, he concluded, but we'll welcome a better one.

"How far dare we go in costs in installing such a system?" inquired J. G. Moxey, Sun Oil Co., explaining that it is only one cog in an elaborate preventive maintenance system. "Shall we adjust the fleet for the system, or the system for the fleet?" A rule to follow is, do not apply it unless it fits into the individual operation of an individual system, he suggested.

The possibility of over-maintenance, when systems cost more than they save, was pointed out by T. C. Smith of American Telephone & Telegraph Co. When we get to know a certain make of vehicle, we frequently can tell what its troubles are without elaborate testing, he maintained.

last call

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Classifications

1. **SAE Activities**
2. **Industrial**
3. **Home**
4. **Pictorial**
5. **Summer Meeting**

Ingenious Devices Described

Although there is no evidence at hand that fleet operators as a class wish to desert their amateur standing as designers, Henry Jennings of *Commercial Car Journal,* reported that many fleet operators have been able to improve their lot by means of a little "rough-and-tumble design" when their shops have unusual maintenance problems, strange shop layouts and conditions, and their vehicles operate under peculiar conditions.

Before reading his paper, "New Kinks in Live Truck Operations," at the session chairmanned by S. G. Tilden, Mr. Jennings explained that he really was not the author of the paper, but merely the reporter for the thirty odd operators who had contributed the material. The ingenious devices described, he continued, were compiled from a large number of shops scattered over a wide area.

Accessory designers seem to be limited by commercial possibilities, he pointed out, depending upon volume. When volume is too low, the fleet operator can substitute or adapt and frequently it takes some thinking to do it, he said.

All in all, the fleet operator would rather buy than make, he continued, but the kinks that I will describe were created when the fleet operator did not buy, but solved the problem in his shop.

H. O. Mathews of Public Utility Engineering & Service Corp., expressed the belief that the suggestions will prove valuable to many operators. He then described a device built in his shop that solved a hoist problem.

Capt. O. A. Axelson of Columbia Gas & Electric Corp., urged that the material be published so that many shop men will have the time and opportunity to study it, and so that the originators get credit for their ingenuity. One of these ideas, he pointed out, may help an operator to solve a similar problem. Wider publicity for the paper also was recommended by E. F. Theisinger of *Bus Transportation,* who suggested that the paper be read at other association meetings where more operating men and mechanics could hear it.

Such home-made gadgets must be checked carefully by engineers to be sure that they are safe, cautioned F. K. Glynn, American Telephone & Telegraph Co., naming several of the devices described as of questionable safety.

Spring Developments Debated

The problem of obtaining the same frequency under all loads is more difficult with trucks and buses than with passenger cars, pointed out N. E. Hendrickson of The Mather Spring Co., in his paper, "Trends in Commercial Vehicle Spring Suspension," at the final session of the meeting at which H. E. Simi, Twin Coach Co., served as chairman.

The reason for this condition, he explained, is that the load variation is much greater in commercial vehicles, and the difficulty of obtaining satisfactory frequencies at light loads is ever present. He also called attention to the need of progressive stiffening to reduce sidesway, a serious problem because of the narrow spring treads made necessary by large tires.

The long bow, that ancient weapon that made England's archers famous, has a section with a flat tension side and modified parabolas on the compression side, Mr. Hendrickson reported. This section is similar to the spring section called, "parabolic-edged spring steel," and saves up to 10% of the spring weight because this shape relieves or eliminates the concentrated corner stresses on the tension side, he contended.

Among the other spring developments discussed by Mr. Hendrickson are the "progressive" and "two-stage" methods of obtaining a variable rate, the "torsion-rod" suspension, and "Magnaflect" spring suspension.

Reviewing spring development from 1900, Roy W. Brown, Firestone Tire & Rubber Co., saw a definite trend toward independent positioning of wheels by members other than springs. "Can we or can we not isolate the spring from these other factors?" he asked. We must determine just what spring properties we are interested in, he pointed out. Dynamic measurements, he said, bring out the absolute necessity of associating shackles, friction, shock absorbing, and damping. He stressed the desirability of obtaining a variable rate in recoil the same as a variable rate in compression.

A. F. Hickman of Truck Equipment Co., gave supplementary information on his company's "torsion-rod" suspension. With the Hotchkiss drive principle, he stated, Mr. Hendrickson says that the engineer receives "God's gift of cushioning" or, in other words, axle roll or rotation which he believes desirable. We find that, by allowing axle wind-up or rotation, there is a time period produced which, upon brake application, allows the brakes to seize onto the drums momentarily and then, of course, the high friction of rest must be broken. This grabbing and releasing lends very well to brake chatter and excessive bearing wear, Mr. Hickman contended.

R. E. Berg of Johns-Manville Corp., stressed the point that a leaf spring performs many other functions in a vehicle besides springing.

The second stage of the "two-stage" spring comes into action too suddenly and will never permit a gradual change from one rate to another, maintained F. B. Lautzenhiser of International Harvester Co., reading from prepared discussion. The progressive type, on the other hand, he continued, permits the gradual change in the rate of deflection and has better pos-



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sibilities but, in his opinion, it is not the final answer.

Speaking from 25 years former experience as one of them, Clinton Brettell, E. R. Merrill Spring Co., emphasized the importance of good springing to the fleet operator, especially where fragile material is transported.

In summary Mr. Hendrickson agreed with Mr. Brown that a variable rate on recoil would be valuable. He expressed disagreement, however, with Mr. Hickman's contention that the Hotchkiss drive causes brake chatter. The brake dive mentioned by Mr. Lautzenhiser has helped to bring about knee action, he pointed out in conclusion.

Students Hear Stout on Future of Engineering

● U. of Michigan

November was an active month for SAE Student Branch members at the University of Michigan. William B. Stout, president, Stout Engineering Laboratories, Inc., and a past-president of the SAE, was the guest speaker at the Nov. 1 meeting, which student members of the American Society of Mechanical Engineers and the Institute of the Aeronautical Sciences also attended. His subject was "Looking Ahead in Engineering."

On Nov. 14 about ten members went into Detroit for the Auto Show and to attend the Detroit Section meeting.

New officers of the Branch are: Fitz Bridges, chairman; Clare MacKichan, vice-chairman and chairman of program committee; David I. Smith, secretary-treasurer; James Mason, chairman of publicity committee; and Foster Gaylord, chairman of membership committee.

Ideas in Zinc

It is significant that 10 of the 15 makes of cars in the medium and high priced groups will have zinc alloy die cast radiator grilles in 1939. Automotive engineers have concentrated their efforts on radiator grilles and hood lines, for these are the features which immediately distinguish one car from another. And visitors at the National Automobile Show have given complete approval to the new front end designs.

In the trend toward the elimination of the familiar center radiator grille, front end lines have been expressed with tapered hoods flanked by side mounted zinc alloy die cast grilles faired gracefully into the front end sheet metal aprons.

But there are still a number of cars which have retained the grille in the center position. The 1939 Hudson Country Club "6" front end is typical of the treatment on these cars. Here we find that the main grille is supplemented by low mounted grilles on either side—all three are die cast of zinc alloy.

It has been definitely established that low mounted grilles provide greater efficiency in air cooling, with the result that they will be found on all '39 cars either as the main grilles, supplementary grilles, or both as in the case of Hudson.

Automobile designers have found that the die casting process provides the most flexible means of securing the unusual contours—the accuracy of dimensions—which are demanded in present day grille designs. This is due, in no small part, to the availability of the research developed Zamak alloys, based on Horse Head Special Zinc of 99.99+% purity. The New Jersey Zinc Company, 160 Front Street, New York City.

Idea No. 19

About Authors

(*Concluded from page 5*)

nance Department, United States Army from 1917 until 1921, and then taught mathematics for a year at Brown University before joining the National Bureau of Standards as associate physicist. For the past 12 years he has been chief of the Bureau's automotive powerplants section, which is charged with testing motor-vehicles and their accessories for Government agencies as well as the investigation of automotive engines and fuels. He prepared a report on the Detonation Rating of Aviation Fuels which was presented at the First World Petroleum Congress in London and, since 1933, he has been chairman of the groups responsible for planning and executing the C.F.R. program on the rating of aviation fuels in full-scale engines.

● **Hector Rabezzana** ***(M '19) has had a technical career which has taken him from Italy to Argentina to the United States. It began when the National Industrial Institute of Alessandro Rossi, at Vicenza, Italy, granted him the degree of Electro-Mechanical Engineer in 1900. It included such offices as technical director of Fiat, factory engineer of Isotta Fraschini, and technical manager of Isotta Fraschini in Buenos Aires, prior to his association, 23 years ago, with what is now the AC Spark Plug Division of General Motors. With AC he has long held the title of chief engineer, spark plug division. In 1909, while with Fiat, the Italian Ministry of Industry awarded him a medal for his cooperation in the development of new industries.***

● **F. Glen Shoemaker** ***(M '20), recently was nominated 1939 SAE vice-president representing the Diesel-Engine Activity. After graduating from the U. of Illinois in 1914 he worked in automobile and machine shops as mechanic and manager for a number of years. He then had three years as experimental engineer with the Buda Co., before spending six years at McCook Field, Dayton, in the airplane-engine laboratory of the United States Army Air Corps Engineering Division. This experience led to the Franklin Manufacturing Co. and the design of air-cooled automobile engines. For the past ten years he has been with General Motors. When the two-cycle Diesel engine program was started he was made project engineer and has followed this development through to his present position of chief engineer of the new Detroit Diesel Engine Division.***

Applications Received

(*Concluded from page 19*)

SCHROEDER, FREDERICK J., JR., sales, Milwaukee Die Casting Co., Milwaukee.

SPEXARTH, CHRISTY L., chief draftsman, Harley-Davidson Motor Co., Milwaukee.

Northwest Section

BARNHART, ROBERT M., chief engineer, Brown Industries, Spokane, Wash.

PATRONI, SALVATORE, U. S. Navy, Patrol Squadron 19, Seattle, Wash.

Northern California Section

CODY, E. H., automotive engineer, Standard Oil Co. of California, San Francisco.

RICH, ROBERT E., operations engineer, Associated Dredging Co., San Francisco.

VAZEILLE, CREED, mechanical salesman, Standard Oil Co. of California, Fresno, Calif.

Philadelphia Section

FAGER, CHARLES H., draftsman, Gannett Eastman & Fleming, Harrisburg, Pa.

JOHNSON, WALTER F., president, Johnson & Towers Inc., Philadelphia.

Pittsburgh Section

COLLIER, WALTER WAYNE, president, Walter W. Collier Co., Pittsburgh, Pa.

CRITTENDEN, A. M., chief chemist, The Freedom Oil Works Co., Freedom, Pa.

St. Louis Section

ROSS, HARRY HILAND, industrial representative, Sinclair Refining Co., St. Louis.

Southern California Section

CARR, GARDNER W., president, Menasco Mfg. Co., Los Angeles.

Southern New England Section

GREENBERG, HERBERT JAMES, draftsman, Hamilton Standard Propellers, E. Hartford, Conn.

Washington Section

WILBURN, HERBERT S., sales manager, Autocar Sales & Service Co., Richmond, Va.

Outside of Section Territory

GIMMEL, ROBERT S., sales manager, Atlas Auto Parts & Grinding Co., Louisville, Ky.

HUNT, FRED B., research engineer, Continental Oil Co., Ponca City, Okla.

NELSON, CLARENCE W., equipment and mechanical foreman, U. S. Bureau of Biological Survey, Middle River, Minn.

SCOTT, WILLIAM KEITH, president and general manager, Mill Gulch Placer Mining Co., Camp Raleigh, Nevada.

SZENASY, ANTON, shop superintendent, Transport Corp. of Virginia, Durham, N. C.

Foreign

BRADBURY, CYRIL HENRY, experimental engineer, Ruston & Hornsby Ltd., Lincoln, England.

HASHIGUCHI, YOSHIO, chief engineer, Kawanishi Aircraft Co., Naruo, near Kobe, Japan.

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Recent Developments in Spline and Gear Cutting and Finishing

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DEVELOPMENTS in spline and gear cutting discussed in this paper include actual production results in climb-hobbing of splines and experimental results in climb-hobbing of gears. "Climb-hobbing" is defined as that method of hobbing wherein the cutting action starts at the surface of the part being hobbed and ends at the root of the spline or tooth – the direct opposite of the conventional method. Important advantages claimed for the method are a superior finish, increase in hob life, and lower power consumption.

Finishing-process developments considered are finish-cutting, burnishing, shaving, and grinding.

THE development of improved processes of manufacture is so continuous that it is seldom possible to discuss a development as a completed project. This condition is especially true of the automotive industry where new methods are originated, improved upon, and discarded so rapidly and with such continuity as to make it practically impossible to say where one development stops and another starts. It is possible, however, to describe processes and results up to a given time but these must be considered, not as accepted and immutable facts, but rather as positions by which to lay tomorrow's course. Nowhere is this more true than in an examination of gear production methods.

Probably the most recent experiment tending toward better gear production methods revolves about the application of so-called "climb-cutting" to gear hobbing operations. The impetus to this experiment was the marked success attending the application of climb-cutting to the hobbing of splines on axle drive shafts and on propeller-shaft fittings. Over a period of ten months actual production results from climb-hobbing of splines as compared to conventional hobbing were so uniformly better that it may be well to review them as being experience factors, whereas the application to gear hobbing is wholly experimental as yet.

Climb-hobbing may be defined simply as that method of hobbing wherein the cutting action starts at the surface of the part being hobbed and ends at the root of the spline or tooth. This method is the direct opposite of the conventional method in which the cut starts at the root and finishes at the surface of the part.

In the application of climb-hobbing to splines the only change necessary in the machines used was to change the direction of rotation of the hob. This change was accomplished by reversing the leads of the main-drive motor and inserting an idler in the index gear train. This arrangement reverses the cutting thrust so that the splines are hobbed against the tailstock instead of against the headstock spindle.

As experience with the method accumulated, various test runs were conducted with the object of eliminating influencing factors so the effect of climb-cutting could be judged solely as a single factor. Various types of coolants were used, but invariably the effective difference between climb and conventional hobbing remained uniform. If a given hob, part, and machine yielded a greater number of pieces per grind when climb-cutting than when used conventionally with soluble oil coolants, the percentage of improvement was not greatly affected when the coolant was changed to an organic paste compound, or any of the commonly used mineral or animal cutting oils.

The same uniformity of difference seemed to exist with regard to hob revolutions per minute, feed per revolution of the work, hardness of material worked upon, type of sharpening of hob, and design of arbors or other work-holding tools.

Finally, to eliminate errors due to comparative tests being run on machines of different degrees of rigidity, accuracy, and age, one machine was alternated between climb-hobbing and the conventional method on the same part over a period of several days, and detailed records kept of the factors in both methods.

From these months of experience and experiment certain conclusions were drawn as applying to climb-hobbing of splines. These are offered here for what they are worth and to display some of the influences which may affect results when the process is applied to gear teeth.

(1) Climb-hobbing always yields a finish on the sides of the splines obviously superior to that obtainable by conventional cutting. This superiority is observed so easily that customers, during the early stages of the application, brought much pressure to bear to hasten the completion of the project.

(2) Climb-hobbing is accompanied invariably by an increase in hob life. This increase normally amounts to from 20 per cent to 30 per cent more pieces per hob.

(3) Climb-hobbing consumes less power per cubic inch of metal removed. This decrease normally will be in the ratio of 1.00 hp. for climb hobbing to 1.10 hp. for conventional hobbing.

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(4) Climb-hobbing produces little or no burr on the splines, thereby reducing or, in some cases, eliminating subsequent burring operations.

(5) The value of climb-hobbing as a production process is not affected materially by coolants, speeds, feeds, or other factors in the operation.

From these conclusions experiments were planned with the object of applying the climb process to gear-tooth hobbing. Here, certain precautions were considered and planned as part of the program. For instance, in the spline-hobbing practices no bad effects were observed from the reversal of the cutting thrust from headstock to tailstock. With gears, however, it was felt that a cutting thrust against a tailstock and away from the spindle designed to take that thrust might lead to trouble in gear forms not noticeable in splines. However, climb-hobbing so that cutting thrusts would be against the work spindle required many changes in machine construction not obtained as easily as in the case of the spline-machine changes. Consequently, such additions and alterations were made to one machine to be used for test purposes as would permit throwing the cutter thrust against either the headstock or the tailstock. By tests on this machine it is hoped to determine the relative value of direction of cutter thrust.

Unfortunately, this machine has been available for test for such a short time prior to the writing of this paper that nothing can be said at the moment pertaining to this phase of the project. Furthermore, delay in getting the machine ready for test inevitably slowed down the other pending gear tests.

Some few tests have been completed, however, under the same conditions that governed the spline-shaft hobbing. These tests are encouraging in that the results conform in the main to those obtained on spline shafts. Although it is too early to draw conclusions, what little that has been observed indicates that the following factors will ultimately emerge as fairly basic:

(1) Finish of contour is superior with climb-hobbing but not to the degree observed in splines. This difference is probably due to the fact that finish of gear-tooth contours has long been given more careful consideration than that of splines. In addition, a school of thought has developed which argues that too fine a finish on a gear tooth that is to be shaved subsequently is not a good thing. This group holds that slight feed waves are an advantage in the shaving operation.

(2) Climb-hobbing of gear teeth is accompanied by the same increase in hob life as was apparent in the spline experience.

(3) Power consumption with climb-hobbing of gears remained in the same ratio to conventional hobbing consumption as was observed in the case of splines.

(4) Climb-hobbing of gear teeth does not affect accuracy of tooth form. This indication is astonishing in view of apparently opposite findings recently made public. Although it is only an indication and may be reversed by all succeeding tests, the fact remains that specimens of the first tests of climb-hobbed gears were not only as correct in tooth form as those produced by conventional methods, but actually a shade better.

The case for climb-hobbing of gear teeth rests at that stage for the moment. It would appear that there is no reason why results in that field should not be as gratifying as those obtained in spline-hobbing. Investigations are continuing, and application will be made as experience upholds or rejects these first indications.

At best, however, current gear-cutting methods leave much to be desired. As in most other metal-cutting operations, maximum accuracy and fine finish are not coexistent with heavy cuts and high feeds. Yet the gear teeth commonly used in automotive transmissions represent heavy cuts by any cutting process so the greatest accuracy and smoothest contours can be obtained only by putting the gears through a preliminary blocking operation to remove the major portion of the stock and then subjecting them to a finishing operation designed to remove a very small amount of stock with equally small deflections and strains in the parts and machine.

In the abstract this finishing operation can be performed satisfactorily on any of the common types of gear-cutting machines by the simple process of repeating the roughing operation with slightly different feeds and speeds due to the smaller amount of stock removed. In practice, however, the time required to perform the finishing operation is too high to warrant any approbation from the production or mechanical divisions charged with the manufacture of high-quality gears at low cost.

This lack of enthusiasm for the finish-cutting operation as such has led to the development of substitute methods which approximate or surpass finish-cutting results in far less time. The oldest of these substitutes is the burnishing process in which the production gear is rolled against a master under pressure. This method is so well known that it need not be examined further here. Suffice it to say that many gears where noise of operation and high load factors are not involved are still finished by this method prior to heat-treatment.

The other process for finishing gears prior to heat-treatment is that usually described as shaving. This process again is broken into two distinct types of shaving, namely, the rack method and the circular-cutter method. In this process the master is an accurately produced rack or circular cutter with small serrations on the tooth contours which, when advanced across the face of the production gear, act as cutting edges under slight pressure and actually shave fine amounts of metal from the tooth contours of the gear being worked on.

This method, in general, produces a very high quality gear with all the accuracy built into the master rack or circular cutter. The relative merits of the rack type as compared with the circular-cutter type have been discussed widely. Each method has its proponents, and the question as to which is best is not answered easily by the most impartial observer. Where the two types are in use side by side, certain factors present themselves repeatedly and with such force that it is impossible not to form some very definite convictions as to the value of each. These conclusions may be summarized as follows:

(1) It is more difficult to obtain an accurate master rack than to obtain an accurate master circular cutter. The master rack is also more expensive in first cost than is the circular cutter.

(2) The rack cannot be used as close to shoulders or other gears in a cluster as can the circular cutter. This is a matter of geometry that is inherent in the type and needs no exploration.

(3) There is a factor in shaving operations which is difficult to define but is commonly referred to as control. The extent of its importance has not been determined, but it can be shown that it is a very important element of good shaving. It must be understood that the gear being shaved is not restricted in its rotation with the master by anything except the master. It is not driven by dogs or collets and, if a series of points on its pitch circle pass through equal distances in equal times, it is because the master makes them do so and for no other reason. The gear is, therefore, under control of the master. It would seem then that, if it is desired to impart uniform angular velocity to a gear, it is only necessary that the tooth spacing of the master be uniform. Actually, some-

thing else is required before angular uniformity is obtained. That something is "wrap" or arcs of contact between gear and master. The greater the "wrap" – coupled with uniform spacing of teeth on the master – the more certain becomes the equal distance in equal times results. The rack type of shaving is superior in this respect to the circular-cutter type because of its inherently greater "wrap." It has more control over the gear being finished.

(4) The rack type is defective to the extent that any error in the master is transferred direct to the gear due to the fact that, as the gear is rotated, each of its teeth always meshes with the same respective space in the master. The circular-cutter type does not have this defect since the introduction of a hunting tooth will average errors around the entire circumference of the gear.

In short, it is evident that each machine has its type of gear for which it is better fitted than the other. Each new gear to be shaved must be analyzed in the light of the conclusions drawn and allocated to the type of machine most likely to produce the equal distances in equal times effect.

Subsequent to heat-treatment, gears are lapped, ground, or used without further finishing operations on the teeth. Where noise and load factors are not highly important, shaved gears, unless of poor distortion-resisting design, can be used without lapping or grinding of tooth forms. Burnished gears, however, due probably to the pressure method of producing the tooth form, usually require lapping to develop a proper bearing surface between contacting teeth.

Gears with sections which shrink non-uniformly in quenching must have the teeth ground to insure the proper spacing and form necessary for quietness and high-load characteristics. The grinding may be performed by the formed-wheel method, but it is usually some form of generation. In one type of machine a large-diameter dish wheel is dressed and set at such an angle that it represents one side of a rack tooth. The gear to be ground is rolled across this wheel, and one side of a tooth is finished at a time. This method, of course, requires putting the gear through the machine twice to finish it completely. The method is also limited to gear teeth having a face length and clearance within the arc and chordal characteristics of the grinding wheel.

A second type of gear grinder utilizes the same general principle, but is equipped with two large wheels so that two sides of a tooth may be ground at one time. Here the wheels are set to clear one tooth which rolls in the space between the wheels while the adjacent sides of the teeth on either side are ground. This method is restricted for face lengths the same as the single-wheel method.

The third type of grinding consists of a small wheel dressed to the shape of a complete rack tooth and passed through the gear-tooth spaces in short, shaper-like strokes, while the gear is rolled in the path of the wheel by a master. Here a complete tooth is ground at one time which means very rapid production. Face lengths are limited only by the stroke characteristics of the machine.

Continued experiment indicates that grinding as a means of producing satisfactory gears will be standard practice until such a time when metallurgists provide a gear steel which changes in heat-treatment less than the tolerances demanded in the finished gear, when engineers are able to design every gear with a section of uniform and predictable distortion characteristics.

In Support of the Camel-Back Truck

THE term "Camel Back" is neither descriptive nor euphonious and consequently is inappropriate. Two names "engine-under-seat" and "cab-over-engine" are descriptive of the two general types of short wheelbase vehicles which are being produced by various manufacturers at the present time. In the first of these the engine is located under the seat and in approximately the same relation to the front axle as in a conventional chassis. In the second, the engine is located adjacent to the front end of the chassis projecting upward through the floorboards and ahead of the front axle.

Our company builds both types. The "engine-under-seat" is used in our heavy-duty units where the engines and transmissions and rear axles are of our own design and manufacture. The size of the engines used in these heavier units is such that, if they should be mounted forward, they would either crowd the driver or require a very wide cab. Mounted under the seat, the only encroachment on the driver or rider is a narrow tunnel which affords a passage for the water outlet pipe and also an air passage from the fan to the engine compartment. It is obvious that this construction calls for some detail modification of the engine, transmission and rear axle which we, as the maker of these parts, can control.

On our lighter units, where cost considerations are of great importance, we use the "cab-over-engine" construction. Here the engines are smaller and can be housed conveniently and we can use standard production-run engines, transmissions, and axles which, if they were changed to meet the requirements of the "engine-under-seat" type, would be affected seriously in cost.

The considerations which affect this problem are evident; they may be enumerated as follows: (1) shorter overall length conserving space – garages – ferries – parking, and so on; (2) shorter wheelbase making turning easier due to smaller turning circle; (3) improved weight distribution; (4) reduced frame stresses; (5) distinctive appearance; and (6) greater driver comfort.

Shorter Overall Length – This feature has become imperative in recent years. The increasing density of traffic has pointed the need for shorter vehicles. The adoption of length regulations has merely given added emphasis to this need.

Shorter Wheelbase – Here again the advantage of a small-diameter turning circle for trucks has always been recognized.

Improved Weight Distribution – The load front and rear should be in the ratio of one to two. To do this with the conventional truck requires an excessively long wheelbase which is objectionable in many ways. On the engine-under-seat or the "cab-over-engine" it is simple.

Reduced Frame Stresses – The shorter distance between axles reduces the value of the bending moment which means a reduction in stress and deflection.

Distinctive Appearance – The elimination of the conventional hood permits the development of distinctive appearance.

Greater Driver Comfort – Cabs are more easily ventilated, vision is better, riding is better and driving generally is easier with the shorter vehicle.

Excerpts from the paper: "Engineering Facts in Support of the Camel-Back Truck," by B. B. Bachman, The Autocar Co., presented at the Southern California Section Meeting of the Society, Los Angeles, Calif., June 18, 1937.

Quality Control of Aluminum-Alloy Aircraft Castings

By Kent R. Van Horn[1] and Howard J. Heath[2]
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ARTICLES of a wide variety of sizes and shapes can be produced easily by the sand-casting process. The advantage of simplicity and low cost offered by the sand-casting process for small numbers of identical parts is offset by certain inherent limitations which must be recognized clearly if it is to yield optimum results. The various methods of inspecting and controlling structural variations that affect the strength and serviceability of aircraft castings are described.

It is recognized that the user of castings must rely to a considerable extent on the foundryman for careful inspection. Several direct tests worthy of consideration as final acceptance tests of quality, such as the proof test, radiography (X-ray examination), and the static breakdown test, are described. However, this discussion leads to the conclusion that there is not yet available a satisfactory direct method of predicting the normal life of an aircraft casting.

The most pertinent evidence that the practices described in the paper are effective is the service record of aluminum-alloy castings. Maintenance and improvement of the service record depend largely on the continued and more complete cooperation of aircraft designers and engineers with the foundry personnel.

ARTICLES of a wide variety of sizes and shapes can be produced easily by the sand-casting process. The necessary equipment is relatively inexpensive, and the process is extremely flexible. It is only when large numbers of identical parts are to be produced that it becomes economical to use other methods of fabrication. The advantage of simplicity and low cost offered by the sand-casting process for small numbers of identical pieces is offset by certain inherent limitations which must be recognized clearly if it is to yield optimum results.

Most structures, particularly in the field of transportation, are required to carry heavy loads with the lowest possible weight. Incidentally, the relatively high unit cost of special materials such as aluminum and magnesium alloys, stainless steel, and so on, is an important added incentive for weight saving. The design of such structures necessitates as accurate a knowledge as possible regarding the forces to be resisted and the properties of the structure carrying the loads. The mechanical properties of castings are, therefore, of importance to the designer, and consideration must be given to their integrity and reliability.

Practically all commercial alloys in the liquid state are less dense and, therefore, occupy a larger volume by several per cent than in the solid state. One of the fundamental problems in the production of castings is to control the sequence of solidification of the various portions of the casting so that liquid metal is available to compensate for solidification shrinkage and to minimize porosity in the casting.

Because of the manner in which sand castings are made, it becomes of importance that standards and methods of inspection be established in order that the engineer will have a definite basis for his design. Such a step in the negotiations too frequently is neglected with subsequent misunderstanding regarding acceptable quality. This paper will attempt to indicate some of the measures utilized in the development and control of quality in aluminum-alloy castings particularly for aircraft use. Some of the factors affecting the strength of castings will be enumerated and their control described.

Common Structural Variations

It may be well to consider the types of structural variation which affect the strength and serviceability of castings before discussing the methods of quality control.

The simplest type of casting is produced by molding a semi-plastic material, usually a mixture of sand, clay, and water, about a pattern of the structure to be made. After the mold has been formed, the pattern is removed and liquid metal poured into the cavity formed thereby. The limitations regarding dimensional tolerances are obvious, and it is appreciated readily that, the more rigid the dimensional specifications, the higher the costs.[3] It is evident that, the better the pattern equipment, the smaller the variation in dimensions. It is uneconomical to demand the same tolerances for castings from loose wood patterns as from metal match-plate equipment.

There are, of course, modern refinements of the sand-casting

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[3] See American Foundrymen's Association Preprint 31-40, 1932, pp. 1-23: "Foundry Considerations in Design of Aluminum-Alloy Castings," by H. J. Rowe.

process which are capable of yielding castings within narrower dimensional tolerances. Metal molds may be used for part or all of the casting with tolerances in the thirty-secondths rather than in the sixteenths of an inch. Similarly, by forcing the metal under pressure into steel dies, as in the die-casting process, even smaller dimensional tolerances may be achieved. By recognizing the limitations of each process and by not requiring dimensional tolerances and finishes not inherent in the process, the designer will insure against excessive castings costs.[3]

As previously indicated, one of the fundamental problems in the production of castings is the arrangement of the sequence of solidification so that the change in the unit volume of the metal on solidification can be compensated for satisfactorily.[4] If this result is not accomplished, voids of various sizes, types, and distribution will be found in the solid metal. These discontinuities are termed "shrinkage porosity" and will be considered in more detail in the section on radiographic examination. Trained inspectors usually can recognize the presence of bad shrinkage areas by the surface appearance of the casting. Small discontinuities of this type, however, are recognized only by sectioning which method, of course, destroys the casting, or by radiographic inspection.

Another common discontinuity is designated as a "blowhole" by the foundry. The green sand mold always contains moisture as a necessary factor in its composition. Under some conditions, the rapid local evaporation of this water by the hot metal forms pockets of water vapor or steam in the liquid metal which then solidifies before the water vapor has an opportunity to escape. Blowholes have a characteristic appearance, are generally found on or near the surface of the casting, and sand-blasting usually renders them easily recognizable on visual inspection.

If the metal is poured too cold, the individual streams of liquid in the mold may not unite completely before solidification occurs. This forms a "cold shut" which is usually apparent on the surface of the casting. An aggravated form of "cold shut" is encountered when a portion of the mold fails to fill. This defect is called a "misrun"; it is recognized so easily that the user of castings seldom encounters an example of it.

Molten metal has a tendency to absorb gas which subsequently is given off during solidification. This tendency to absorb gas increases with temperature and is affected by many foundry conditions. The gas liberated on solidification frequently is trapped, forming small holes quite generally distributed throughout the casting.[5] These holes are known as gas or pinhole porosity, and are usually evident in the casting as a whole only on a carefully machined surface or a radiographic negative. Fortunately, as illustrated in Fig. 1, gas porosity, although undesirable from an appearance standpoint, has probably less effect on the properties of the casting than many other quality factors. The tensile strength of specimens cut from castings is reduced a maximum of only about 7 per cent by a difference in porosity much greater than that allowed in production aircraft castings. At the present time, it is not possible to eliminate gas porosity completely from commercial sand castings, although its size and distribution may be governed to a considerable extent. For example, gas porosity is not particularly evident in small or severely chilled castings largely because of its small size. The total volume of voids, however, would be found to be of the same order of magnitude as in more slowly cooled sand castings. Voids resulting from gas porosity and shrinkage are sometimes confused. The photographs and radiographs of Fig. 2 illustrate the fundamental difference, the gas porosity being distributed generally throughout the casting, whereas shrinkage occurs only in localized areas.

Practically all commercial castings are made of alloys rather than pure metals. An inherent characteristic of the solidification process in alloys tends to increase the concentration of the alloying material in the liquid metal while the portion first solidified contains relatively low concentrations of alloying material. This concentration results in microscopic segregation and non-uniformity of composition. In the case of alloying elements which are soluble in the solid metal, this non-uniformity in composition can be reduced and the metal sometimes made homogeneous by the heat-treatment of the casting. The distribution of alloying elements which are not soluble in solid metal cannot be changed after solidification. Only under very unusual conditions does this segregation become macroscopic and of importance in the service of the casting. When present to such an extent, segregation generally is accompanied by an unusual amount of shrinkage and other structural characteristics which are recognized readily.

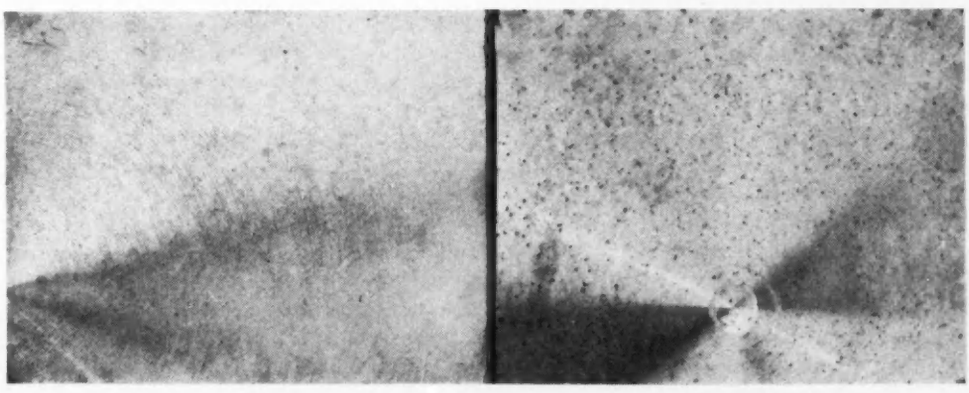
Cores are required in many castings to form hollow spaces within the casting. These cores are made from sand held together with a suitable binder and are more or less rigid. The thermal contraction of the solidified metal about these rigid cores sets up stresses which are sometimes sufficient to cause cracking of the casting.[6] The recognition of such cracks is usually relatively easy for an experienced inspector. The designer must keep in mind the necessity for visual inspection and not so design a casting as to make this inspection impossible or unduly difficult.

The fractured surfaces of heat-treated castings sometimes exhibit various degrees of discoloration as contrasted with the

[4] See Transactions of the American Foundrymen's Association, 1930, Vol. 38, pp. 65-91: "Stronger Aluminum Castings by Improved Foundry Practice," by T. D. Stay, E. M. Gingerich, and H. J. Rowe.

[5] See Transactions of the American Foundrymen's Association, 1932, Vol. 40, pp. 527-546: "Practical Foundry Considerations for Improving the Soundness of Aluminum-Alloy Castings," by H. J. Rowe and E. M. Gingerich.

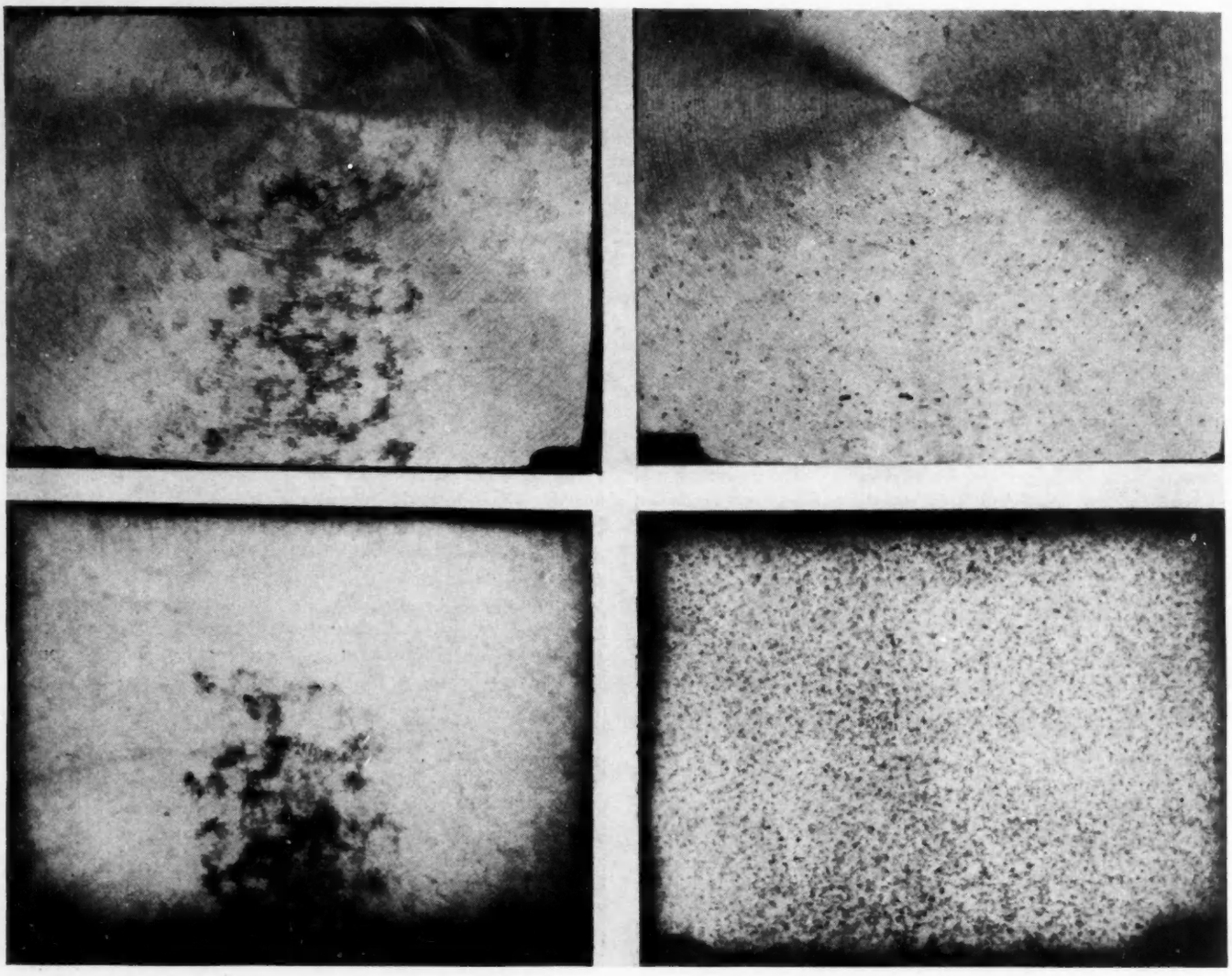
[6] See Transactions of the American Foundrymen's Association, 1931, Vol. 39, pp. 297-310: "Molding Practice for Heat-Treated Aluminum-Alloy Castings," by L. H. Fawcett.



Section Containing Normal Gas Content – A Small Degree of Porosity Is Evident – Average Tensile Strength, 28,820 Lb. per Sq. In.; Elongation, 4.5 Per Cent

Section Containing Unusually High Gas Content – General Heavy Porosity Is Revealed – Average Tensile Strength, 26,740 Lb. per Sq. In.; Elongation, 3.1 Per Cent

Fig. 1 – Photographs of ¾-In. Thick 195-T4 Alloy Sand-Cast Sections



Section Containing Shrinkage Porosity — Section Containing Gas (Pinhole) Porosity

Fig. 2 – Photographs (top) and Radiographs (bottom) of 3 In. Thick Sections Removed from 195-T4 Alloy Sand Castings

bright, highly reflectively fractured, surfaces that are characteristic of non-heat-treated castings. Similarly, the exterior surface of a heat-treated casting is usually a dull gray. This appearance is thought to be the result of a superficial reaction between the furnace atmosphere and the metal. Extensive investigation has failed to establish any definite correlation between degree and amount of discoloration and variation in mechanical properties.

Indirect Methods of Quality Control

Like all pure metals, aluminum and magnesium, because of their solidification characteristics, cannot be cast easily. They also lack suitable mechanical properties. Alloying is resorted to for increasing the strength and improving the casting characteristics. The determination of optimum compositions involves extensive investigation in laboratory and foundry. After production limits have been established, it is necessary to maintain the composition in large-scale production. This control is accomplished usually by alloying in large heats, analyzing carefully, and remelting in small quantities for making castings.

The optimum pouring temperature is specific for each casting; small-quantity remelting usually is to be preferred because it is advisable to cast from the lowest possible temperature with as little overheating as is necessary.[4, 5, 6] The curves of Fig. 3 illustrate the decrease in mechanical properties with increase in casting temperature of 195 and 220 alloys. The minimum pouring temperature usually is established by the thinnest section of the casting. By giving careful consideration to this fact, it may be possible for the designer, in close consultation with the foundryman, to obtain relatively higher properties in the highly stressed and, thus, thick sections by small increases in thickness of the thinnest sections. This procedure frequently will result in greater net savings in weight than are obtainable by reducing the unstressed sections to minimum thickness.

Relatively small variations in composition usually have greater effect on casting properties than on the strength of the test bars. It is even becoming increasingly common for the foundry to control composition by spectroscopic means, thus restricting the variation in concentration of elements which normally occurs only in quantities formerly reported as traces. Similarly, the structure of an alloy may affect markedly the properties and casting characteristics. The microscope is called upon to assist in maintaining uniformity in these attributes.

Even chemical composition and microstructure are not considered sufficient indications of the properties of the metal in the casting, and it is customary to cast test specimens from the identical melt of metal from which the casting is made. These specimens are tested to destruction for the determination of yield strength, tensile strength, hardness, and ductility. It is not intended that these separately cast test specimens

should be a quantitative measure of the strength of the casting since it is only rarely and almost pure coincidence that the casting conditions of every part and section of a commercial structure can be reproduced in test sections. The assumption that the properties of a commercial casting can be represented accurately by small test bars is responsible for considerable discussion in trade circles regarding the manner of casting test coupons. It can be demonstrated readily that the tensile properties of test coupons can vary over a wide range independent of the properties of the casting even when the coupon is attached to the casting and apparently cast under similar conditions.[4] Under such circumstances, it appears more logical to cast separate test specimens under standardized conditions and consider the data obtained with them, similar to that from chemical analysis, as an indication of whether the molten metal as poured into the casting meets certain standards and is capable of producing castings of specific properties. With heat-treatable castings, such separately cast test

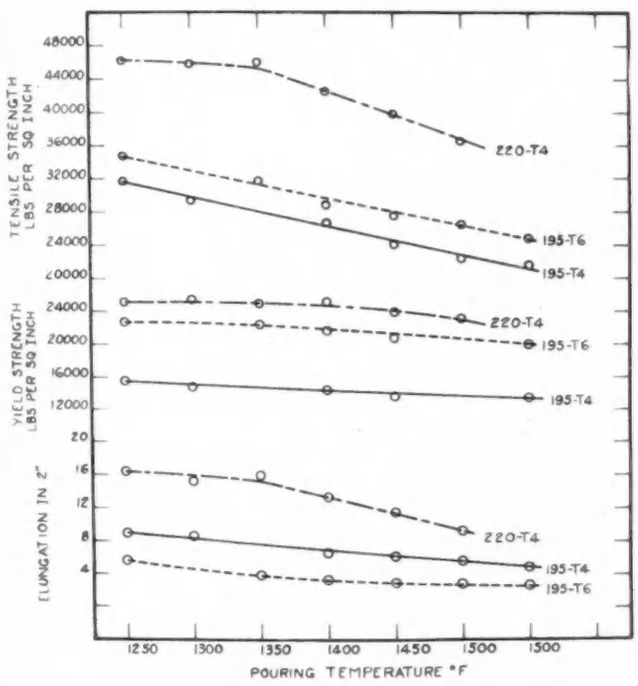


Fig. 3 – Effect of Pouring Temperature on the Tensile Properties of Separately Sand-Cast Test Bars of Alcoa 195-T4, 195-T6, and 220-T4 Alloys

specimens, when heat-treated with the castings, can also indicate the quality of the heat-treatment. Through long experience and the investigation of many castings, some foundries have been able to establish qualitative relationships between test specimens cast under standardized conditions and the mean properties of similar test specimens cut from various sections of well-made commercial castings.[4]

It is an interesting fact, even when the tensile strength of bars cut from a casting varies considerably from the tensile strength of the accompanying separately cast specimens, that the corresponding yield strengths are much more nearly equal. This is an important point inasmuch as the designer usually is interested in conditions required for yielding rather than for breaking. Even such data, however, may not give an accurate indication of the ability of the casting to withstand service loads. Small discontinuities, which would not affect appreciably the strength of the casting as a whole, might reduce greatly the properties of a test specimen with relatively small cross-sectional area.

Usually one of the major reasons for fabricating a specific structure by the casting process is that its complexity precludes economical production by any other means. In fact, a casting as a whole not infrequently will have a better combination of properties with less weight than a built-up structure, principally because of the difficulty of distributing the metal to the best advantage by any other method of fabrication. Usually the various portions of a casting are not required to resist stresses of equal magnitudes; frequently machining, casting, or even purely esthetic considerations require sections much heavier than are demanded by loads to be sustained. It is, therefore, uneconomical to attempt to obtain uniform strengths in all portions of a casting. Optimum results will be obtained by the close cooperation of the designing engineer and foundryman looking toward the production of a design and a casting which will sustain satisfactorily the imposed loads at the lowest total cost.

Direct Methods of Quality Control

Proof-Load Test. – It is recognized that the user of castings must rely to a considerable extent on the foundryman for careful inspection. There are, however, several tests worthy of the user's consideration as final acceptance tests of quality. One of these is the proof-load test whereby each casting is subjected in a specific manner to a load which bears some relation to the design stress. It is assumed that the ability to resist deformation under this load, other than a specific, small amount, is a satisfactory measure of the capacity of the casting to meet the service requirements.

It may be advantageous to describe an example of an application of this type of test. A base casting for a portable road scale arranged for proof-testing as a simple beam, supported at the ends by means of V-blocks and loaded in the center, is represented in Fig. 4. The strain gages are shown attached to the casting for the measurement of strains occurring during the application of the load. Dial indicators also are placed beneath the casting to determine the deflection produced by definite loads, as well as to obtain the permanent deformation of the casting after the proof load has been released.

The proof-load test has certain limitations which tend to restrict extensive use. It is applicable only in the case of well-determined and simple types of loading. In most applications the service loads are so little known as to make difficult or impossible the definition of a proof load which could be taken as a satisfactory measure of the ability of the casting to withstand service conditions. The expense of an elaborate set-up and the time necessary to perform the test frequently may

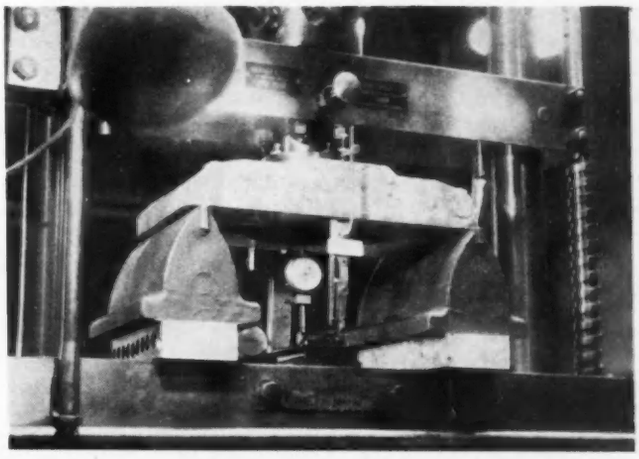


Fig. 4 – Base Casting for a Portable Road Scale Arranged for a Proof Test as a Simple Beam Supported at the Ends by V-Blocks and Loaded in the Center

warrant the selection of another form of quality inspection. The proof-load test yields data on the properties of the casting principally in the elastic range, but gives no indication of the behavior of the casting beyond the yield strength.

It is important, in specifying the proof test, to decide who shall be qualified to prescribe the proof load and in what manner this load is determined. This may be a relatively easy task in simple beam sections but may become increasingly difficult in more complicated aircraft castings.

Radiographic Examination

Radiographic methods may be used for the direct examination of castings.[7] They have the advantage of being non-destructive and warrant more general consideration.

The degree of absorption of X-rays is a direct function of the density of the substance exposed to them; that is, all materials do not possess the same degree of X-ray transparency. A sound section of steel will absorb more radiation than a similar section of aluminum. With X-rays it is possible, therefore, to locate inclusions differing appreciably in density from the surrounding material. These relations are the basis of radiography, which may be defined as the non-destructive testing of matter with radiant energy (either X-rays or gamma rays). The experimental procedure consists of passing radiation through the specimen and detecting the various intensities of the emergent beam by a photographic film or fluoroscopic screen.

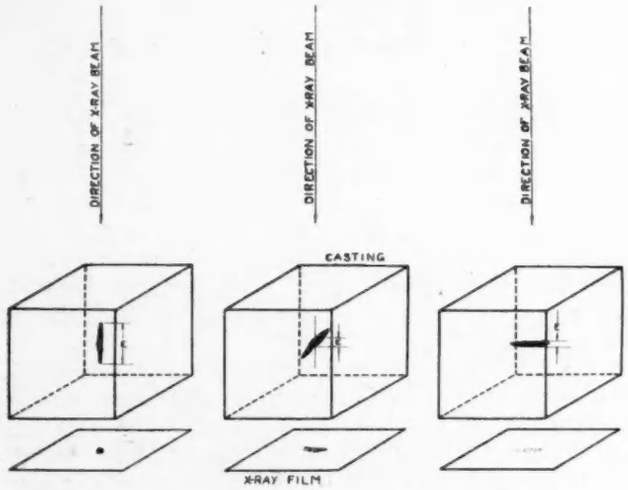


Fig. 5 – Diagram Illustrating the Effect of Defects in a Casting Inclined at Various Angles to the Path of the X-Rays, on Radiographic Sensitivity

Sensitivity: General Technique. – The minimum perceptible variation in density within a specimen is the measure of the overall sensitivity of the radiographic method which may be expressed as the smallest thickness of defect in inches that is revealed on the radiograph.[7] It also may be considered as the percentage of the total thickness examined, occupied by the smallest defect detectable on the radiograph. Some of the factors that affect sensitivity may be controlled, whereas others are permanent limitations imposed by the apparatus.

The technique for maximum sensitivity and definition will consist of using a fine-focus X-ray tube, as low voltage as possible, large tube-to-film distance, high-contrast film exposed with intensifying screens, and with the scattered radiation and chemical fogging reduced to a minimum. The size of the focal area of the X-ray tube is one of the factors that determine the definition of the macroscopic defects but does not affect the radiographic sensitivity. Theoretically, the smaller the focal area, the greater will be the definition. Fortunately there is a considerable range of focal areas, such as embraced by the "fine-focus" class of commercial X-ray tubes, within which variation in area of focal spot has no practical effect on the definition of the defects. The finest focus tube should be selected which will penetrate economically the various sections encountered. It is sometimes claimed that large-capacity (high-voltage) equipment is more sensitive than a lower capacity, which is not the case as the contrast (sensitivity) decreases with an increase in applied potential. The advantage of larger capacity is greater penetration. Sensitivity values are available and indicate the smallest detectable difference in thickness that may be obtained parallel to the path of the X-radiation. In 1- to 2-in. sections of an aluminum alloy a sensitivity of at least 1 per cent is obtained. Generally in sections less than 1 in., a sensitivity of 0.5 per cent is attained. (A 0.5 per cent sensitivity in a 1-in. section of an aluminum casting indicates that defects as small as 0.005 in. will be revealed.) In sections between 2 and 5 in. of aluminum, a sensitivity of 1.5-2 per cent may be expected.

Cavities, cracks, and dross films in a certain section frequently may present too small a variation in thickness or composition parallel to the X-radiation for detection. Yet discontinuities of similar dimensions whose longitudinal axes approximately coincide with the path of the X-rays might be revealed readily. This point is illustrated in Fig. 5 where the effective thickness of the cavity (distance E) parallel to the X-ray beam determines whether the cavity will escape detection. If the dimension E is less than the radiographic sensitivity, the discontinuity will not be revealed.

Obviously radiography is no measure of the discontinuities which escape detection by this method. X-rays do not have the capacity either to magnify or to reduce the size of the various defects. Minute imperfections, smaller than can be revealed by X-rays, might be serious stress raisers because of a rapid change in dimensions. It is known that a casting containing microscopic channels but no apparent macroscopic defects frequently will possess lower properties than a similar casting containing heavy general porosity or localized blowholes. The remarks concerning microscopic imperfections have been introduced because of some tendency to consider radiography a sure method of revealing all casting discontinuities.

Interpretation of Radiographs. – A radiograph is a shadow picture of material more or less transparent to radiation. The X-rays darken the film so that regions of lower density which readily permit penetration appear dark on the negative in comparison with areas of higher density which absorb more of

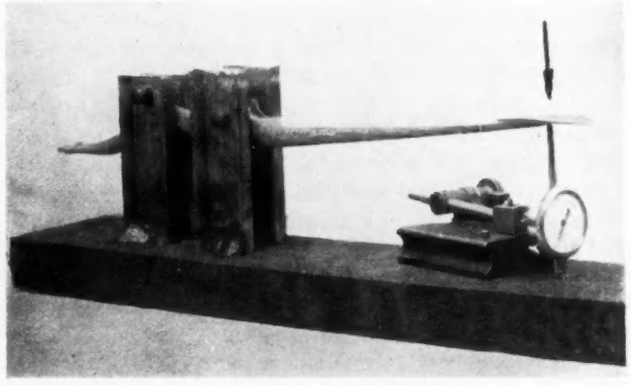


Fig. 6 – Arrangement for a Static Breakdown Test of an Airplane Lever Casting

[7] See National Metals Handbook, 1936, pp. 70-77: "Radiography of Metal," by Kent R. Van Horn.

the radiation. Thus, cavities in metals are recorded as darker areas, whereas heavy inclusions, such as iron-scale or copper segregation in an aluminum alloy, register a correspondingly lighter region on the radiograph.

In castings, differences in thickness due to the form, and intentional cavities, as cored holes, are regions of low density and are revealed as darkened areas. Castings containing variations in thickness caused by the unintentional absence of metal also result in darkening of the film. Excess metal and inclusions more dense than the base metal appear as light regions. The casting discontinuities of the latter two types may be recognized on the negative by the following characteristics:[7]

(1) Surface roughness appears as white or dark irregular areas with a smooth contour. The coloration depends, of course, on whether the surface protrudes or is depressed. The *surface irregularities must be compared carefully with the radiograph and identified before an attempt is made to interpret the internal discontinuities.*

(2) Gas cavities, blowholes, and minute pinhole porosity appear as well-defined spherical or rounded darkened areas.

(3) Shrinkage, pipe, and shrinkage porosity are represented by filamentary or dendritic dark regions of irregular dimensions and indistinct outline.

(4) Cold shuts appear as very prominent darkened areas of variable dimensions with definite, smooth outlines where the molten metal failed to unite completely before solidification.

(5) Cracks are represented by darkened lines of variable width and are filamentary when derived from insufficient feeding or contraction (shrinkage) on cooling from the melt. Cracks originating from internal stress in solid metal are either angular or straight and are of more constant width.

(6) Sand inclusions occur as gray to black spots of an uneven or granular texture with indistinct extremities.

(7) Inclusions such as copper or iron segregation, and foreign matter of greater density than aluminum, occur as well-defined, white areas. Dross inclusions appear as darkened regions of irregular and indistinct outline on the radiograph. The darkened area generally varies in intensity, indicating variations in the quantity of aluminum oxide and air.

The present principal disadvantage of radiographic examination is the lack of experience on which to base a correlation of structure, as revealed by radiography, and service life. Practically any properly made radiograph of a commercial casting will reveal discontinuities of structure, and a great deal of judgment must be exercised in using radiography as an inspection tool. The foundryman has made good use of radiography to improve his technique; it remains for the user of castings to equal this diligence by establishing a background of experience in relating radiographic structure to service life. A minor disadvantage of radiographic inspection of aircraft castings is its cost. This amount usually is expressed in terms of area of film and is generally within the range $1 to $3 per sq. ft.[8, 9, 10]

Static Breakdown Test. – The static breakdown test, although not yielding data directly applicable to service life inasmuch as the casting tested is destroyed completely, does give valuable information regarding design and foundry technique. It is classified here under the direct type of test largely for convenience of treatment. If only one of the many tools for developing and testing aircraft castings were available, this method probably would be selected as the most valuable. The effects of casting conditions, existing discontinuities, the properties of the alloy, and so on, are integrated in one overall result. Here again it is believed that the castings user has not given this method of investigation all the consideration that it deserves, particularly with regard to correlation of tests and service life. The foundry is able to obtain maximum benefit of the breakdown test only through the cooperation of the aircraft designer. The designer submits a print of the drawing of the part showing the magnitude and direction of service loads. This information permits the foundry to decide how best to adjust foundry technique to provide maximum strength and soundness in critical, if not in all, sections. In questionable cases the stress data enable comprehensive tests to be conducted in the laboratory on the actual casting, the results of which can be used for the improvement of foundry technique or design if required.[4]

The static breakdown test necessitates the construction of a fixture which will allow the loads to be applied, in a suitable testing machine, in the same direction as service loads. The static loads may be either tension or compression or combina-

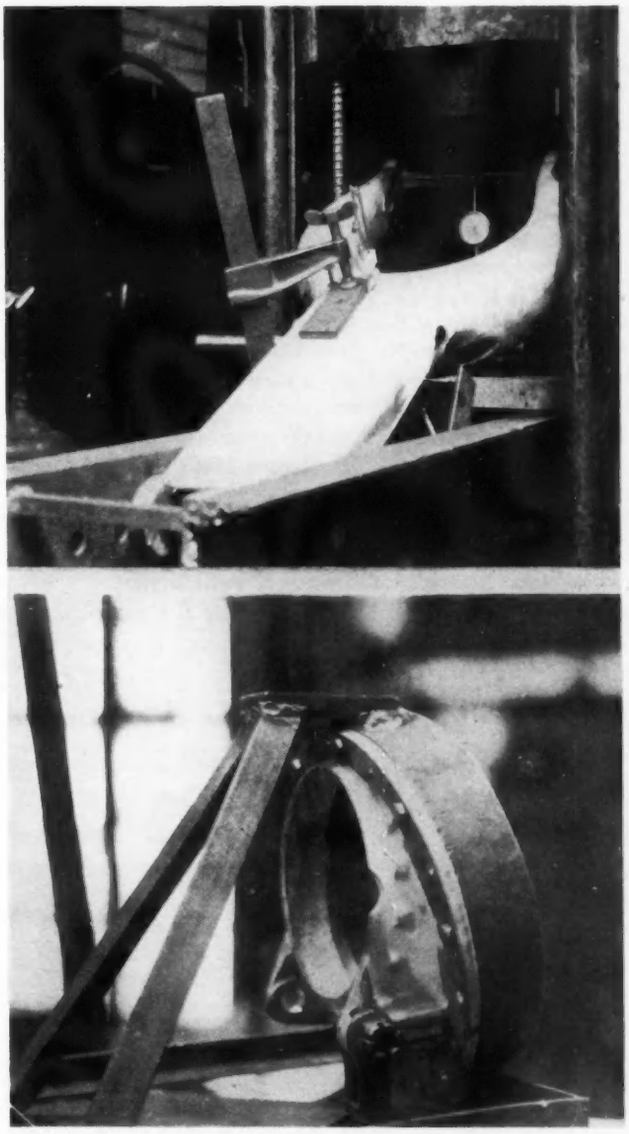


Fig. 7 (*Above*) – Airplane Tail-Wheel Fork Mounted on a Tensile Testing Machine for a Static Breakdown Test

Fig. 8 (*Below*) – Arrangement for a Static Breakdown Test of a Housing for the Lower Bearing of an Airplane Tail-Wheel Spindle

[8] See Transactions of the American Foundrymen's Association, 1931, Vol. 39, pp. 174-194: "X-Ray as a Production Tool to improve Quality of Aluminum-Alloy Castings," by E. M. Gingerich and H. J. Rowe.

[9] See Transactions of the American Society for Steel Treating, 1929, Vol. 16, pp. 551-599: "Radiography as a Tool in the Metal Industry," by W. L. Fink and R. S. Archer.

[10] See Transactions of the American Foundrymen's Association, Vol. 45, 1937: "X-Ray as an Aid in the Manufacture of Aluminum Castings," by G. E. Stoll and A. T. Ruppe.

tion of the two forces. The load-marked print of the part to be tested serves as a guide in establishing the location and the size of holes by means of which the casting is fastened securely to the test fixture. Deformation or strain may be measured by means of indicators or strain gages, and load-deflection curves can be plotted. This procedure permits a complete survey of the casting under the applied static load up to and including the breaking load.

A simple form of breakdown test is illustrated in Fig. 6 which shows an airplane lever casting supported in the test fixture in such a manner that a dial indicator can be placed beneath the point of loading to measure the deflection produced by definite loads. The load is applied in the direction represented by the arrow in the photograph. The indicator is removed when the yield strength of the alloy is exceeded, and the casting is then stressed to failure.

An airplane tail-wheel fork mounted on the tensile machine for a breakdown test is represented in Fig. 7. In addition to the dial indicator beneath the point of loading, a strain gage is attached to the yoke section to measure the strains produced in this area. Thus, in addition to the determination of the ultimate breaking strength of the casting, it is possible to calculate the magnitude of the developed stress in the outer fibers of the casting at this point which may be compared with the service requirements recorded on the load-marked print of the fork.

The arrangement for the breakdown test of a housing for the lower bearing of the tail-wheel spindle of an airplane is illustrated in Fig. 8. The casting is attached to a fixture by means of bolts and the load is applied vertically downward through the clevis shown in the center foreground. The point of interest in this test is the determination of the ultimate breaking strength of the two lugs.

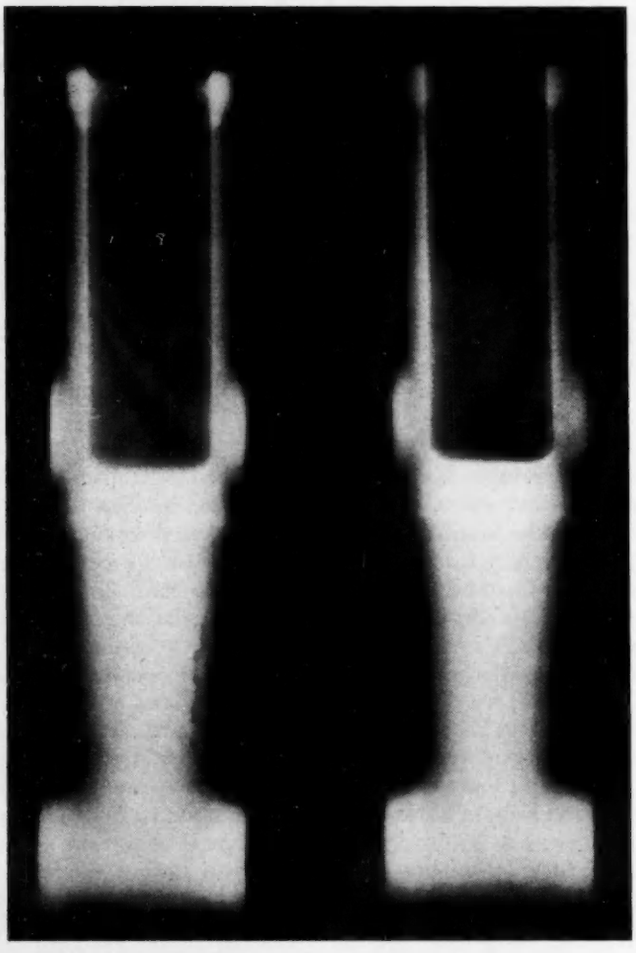


Fig. 9 – Radiographs of an Airplane Hydraulic-Pump Handle Socket Casting – Casting at Left Contains Large Shrinkage Cavity in Heavy Section That Was Eliminated in the Casting at the Right by a Gating Revision

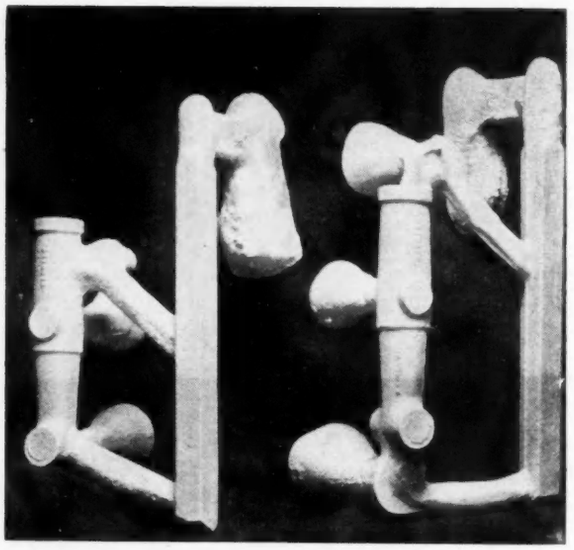


Fig. 10 – Incorrect (*left*) and Satisfactory (*right*) Method of Gating an Airplane Hydraulic-Pump Handle Socket Casting – A Relocation of the Riser at the End, the Addition of Another Riser at Opposite End, and a Form Chill Eliminated the Shrinkage Cavity in the Heavy Section

Tests and Service Performance

This brief and somewhat cursory discussion of methods of testing and control leads to the inevitable conclusion that there is not yet available a satisfactory direct method of predicting the normal service life of a casting. All the tests described have found considerable use in the hands of the foundryman as a means of developing and improving the technique of making castings.

Over a period of time and with the accumulation of experience, such tests also may serve as a background for an estimation of the probability of a casting meeting service requirements. The Aluminum Co. of America has made extensive use of all methods of quality control.[4] Two radiographic units have been installed in the foundry division, the first in the Cleveland, O., works in 1927, the second in the plant at Fairfield, Conn., and a third is about to be placed in service at the new Los Angeles, Calif., plant. This X-ray equipment has been used primarily to assist in the development of proper casting technique.[8, 9] With our present knowledge of the mechanism of the casting process, it seems likely that any serious defects occurring in a particular casting would be present in corresponding locations in other castings of the same design. Consequently, the first castings that are made are examined immediately by the X-ray to determine, as soon as possible, whether the foundry practice has produced a serviceable part. If not, the foundry technique is altered until a procedure is established which will yield a satisfactory casting. Often a number of gating methods will produce acceptable appearing castings, but wide variation in mechanical properties. The gating which permits the highest mechanical properties in critical sections, combined with the maximum casting soundness, is selected. Frequently this choice may require but a minor change, such as chilling or shifting a gate or riser a short distance. After the proper gating technique has been determined, additional castings are inspected occa-

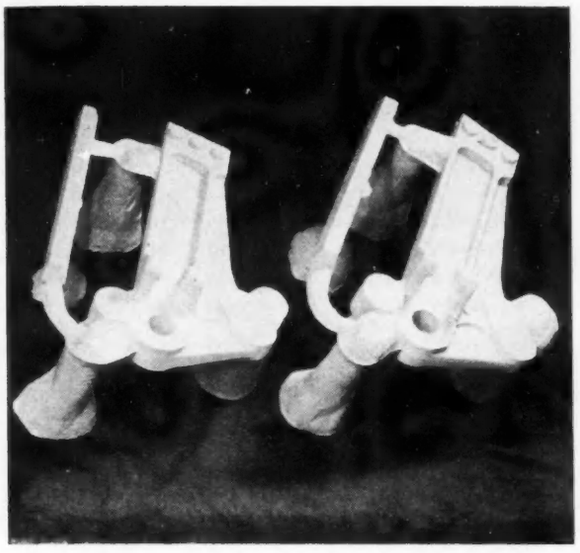


Fig. 11 – Inferior (*left*) and Satisfactory (*right*) Method of Casting an Outboard Support Central Section of an Airplane Engine-Mount Casting – The Addition of Chills Without Altering the Gates and Risers Reduced the Extent and Size of the Porosity

sionally during production to assure that no inadvertent variation in foundry practice has occurred.

A typical example of the X-ray control of foundry practice is illustrated by the radiographs reproduced in Fig. 9 and the gating photographs shown in Fig. 10. The first production of this simple aircraft casting, a hydraulic-pump handle socket, using a type of gating which was thought to be correct from the foundry standpoint, resulted in a large shrinkage cavity in the main body (radiograph at the left of Fig. 9). Relocating one riser, placing an additional riser at the other end, and a form chill in the drag (bottom half) of the mold of the offending area combined to produce a sound casting. It should be noticed that the riser at the cored end is placed so that trimming is relatively easy. Fig. 10 shows both the initial and final methods of gating.

Figs. 11 and 12 also illustrate the X-ray control of soundness in an outboard support central section of an airplane-engine mount. It was found that the heavy general porosity, as revealed on the radiograph, could be prevented effectively by greater chilling without any change in the arrangement of the gates and risers. The two methods of casting are illustrated in Fig. 11.

In simple castings the macroscopic discontinuities may be eliminated or reduced appreciably. In more complicated castings containing thin sections adjoining thick sections, attempts are made to reduce or transfer the inevitable discontinuities from critical areas. It is quite important, therefore, to consider the load-marked print of a casting in determining the significance of the density variations revealed on the X-ray film. The aircraft designer should be the best judge of what constitutes serious discontinuities in the highly stressed portions of a casting. In the interest of economy the designer has selected a casting for use in the plane structure. In so doing he expects and allows for reasonable discontinuities in cast metal. The views of the designer are incorporated in the load-marked print which should be compared with the radiograph in developing foundry practice.

A number of aircraft castings, which were considered questionable for service on the basis of radiographic defects, were subjected to breakdown tests in accordance with load-marked prints. These tests indicated that, even though there were evident macroscopic discontinuities in the castings, these same castings were many times stronger than required by the aircraft designer. Some of the castings contained general heavy pinhole porosity, thus emphasizing the minor effect of this discontinuity on the properties of castings. It would seem that the value of the X-ray method to the foundry is in locating the larger macroscopic defects such as blowholes and internal shrinkage, particularly the latter.

It is, of course, true that the proof or breakdown tests represent static loading and are not necessarily a measure of the strength of the castings when subjected to vibration or impact loads. In fact there are few tests available which give an indication of the serviceability of a casting under a reversal of stress or bending. There is relatively little known concerning the relation of defects in the macrostructure, or of the breaking strength of a casting, to the endurance qualities. When designers are able to state definitely the magnitude of the vibrating stresses that occur in a particular section during service, metallurgists will be able to predict the safe sectional thicknesses of castings or forgings. Until the designer can determine the relation of dynamic to static stress in aircraft parts, it will be necessary to continue to compute a static stress and add an additional factor to compensate for these indeterminate stresses. At present, the load-marked blueprints endeavor to allow for these indeterminate stresses by specifying loads appreciably in excess of the calculated loads.

It has been suggested that fatigue test bars, or impact specimens removed from the castings, would solve the problem. The reasons for the unsuitability of this procedure are the same as those previously given in the discussion of the disadvantages of machining tensile-test bars from castings. In addition, the sections in most aircraft castings are too thin to provide standard impact or fatigue specimens. Such variations as the location of the area tested with reference to chills, gates, or risers, the size of the section, and casting conditions, induce

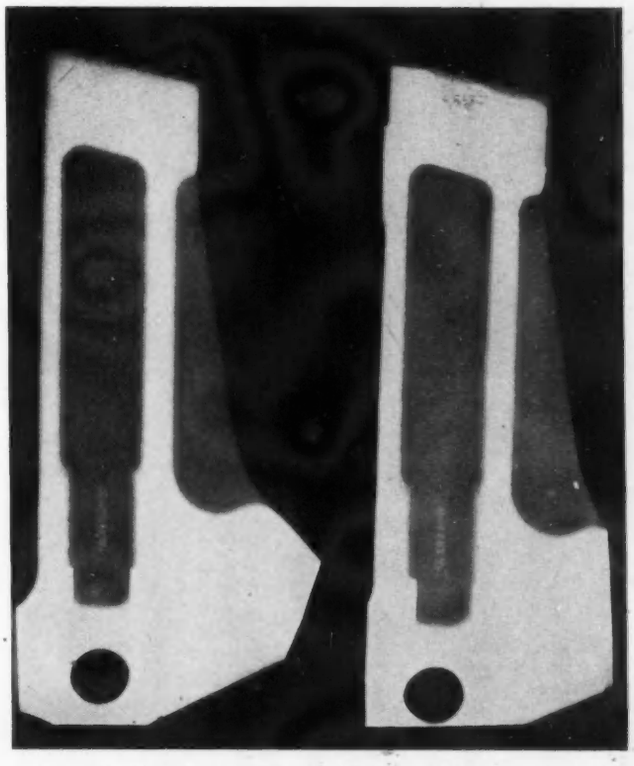


Fig. 12 – Radiographs of an Outboard Support Central Section of an Airplane-Engine Casting – Casting at Left Contains Heavy General Porosity Which Was Produced by the Original Casting Procedure – The Use of Additional Chills on the Casting at the Right Reduced the Extent and Size of the Porosity

considerable variations in the properties of individual castings. A large number of tests would be required before data of this type would be significant. No aircraft manufacturer would think of removing a section from a casting and utilizing it in an aircraft assembly and expect it to render efficient service; it is just as illogical to expect a test specimen removed from a casting to be a true indication of the strength of the entire casting. The casting is used as a unit on the airplane and should, therefore, be tested and inspected as a unit.

As an example of the effect of discontinuities in a casting on tensile, fatigue, and impact characteristics, the properties of die castings may be considered. This type of casting usually contains considerable internal unsoundness, as indicated in the radiograph reproduced in Fig. 13, but has a sound outer surface. Die castings, as a result, have excellent mechanical properties as long as the outer skin remains intact. Typical properties follow for an 8 per cent copper, 3 per cent silicon aluminum alloy cast in sand and in a cold steel die:

	Tensile Strength, lb. per sq. in.	Endurance Limit, lb. per sq. in.	Charpy Impact, (ft-lb.)
Sand Cast	23,000	7,500	0.6
Die Cast	32,000	16,000	3.0

The die-cast fatigue bars before testing are shown at the bottom of Fig. 13, whereas the sand-cast bars at the top happened to have been exposed after failure. Obviously the surface of a casting, which can be inspected readily, is the most important part of the structure from the standpoint of resisting static, alternating, and impact stresses.

In regard to predicting serviceability of a casting under a reversal of stress, there is one helpful criterion. Generally, the higher the tensile strength, the greater the resistance to alternating stresses. Consequently, the higher the ultimate breaking strength of a casting, the higher will be the resistance to vibrating stresses. It is for this reason that the improvement in breakdown strengths of castings and tensile properties of alloys has received so much attention in the aluminum foundry.

It would seem that the effects of discontinuities on the properties of castings which are subjected to vibrating stress can be determined best by service experience because of the lack of a suitable test procedure. There is sufficient information at hand to demonstrate that the omission of fillets in casting or finish-machining and the presence of sharp notches on the surface have a greater effect on the resistance of a casting to fatigue than the usual casting discontinuities. At the risk of appearing to stress the obvious, it must be reiterated, since the majority of aircraft-casting failures which have come to our attention have been the result of sudden changes in section, that any form of stress-raiser must be avoided.

The most pertinent evidence that the practices described in this paper have been effective is the service record of aircraft castings. During the past ten years about 4,000,000 aircraft castings weighing 3,000,000 lb. have been produced by the Aluminum Co. of America. Only a few of these castings produced under rigid control of tensile properties, chemical composition, casting discontinuities, and so on, have failed in service. Maintenance, or what is more desirable, improvement of the service record of aircraft castings depends largely on continued and more complete cooperation of aircraft designers and engineers with the foundry personnel.

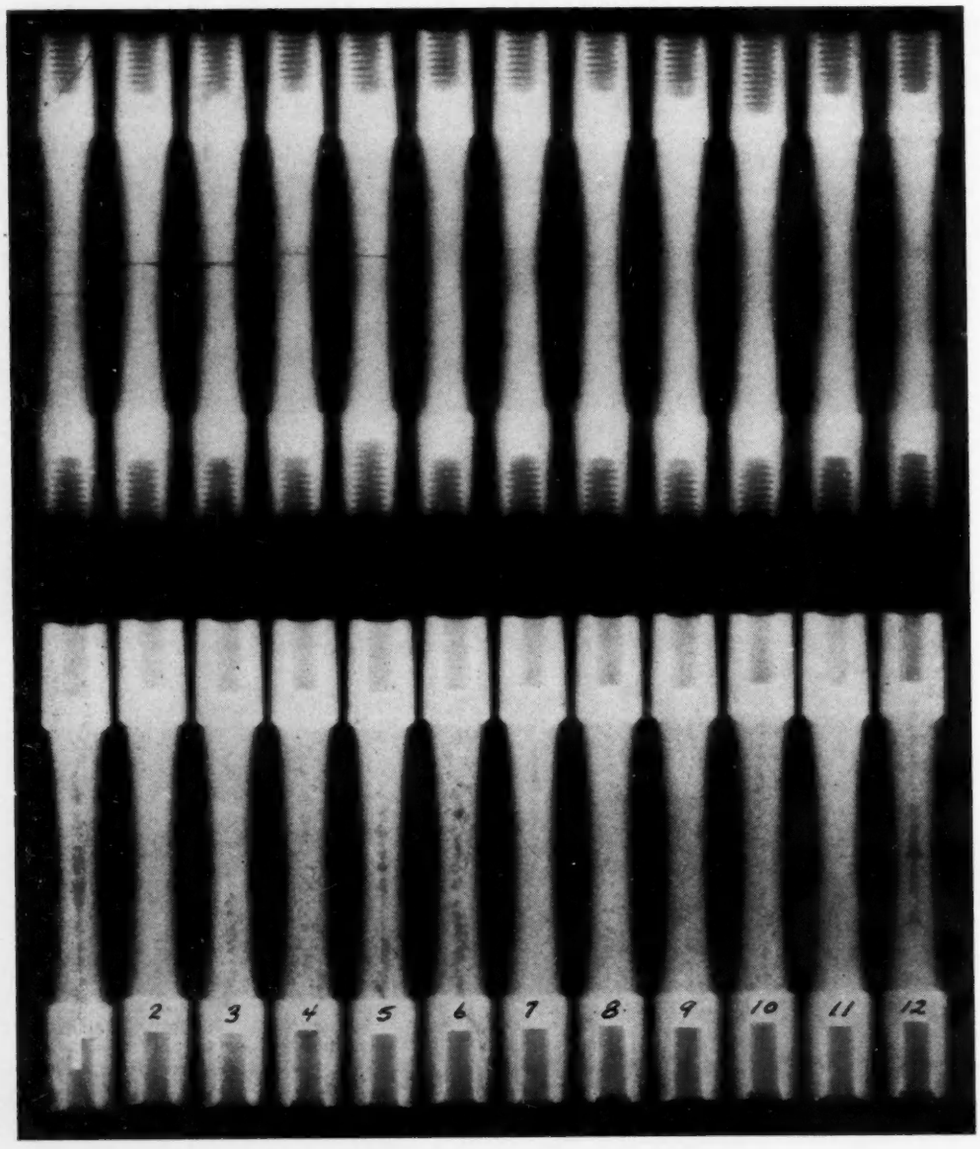


Fig. 13 – Radiograph at the Bottom Is of 8 Per Cent Copper, 3 Per Cent Silicon Die-Cast Fatigue Test Bars – The Internal Discontinuities Are Revealed – Radiograph at the Top Is of Sand-Cast Fatigue Test Specimens of a Similar Alloy – The Sound Sand-Cast Bars Happened to Have Been Exposed After Testing, So the Fractures Are Evident

The Traction of Pneumatic Tractor Tires

By the S.A.E. Cooperative Tractor Tire Testing Committee *

THE cooperative tractor tire tests described in this paper were discussed originally at a meeting of the Society several years ago. The tractor engineers present at the discussion suggested to the tire engineers that there was need for a cooperative test program to determine the efficiency of the various tire sizes over a range of soil conditions.

Among the ten conclusions drawn from the comprehensive tests reported in this paper are that the most important factor affecting the coefficient of traction or tire thrust of rubber-tired tractors is the nature or surface of the operating soil; that, for a given soil, the most important factor is the weight that the tire carries; and that inflation pressure has a relatively small effect.

IT is perhaps best at the very outset to discuss why the tests herein described were considered necessary.

At the time this program was first conceived (the fall of 1935) the tractor industry had manufactured just enough rubber-tired tractors to discover a lot of questions that needed answering before they could make the best use of the rubber tire. At that time the majority of rubber-tired tractors were built primarily for steel wheels and the rubber tires were merely optional equipment. Some pioneering had been done on tractors designed for rubber tires exclusively, but the designers were faced with a difficult problem in choosing the correct tire size. There was a large range of sizes available, but there was no information as to their relative tractive ability as related to the static load ratings experimentally adopted by the Tire and Rim Association. This was a serious problem, and was confused further by conflicting reports from the field, and by the inconsistency of data supplied by various tire manufacturers as to the tractive ability of different tire sizes.

History of Test Program. – The tests covered in this article originally were discussed informally at the December, 1935, Annual Meeting of the Society. The tractor engineers present at the discussion suggested to the tire engineers that there was need for a cooperative test program to determine the efficiency of the various tire sizes over a range of soil conditions. It was felt that, with the tire companies working independently, a mass of unrelated data would ensue which would result only in confusing the tractor engineers. It was suggested that a cooperative program would produce results permitting the comparison of tire sizes for each soil condition, as well as the comparison of different soil conditions.

As a result of this discussion, A. W. Lavers, then Chairman of the S.A.E. Tractor and Industrial Power Equipment Activity Committee appointed a subcommittee to consider the problem and arrange for the tests if possible. This subcommittee consisted of Prof. C. W. Smith of the University of Nebraska, chairman; E. F. Brunner, representing the rubber industry; and O. E. Eggen representing the tractor industry. This subcommittee turned the problem over to the rubber companies as it was felt that they should conduct the tests, subject to the approval of the tractor engineers.

Several meetings of representatives of the four major rubber companies were held in Akron, and details worked out for submission to the tractor engineers. At these meetings discussed and decided upon were the sizes to be tested, the tread design to be used, and similar details. The tractor manufacturers then met in Waterloo, Ia., and drew up a program covering desired soil conditions, and the general type of results wanted, and also arranged to supply the necessary tractor. Following this meeting a general meeting of tractor and tire engineers was held in Lincoln, Neb., on June 29, at which time a final program was drafted and the entire procedure was discussed.

A trip to Auburn, Ala., also was suggested at this time to investigate the possibility of conducting the tests in the soil plots at the farm tillage machinery laboratory of the U. S. Department of Agriculture. It was discovered that the tests could not be made in Auburn for some time as the necessary machinery and instruments had not been designed. At that time (July 20, 1936) the actual test preparations were started.

Proposed Test Program. – As a result of these meetings it was stated that the object of the tests should be to determine the ratio between drawbar pull and tire load with various combinations of tire sections and diameters in several soil conditions.

The final program adopted called for testing eleven sizes of tires on five soil conditions. The soil conditions chosen were described as follows:

(1) Gravel road, hard dry, with traces of loose gravel on it.

(2) Sand – dry – natural moisture, covered with oat or

[This paper was presented at the Regional Tractor Meeting of the Society, Akron, O., Sept. 17, 1937.]

* This committee consisted of one representative from each of the following companies: Firestone Tire & Rubber Co., B. F. Goodrich Co., Goodyear Tire & Rubber Co., U. S. Rubber Products, Inc. The paper was presented by R. P. Gaylord, Goodyear Tire & Rubber Co.

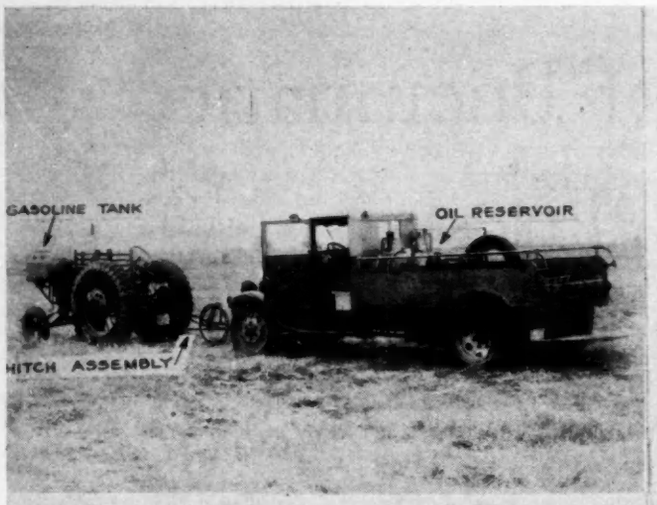


Fig. 1 – General View of Test Equipment

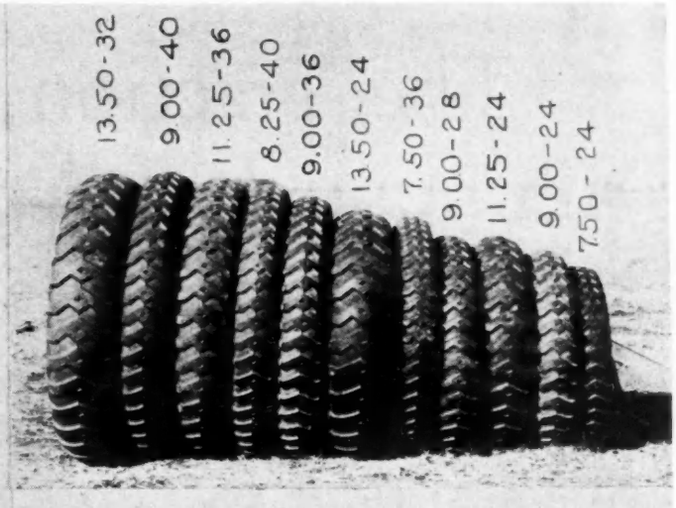


Fig. 2 – Set of Tires Tested

grain stubble. Sand to be similar to soil found in western Kansas and the Panhandle in Texas.

(3) Firm dry soil with vegetation such as alfalfa or any green cover crop. Crop to be at least 6 in. high. Cover crop must be green.

(4) Firm dry soil with stubble vegetation and the natural conditions found approximately 30 days after harvest time (moisture content to be at minimum).

(5) Wet clay or gumbo.

It also was agreed that a speed of 2 m.p.h. should be used for all tests, as close as available gears would permit.

Actual Test Program. – As the tests were conducted finally the program was carried out substantially as just outlined with the following changes:

(1) Gravel road – as specified.

(2) Sand – as specified – loose and flowing – no vegetation.

(3) Cover crop – alfalfa chosen – average height approximately 24 in. – yield about 1¼ tons per acre.

(4) Firm dry soil – heavy clay chosen as being an opposite extreme to the sand, thus covering the widest range of soil types.

(5) Wet clay or gumbo – this item was finally dropped due to inconsistency of wet soil which would give anything from 0 to 2000 lb. drawbar pull in same field.

(6) Sandy loam – this soil was added in Sterling, Ill., as a demonstration of how tests would be run and what results could be obtained.

(7) Concrete road – this surface also was added in Sterling as being of interest for industrial tractor design.

The first four items listed were run in the vicinity of Phoenix, Ariz., the last two in the vicinity of Sterling, Ill.

After drafting the program it became necessary to consider the problem of apparatus for testing.

Apparatus

As finally evolved the testing apparatus consisted of a modified tractor and truck coupled through an integrating dynamometer, and specially designed equipment for measuring travel reduction (Fig. 1).

Tires. – Eleven sizes of tires covering a wide range of sections and diameters, having a tread pattern of neutral design and of conventional non-skid depth (1936) and dimensions, were used in the tests (Fig. 2). Their complete characteristics will be found in the table of Fig. 3. The tires were mounted on standard recommended rims, the wheels being fabricated to special order by French & Hecht to their usual design.

Tractor. – A Minneapolis-Moline Model "JTS" tractor was used in these tests and altered specially to provide room for applying load (Fig. 4). The standard equipment of fenders and platform was dispensed with to attain a minimum load. The gasoline tank was relocated on the left side of the engine over the front axle, and a cradle was built in the usual gasoline tank position for carrying lead pigs. Wheel weights also were used when necessary to attain the required load. The standard cylinder-head (compression ratio 4.5:1) was replaced by a high-compression head with a compression ratio of 5.4:1, and a type of gasoline that would not detonate under high compression was used.

The level of the tractor was maintained by using two front-tire and wheel assemblies with auxiliary spacers between the front axle and the bolster, and the reach was adjusted suitably for each front level adjustment. The front tire sizes used were 4.00-8 and 6.00-16. The standard hitch was replaced by a new drawbar hitch designed to permit a vertical adjustment so as to maintain approximately uniform hitch heights (14 in.) for all tire sizes involved in the test.

Dynamometer Truck. – A 1929 model, 2½-ton, White truck equipped with 7.50-20 dual rear tires was used as the drawbar loading vehicle. The drawbar load was applied by a gear oil pump driven by the rear wheels of the truck through the power take-off and suitable gears. The load thus produced

TIRE SIZE	PLIES	OUT-SIDE DIAM. IN.	CON-TACT AREA SQ. IN.	RATED ROLLG RADIUS IN.	WHEEL & TIRE WEIGHT LBS.	GEAR USED	TIRE & RIM ASSOCIATION LOAD RATINGS - LBS. EXPERIMENTAL PRACTICE				
							8	10	12	14	16
7.50-24	4	40.1	50	18.7	191	3	760	870	960	1060	1140
7.50-36	6	52.1	56	26.7	266	1	910	1050	1170	1300	1410
8.25-40	6	57.6	72	27.1	303	1	1150	1320	1470	1640	1770
9.00-24	6	43.6	72	19.9	218	3	1070	1220	1360	1490	1610
9.00-28	6	47.6	76	21.9	248	2	1140	1300	1450	1590	1720
9.00-36	6	55.6	84	25.9	292	1	1280	1460	1630	1790	1940
9.00-40	6	59.6	88	27.9	322	1	1350	1540	1720	1890	2050
11.25-24	6	47.4	102	21.6	260	2	1410	1610	1790	1960	2120
11.25-36	6	59.4	120	27.6	336	1	1710	1940	2150	2350	2540
13.50-24	6	52.4	162	23.3	307	2	2080	2380	2650	2900	3140
13.50-32	6	60.4	180	27.3	367	1	2400	2710	3000	3270	3530

Fig. 3 – Tire and Wheel Characteristics

was regulated by controlling the speed and discharge pressure of the pump. The rear end of the truck was ballasted with lead and cast iron to provide the required braking without sliding the tires. A Northern Rotary herringbone-gear oil pump, 4600 Series, having a rating of 20.1 hp. at 1750 r.p.m. and 1000 lb. per sq. in. discharge pressure and an auxiliary eight-speed truck transmission were mounted on the frame of the truck and coupled to the two-speed power take-off. The pump was driven by the rear wheels of the truck through the standard three-speed truck transmission, power take-off, and auxiliary transmission, which provided a wide range of pump speeds. The discharge of the pump was carried forward to the truck cab and there through parallel lines containing ¼-in., ½-in. and 1½-in. valves respectively (Fig. 5). These valves were used for controlling the discharge pressure of the pump. A pressure gage in the line on the pressure side of the valves indicated the discharge pressure of the pump. The return line fed the oil back to a 50-gal. steel drum supply tank mounted in the rear of the truck, which tank also served as a means of cooling the oil. S.A.E. 30 motor oil was used in the system.

The drawbar pull of the tractor was carried to the truck through a rigid assembly with a double clevis arrangement at each end. A dynamometer was linked in the drawbar in such a manner that it was supported by the drawbar when no pull was exerted (Fig. 6). The assembly was attached to the hitch on the tractor and to one arm of a bell crank attached to the front of the truck.

Integrating Dynamometer. – The integrating coil-spring dynamometer used in these tests was borrowed from Iowa State College and was used for determining the actual average drawbar pulls exerted by the tractor. This dynamometer was calibrated when obtained and was rechecked periodically during the tests. The dynamometer consists of a moving unit and a fixed unit attached to opposite ends of a coil spring which is compressed under load through a parallel linkage. The fixed unit consists of a steel frame upon which is mounted a shaft driving a carbon disc at nearly constant speed. The movable unit consists of a circular revolution counter and vernier driven by a roller which moves across the face of the carbon disc, its shaft axis being parallel to the carbon disc. As the load is applied to the spring the roller moves from the center of the carbon disc to the outer circumference and is thereby driven at increasing speed relative to

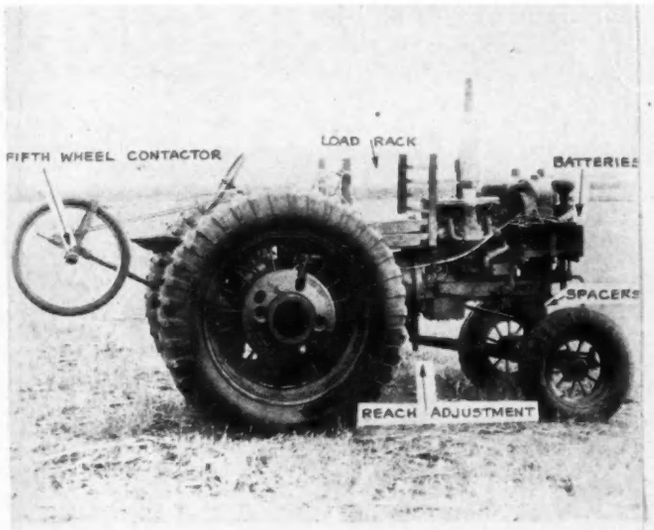


Fig. 4 – Side View of Test Tractor

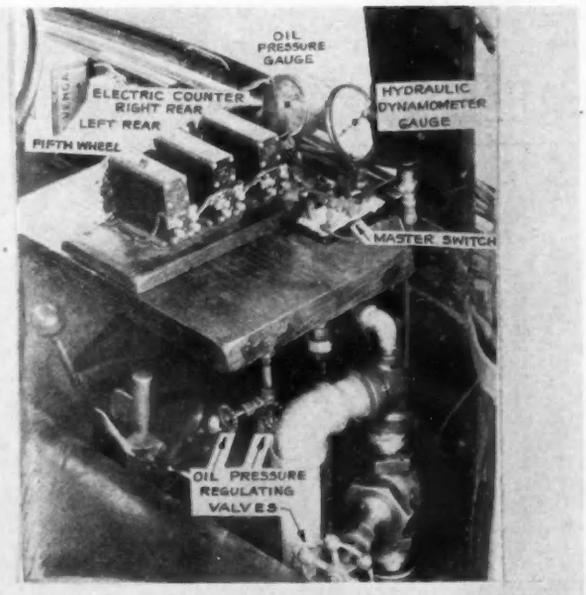


Fig. 5 – View of Truck Cab

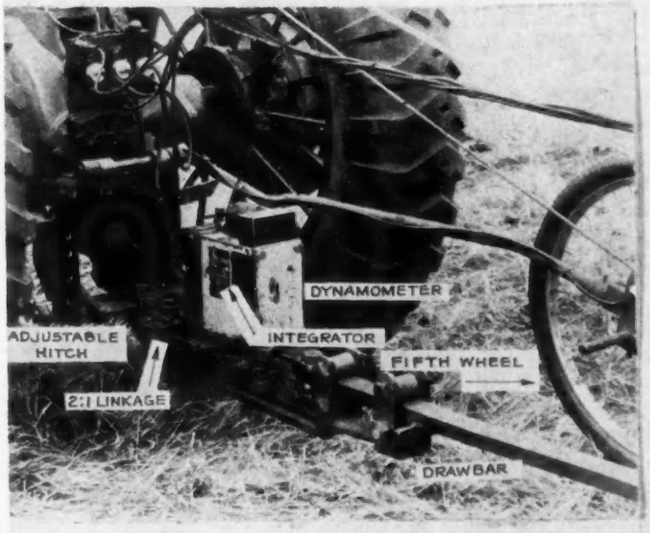


Fig. 6 – Tractor Hitch and Dynamometer

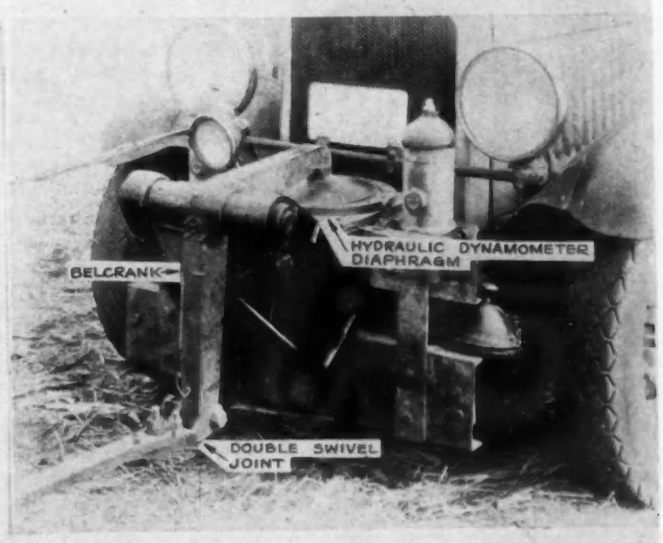


Fig. 7 – Truck Hitch and Dynamometer

tne speed of the carbon disc. The carbon disc was driven by a six-volt direct-current motor equipped with a revolution counter. The ratio of the two revolution-counter readings was a measure of the average pull exerted. Upon final evaluation of the tests it was found that there was some inconsistency of the instrument in the higher load ranges. Results in such cases have been marked as subject to possible question.

Hydraulic Dynamometer. – A hydraulic diaphragm dynamometer was mounted at the front of the truck and actuated by the horizontal arm of a bell-crank assembly, the vertical arm of which was attached to the drawbar (Fig. 7). The bell-crank arms were of exactly equal length and the assembly so mounted that a hitch height of 14 in. was maintained, thus keeping the drawbar substantially level at all times. The drawbar pull was transmitted through a steel plate of 108 sq. in. to the hydraulic system and the pressure built up was indicated on a pressure gage with 2 lb. per sq. in. graduations mounted in the cab of the truck. The liquid in the hydraulic system was maintained at a suitable level by means of a reservoir situated at the highest point of the system. The hydraulic dynamometer was used by the operator as an indicator, the integrating dynamometer furnishing the exact average pull.

Travel Reduction Measurement. – Automobile distributors with four contacts each were attached to each brake drum of the tractor and connected in an electrical circuit with magnetic counters to count the drive-wheel revolutions. The 4.71:1 gear ratio of brake drums to drive wheels provided

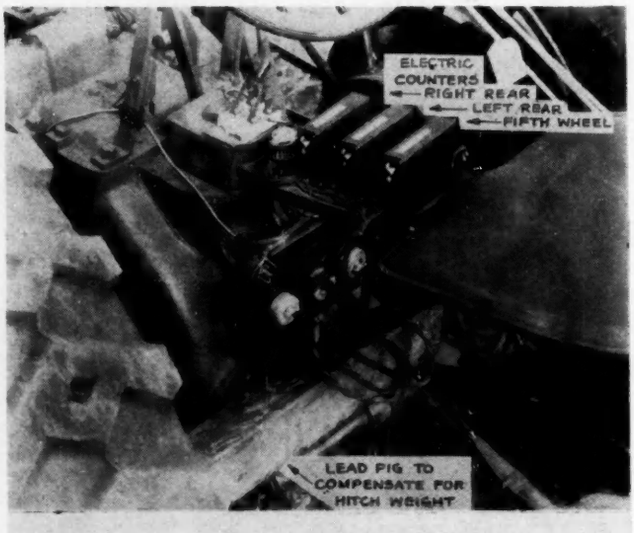


Fig. 8 – Electric Counters Mounted on Tractor

means of measuring the drive-wheel revolutions to approximately one-nineteenth of a revolution. A fifth or bicycle wheel was attached to the rear of the tractor by a swivel arm and located so as to track the right-rear drive wheel of the tractor. A four-contact distributor on the fifth wheel similarly connected made it possible to measure actual distance to one-fourth of a revolution of the 26-2.125 bicycle wheel used. After the first tests the four-lobe cam was replaced by a ten-lobe cam to increase the accuracy.

Magnetic counters were mounted on an instrument board which could be placed in the electrical circuit when mounted either in the truck cab or on the horizontal seat support of the tractor (Fig. 8). Current for the circuit was obtained from two six-volt storage batteries mounted on the right side of the tractor engine. The batteries were connected in series to provide twelve volts and were so wired that they could be charged separately by a six-volt generator installed on the tractor engine. An ammeter also was included in the circuit to indicate the charging rate of the batteries. The dynamometer motor also was connected in the twelve-volt circuit with a suitable variable resistance to control its speed. The entire electric system was so connected that all counters as well as the dynamometer were started and stopped simultaneously by a single switch.

Soil Samples. – Soil samples for moisture-content determinations were extracted with a 1½-in. soil auger borrowed from the University of Illinois, and were carried in 1 by 5 in. aluminum evaporating dishes. The empty weight of the dishes was determined and tabulated when they were first used, and these weights were checked periodically. The samples were weighed on a balance to the nearest 0.1 gm. An electrical thermostatically controlled (100-110 deg. cent.) drying oven was used for drying the samples.

The penetration of the tires into the soil was measured where possible with an improvised depth gage set flush with the undisturbed field surface.

The tire air pressures were measured with a low-pressure gage graduated to 1 lb. per sq. in., and the calibration was checked periodically with a master test gage. A stop watch graduated in one-hundredths of a minute was used for measuring the time of test runs.

The tests were conducted in accordance with the prearranged program of loads and inflations shown in the table of Fig. 9.

Procedure

Adjustment of Load. – The procedure used for making one test run (one load and inflation step), in general, consisted of adjusting the tire load and inflation to the requirements of the program, making a "zero run" in which the tractor was operated without drawbar and truck attached, then making a series of runs under various drawbar pulls measuring speed and travel reduction.

Lead pigs were placed in the cradle of the tractor to load the rear axle as required. For the larger loads standard French & Hecht wheel weights were attached to the wheels with the usual bolts. With the load set the inflation pressure was adjusted to the desired value, and the hitch height was set to the proper point.

Zero Run. – For a zero run the electric counters were mounted on the tractor and hooked into the electrical circuit by means of conveniently located plugs. A 100-lb. lead pig was placed on the tractor hitch to compensate for the weight of the detached drawbar and integrating dynamometer (Fig. 8). The fifth wheel was placed in position, and the tractor was started in the proper gear to give the required 2 m.p.h. with the throttle opened wide. After the tractor attained its full speed, the counters and stop watch were started simultaneously. They were stopped similarly after traversing the desired distance (250-300 ft.), and the tractor was brought to a stop. Readings on the stop watch and counters were checked by a second observer after recording on the field data sheet.

Drawbar-Load Runs. – The drawbar-load runs were made with the truck coupled to the tractor by the drawbar assembly including the integrating dynamometer. The electric counters were mounted in the cab of the truck and plugged into the electrical circuit. The inflation pressure was checked and maintained within ½ lb. per sq. in. during all runs. The revolution counters of the integrating dynamometer were all set at zero before starting each run. The tractor was then started in the proper gear and driven with wide-open throttle. The truck was towed in gear with the ignition switch off and the power take-off engaged, the truck motor thus being oper-

RIM DIAM	SECTION DIAMETER														
	7.50			8.25			9.00			11.25			13.50		
	8	12	16	8	12	16	8	12	16	8	12	16	8	12	16
24	1260	1160 1260 1360 1460	1260					1190 1350 1520 1680		1390 1440 1890 2140	1390 1640 1890 2140	1390 1640 1890 2140	2430	2060 2430 2790 3120	2430
28								1220 1420 1610 1810							
32														2370 2730 3100 3490	
36	1400	1240 1400 1530 1700	1400				1240 1490 1720 1950	1280 1490 1720 1950	1260 1490 1720 1950	1910	1680 1910 2210 2610	1910			
40					1270 1440 1600 1770			1360 1590 1820 2050							

Fig. 9 - Program of Test Loads

ated against compression. The speed of the oil pump was pre-selected by choosing gears to give approximately the range of resistance required. The valves in the oil-pump system were then adjusted to provide resistance such that the desired drawbar pull was indicated by the gage of the hydraulic diaphragm dynamometer. When the valves were set, the master switch starting the counters and dynamometer was closed coincident with the start of the stop watch. On completion of the run, the watch was stopped simultaneously with the opening of the master switch. The tractor was then stopped, the dynamometer readings taken, and the counters and stop watch read. All readings were made by two observers. The counters and dynamometer were then reset to zero for the next run.

Determinations also were made in each series for the maximum drawbar pull, increasing the drawbar load until the tractor's forward advance was practically zero. The average of 4 or 6 readings was taken directly from the gage of the hydraulic diaphragm dynamometer for the maximum pull value.

The tire penetrations in many of the runs were measured as the depth from the surface of the undisturbed soil to the plane at which the soil sheared. This measurement was done by laying an improvised depth gage flush with the undisturbed soil surface and measuring, with an adjustable scale, the depth of the tire track with the loose soil removed.

Soil Samples. - The soil samples for moisture determinations in the two light soils were taken from three levels of soil: 2 in., 4 in. and 6 in., the soil auger flange being adjusted so that a 2 in. height of sample was removed. Samples were taken from portions of the field on which tests were made that particular day. The 2-in. sample was taken from the surface of the soil to a 2-in. depth. The soil was cleared away carefully to the 2-in. level and another sample taken from the 2- to 4-in. depth. Similarly a sample was taken from a 4- to 6-in. depth. The samples were placed immediately in aluminum evaporating dishes with snug-fitting covers on removal from the soil. On the heavy dry clay soil where no penetration occurred, the samples were taken only for the top 1 in.

Each dish containing a sample was weighed to the nearest 0.1 gm. and then placed in an oven maintaining a temperature of 100 to 110 deg. cent. for 16 to 24 hr. The dishes were weighed with the soil sample on removal from the oven, thus measuring the amount of moisture removed.

Evaluation

General. - The limits of the evaluation were kept within the experimental accuracy involved, and no results are given to an accuracy greater than that involved in the original data.

All calculations were made with slide rules and have been checked independently by a second observer.

The unknown discrepancies introduced by varying soil conditions are believed to produce errors far greater than those to which the evaluation was subject.

In the following description of the evaluation, a statement is made at each step giving the limits to which the calculation was carried and the possible error involved.

The original data were transferred to a 30-column work sheet and all calculations carried out and tabulated on these forms.

Tire Load. - By means of an accurate recently checked platform scale, it was found that the tractor with no weights, tires or wheels, but with driver, drawbar, and integrating dynamometer attached, weighed 1940 lb. on the rear axle or 970 lb. per rear tire. The weight distribution of lead pigs placed in the load rack of the tractor was determined for the rear axle by weighing. The weights of all the wheels with tires and tubes also were determined similarly.

The static load with any particular tire size and given number of lead pigs was determined by adding to the wheel, tire, and tube weight the proper proportion (29.6 per cent Sterling and 32.4 per cent in Phoenix) of the lead weight, and all of the wheel weights, to the base weight of 970 lb. per rear wheel.

The minimum weight of the tractor was such that five tire sizes could not be tested at their minimum rated loads. These tires were the following:

Tire	Minimum Test Load, lb.	Rated Load Range: Minimum 8 lb. per sq. in.	Rated Load Range: Maximum 16 lb. per sq. in.
7.50-24	1160	760	1140
9.00-24	1190	1070	1610
9.00-28	1220	1140	1720
7.50-36	1240	910	1410
8.25-40	1270	1150	1770

Drawbar Pull. - The calibration curve of the integrating dynamometer for a linkage ratio 1:2 which was used throughout the tests, is a straight line up to approximately 3500 lb. as shown in Fig. 10.

The calibration is based on 50 revolutions of the carbon disc which drives the graduated recording disc. Inasmuch as the number of carbon-disc revolutions in actual tests varies according to circumstances, a correction back to the basic 50 revolutions can be included in an equation which would cover the straight-line range. The counter is arranged so that each

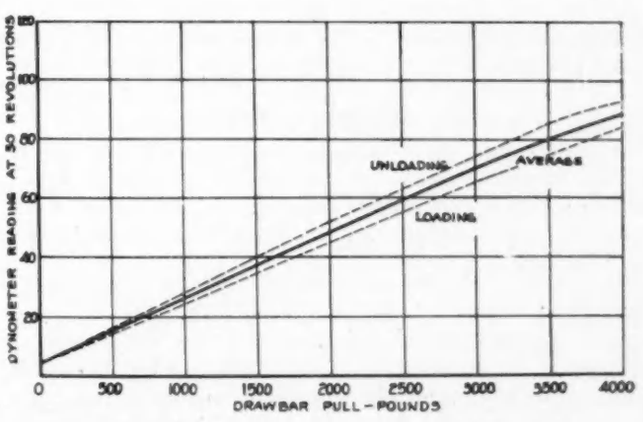


Fig. 10 - Calibration Curve of Dynamometer

fifth of a revolution is counted, and the calibration equation included this factor.

In the range above 3500 lb. all recorder readings were corrected to their equivalent at 50 revolutions of the counter shaft, and the pull was taken directly from the calibration curve. Some inconsistency appears in the instrument above 3500 lb. and results showing pulls above that amount are in question to the extent of a few hundred pounds. All evaluations of drawbar pull were to the nearest 10 lb. The maximum error due to limits of reading and calculation was 18 lb.

Distance, Horsepower, and Speed. – The fifth wheel used for measuring distance was calibrated by counting the number of revolutions it made in running over a 1000-ft. measured course. It was calculated from this calibration that the rolling circumference of the fifth wheel was 6.80 ft. Having four contacts in the distributor, the counter registers each quarter revolution or 1.70 ft. whereas, with 10 contacts, the counter registered each 0.68 ft. The distance traveled by the fifth wheel, in feet, is therefore equal to the counter reading times 1.70 for the Sterling tests or multiplied by 0.68 for Phoenix tests.

The calculation for distance was carried to the nearest foot. The maximum possible error involved was 2.2 ft. in Sterling tests which amounts, on the average, to about 1 per cent. With the 10-contact cam used in Phoenix, the maximum error was approximately 0.9 ft. or 0.4 per cent.

The tractor speed is presented in feet per minute and also in miles per hour. The speed in feet per minute was calculated by dividing the distance covered in feet by the time required in minutes. The speed in feet per minute was converted to miles per hour by dividing by 88. The speed was calculated to the nearest foot per minute and nearest 0.01 m.p.h. The maximum possible error involved is approximately 2 ft. per min., or about 1 per cent.

The product of the measured drawbar pull exerted by the tractor and its speed in feet per minute divided by 33,000 equals the drawbar horsepower. The calculation was carried to the nearest 0.1 hp. The maximum error possible would be approximately 0.2 hp.

Advance and Travel Reduction. – The advance of the tractor with reference to the ground was expressed in terms of advance per drive-wheel revolution. These values were obtained as the quotient of the distance traveled and the number of drive-wheel revolutions. Inasmuch as the drive-wheel revolutions were calculated from the revolutions of the brake drums, the actual counter readings were fractional parts of the drive-wheel revolutions. The ratio of brake-drum revolution-counter reading to drive-wheel revolutions was 18.84:1. 4.71:1 was the gear ratio and the distributor had 4 contacts, hence 4 times 4.71 equals 18.84. Thus the counter reading was 18.84 for each drive-wheel revolution, and the advance per revolution was calculated by dividing the distance covered in feet by the counter reading (C.R.) over 18.84.

$$\frac{\text{Distance}}{\frac{\text{C.R.}}{18.84}} = \frac{\text{Distance} \times 18.84}{\text{C.R.}} = \text{Advance.}$$

The advance was calculated to the nearest 0.01 ft. per revolution. The maximum possible error involved was approximately 0.2 ft. per revolution or about 1.5 per cent.

The apparent travel reduction was calculated from the differences in advance per revolution with and without drawbar load on the tractor, the advance at no drawbar load being used as the basic value or zero for computing travel reduction. The question of what is a true zero is complicated by several

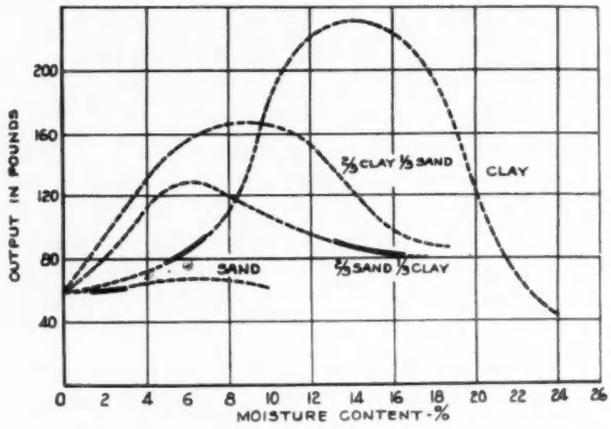


Fig. 11 – Effect of Moisture on Soil Strength

factors, notably the creep of tires on hard surfaces and the slip due to rolling resistance of the tractor alone. The method used is considered a good compromise. The advance decreased as the travel reduction increased. The difference in advance divided by the basic advance and multiplied by 100 represents the per cent of apparent travel reduction. For convenience and accuracy, counter readings instead of actual wheel revolutions were used in computing the ratios. This method did not affect the results as the difference in numerical values is due only to constant factors.

The apparent travel reduction was calculated to the nearest 0.1 per cent. The maximum error involved was 0.9 per cent travel reduction with four-contact fifth wheel or 0.4 per cent with ten contacts per revolution.

Hitch Height and Weight Transfer. – Due to the difficulty of obtaining any reliable measurements of hitch heights on the rough field surface, all hitch heights were calculated from physical tire measurements and measured distances of the hitch below the tractor rear axle.

The outer diameter and deflection characteristics of the tires were obtained from previous measurements on similar tires. The hitch height was calculated as the difference between the rolling radius of the tire at static load and the known distance in inches of the hitch bolt from the axle centerline of the rear axle. Actually the hitch height decreases as the weight transfer is added, but calculation showed the increases in weight transfer thus obtained were within the 10 lb. limit used on weight calculations.

The weight transfer due to drawbar pull on the hitch of the tractor was calculated by considering moments about the center point of contact of the rear tire and ground. The moment of hitch height in inches times drawbar pull in pounds was balanced by the moment of tractor wheel base in inches times load transfer from front to rear wheels in pounds.

Rolling Resistance. – On the soft sand soil the rolling resistance of the tractor alone was high enough to produce weight transfers of appreciable amounts. Hence rolling-resistance runs were made, and a correction applied to the true tire load for the weight transferred due to this item.

The weight transfers were calculated to the nearest 10 lb. The maximum possible error would be about 10 lb. transfer, or 5 per cent.

The true tire load was computed as the static load of the tire plus one-half the weight transferred to the rear axle. The maximum error in the true tire load would be 10 lb.

All tire deflections were taken from the load-deflection curves at the true tire loads. The true rolling radii values were computed as the difference between radius of the tire at

zero load and the tire deflection under the true tire load. All deflections and rolling-radii data were computed to the nearest 0.01 in.

Correction of Travel Reduction. – Due to the weight transferred to the rear axle of the tractor under drawbar load, it was considered advisable to correct the travel reduction percentage in the amount that the rolling radius affected the travel reduction. This correction will be proportional to the change in rolling radii or if

R = rolling radius under static load.
r = rolling radius under true tire load.

$\frac{r}{R}$ = correction factor to be applied

True travel reduction = apparent travel reduction times $\frac{r}{R}$ or

True travel reduction = apparent travel reduction $- \frac{R - r}{R}$

The quantity $\frac{R - r}{R}$ multiplied by 100 is the percentage value subtracted from the apparent travel reduction to arrive at the true travel reduction, ignoring creep and rolling resistance at no drawbar pull. The correction was carried to the nearest 0.1 per cent of travel reduction.

Soil Moisture Content. – The moisture content of a soil is by definition the percentage of moisture contained in the soil above its dry weight. All the moisture contents presented in these reports are calculated on this basis. The three moisture figures listed in the tables are the three levels reading from the surface down.

Soils

General. – Before proceeding to the results it perhaps would be well to discuss the problem of soils as they affect the tractive ability of tires.

It is known generally among those working in agricultural fields that soils are classified in a number of ways, as light or heavy, as sand or clay, damp or dry, and so on. The older and more established way of classifying a soil is by determining the proportions of the different sized particles that it contains. Thus the coarser particles are classed as gravel, the next finer class being coarse sand followed, in order, by fine sand, silt, fine silt, and clay. Each of these classifications was defined by size, the fine gravel being from 0.12 to 0.04 in. in diameter at one extreme, and the clay being all particles below approximately 0.0001 in. in diameter. Knowing this analysis one can then class a soil as sand, loam, silt, clay, or any combination thereof depending upon the relative proportions of the various items. Those soils containing predominantly the sandy constituents are said to be light, whereas those containing largely silt and clay are classed as heavy soils.

Soil Type and Moisture Content. – However, such a classification does not define the physical resistance, or shearing strength, of the soil although it can be stated readily that, at proportionate moisture contents, a heavier soil is more resistant than a light soil. Some work has been done to show that the physical resistance of the soil is a function of the per cent colloid content and the maturity of the soil as measured by the silica-sesqui-oxide ratio. Other factors which vitally affect the resistance of the soil are its density, or apparent specific gravity, and its moisture content. Fig. 11 shows some data presented by J. W. Randolph of the U. S. Department of Agriculture at the A.S.A.E. meeting in Chicago last December in his paper entitled "Testing of Wheels and Lugs," that aptly illustrate these points.

The heavy portions of the curves represent the percentage moisture at which our tests were made. It is worthy of note that all tests were conducted in a region of the curves unfavorable to high soil strength.

The soils used by the Committee in its tests covered a range of conditions which indicated fairly well the relation between the tire and the soil in producing traction.

Dry Sand. – In the tests on dry sand, the soil was in a loose granular condition and flowed readily (Fig. 12). In fact, after the tractor and truck passed over the soil, the tracks readily filled in with sand leaving just a triangular trough to mark the track and no evidence of tread pattern could be seen (Fig. 13). The soil was disced after each run so as to maintain the loose condition for all tests. The traction was produced apparently by a combined shearing of the sand under the compression of the tire load together with a sort of paddling action through the loose soil. These tests were deliberately run on the soil in the condition considered worst from a traction standpoint. In other words, the results obtained should be a minimum for sandy soil.

Sandy Loam. – The tests run on sandy loam soil on the other hand were run as a more average condition (Fig. 14). The soil was somewhat soft, having been plowed about six months before, and stood in wheat stubble. The tire lugs were able to penetrate quite readily and get a good grip on the soil. The soil had a moisture content varying between 11 per cent and 23 per cent which gave it considerable resistance when compressed under the tire load. The traction was obtained by shearing the soil at a plane between the tire's lowest point of contact and a point about one inch lower.

Dry Clay. – The dry-clay tests represent an entirely different set of conditions (Fig. 15). The soil here was packed quite hard and dry with just a slight surface softness. The tires were absolutely unable to penetrate the soil and obtained their traction purely by reason of the friction between tire and surface – just as is the case on a paved road. This condition is probably the worst in which this soil would be found, barring extreme wetness. If it were softer so the tires could penetrate or had a somewhat higher moisture content, it is reasonable to suppose that a higher coefficient could be obtained.

Gravel Road. – At this point it might also be well to discuss the other test surfaces and conditions. The gravel road was an excellent piece of road, quite hard and containing only slight traces of washboard (Fig. 16). It had quite evenly distributed over it rather fine loose gravel in small enough quantities that the base was clearly visible. The traction here was apparently also a case of pure friction until the drawbar pull exceeded a certain amount when the gravel tended to roll under the tires and the travel reduction built up rapidly to 100 per cent with only very slight increase in traction.

Cover Crop. – The cover-crop tests were made on a field of alfalfa that had been growing for two years (Fig. 17). The stand was from 18 to 30 in. high and averaged about 25 in. The yield of the field when cut and baled was about 1¼ tons per acre. Although it was not what could be called a dense growth, it did mash down under the tires so as to provide a mat of vegetation that was quite juicy and did not allow the tire to reach the soil (Fig. 18). The soil was a loam soil – quite dry and hard – and did not crumble under the tires except on long pulls at 100 per cent travel reduction. Traction was obtained here also as pure friction and, when a critical value was exceeded, the tires immediately jumped to 100 per cent travel reduction with a drop of 30 per cent to 50 per cent in drawbar pull.



Fig. 12 – Close-up of Sandy Soil



Fig. 13 – Tracks Left by Tractor and Truck in Sand



Fig. 14 – Close-up of Sandy Loam Soil



Fig. 15 – Close-up of Dry Clay Soil

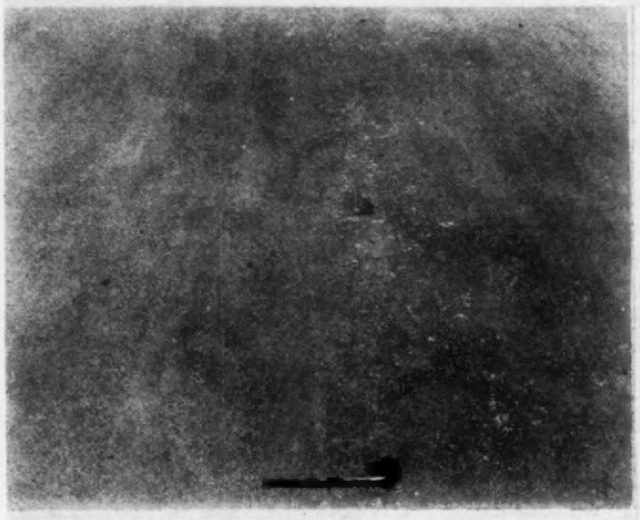


Fig. 16 – Close-up of Gravel Road Surface



Fig. 17 – Alfalfa Crop

Concrete Road. – The concrete road used was a smooth level pavement about two years old having a very light volume of traffic (Fig. 19). The surface was a rough-machine-finish portland cement which provided probably maximum traction. The action was friction to a certain degree, but, at 5 per cent or higher travel reduction, a shearing of the tire treads started as evidenced by the black track left on the pavement.

Miscellaneous tests on speed and the effectiveness of water as weight were conducted on a field of disced loam that was

at a minimum moisture content. Fig. 20 shows the mechanical analysis of test soils.

Results

The results of all of these tests comprise a large amount of data – far too much to present the details here. It is proposed to show typical relationships and present the results as simply as it possibly can be done.

Basic Relationship. – The basic relationship of this entire work is that of drawbar pull and travel reduction (Fig. 21). The curve shown is typical of an average condition. It can be described as consisting of two nearly straight-line sections joined by a curved central portion. The initial straight line extends to approximately 10 per cent travel reduction and covers nearly 60 per cent of the maximum drawbar pull. The final straight line begins at approximately 25 per cent travel reduction and covers the final 10 per cent of the drawbar pull.

All of the data shown in travel-reduction curves can be presented in another and more easily used way if we plot drawbar pull against tire load for any one degree of travel reduction. This idea leads us to the question of which amount of travel reduction should be used as a basis of comparison. There are several points that enter into this question – efficiency, wear, and practicability.

Horsepower. – Fig. 22 shows the relation between travel reduction and the drawbar horsepower developed which is a measure of the efficiency of the tire. A study of these curves especially for the soil conditions shows the maximum horsepower developed between 15 and 20 per cent travel reduction with a very flat section of the curve over that range. We have chosen 16 per cent travel reduction as a standard rating point on soils since it has been used in so many previous investigations and because it is reasonably close to the point of maximum efficiency. However, the two road conditions and the alfalfa present problems that require rating the tires on a basis other than efficiency. Fig. 23 shows the travel-reduction curves for all six test conditions for the same tire, load, and inflation. On these six curves are noted the chosen rating points. As discussed previously, the soils are rated at 16 per cent travel reduction. The other conditions introduce problems of wear and practicability.

For instance, on a concrete road, a tire probably would develop the most drawbar horsepower at around 15 per cent to 20 per cent travel reduction, but the wear of the tire would be prohibitive at such values. In this case we have established 5 per cent as the rating point as the lowest figure available where reasonable comparison is possible. However, 5 per cent is felt to be too high a figure for operating tires on concrete if a reasonable amount of wear is to be expected. This low rating point, of course, leaves a large reserve before the 100 per cent travel reduction point is reached. For similar reasons, the same point of 5 per cent is used for rating the tires on a gravel road although the wear at this point should be satisfactory for this surface. Again on the cover-crop condition at a travel reduction in excess of 10 to 15 per cent the pull becomes unstable and the tire may stall itself on the slippery vegetation. Hence on this condition we must rate the tires at a travel reduction below this critical point, and the values at 8 per cent have been used.

Effect of Weight on Traction. – The effect of weight is shown very graphically in the curve of Fig. 24. Four curves are shown for four different tire loads. Also shown is the 16 per cent rating point and the relative ratings of the four loads thus indicated.

If we now plot these drawbar pulls vs. tire load, Fig. 25, we find a substantially straight-line relationship holding, although the line does not start at the origin. Note that, if we extend this line, it will pass to the right of the zero point and show a negative drawbar pull. This is, of course, the rolling resistance of the tractor in the test range. This fact causes the coefficient to increase with increasing tire load. From this type of curve we can determine for any given tire load what traction was developed by the tires. It should be noted that, in all cases, the tire loads shown include a correction for weight transfer and are not static loads.

In order to illustrate the use of this chart, let us assume a tractor with a rear-axle load of 4000 lb. and having proportions such that the weight transfer is 20 per cent of the drawbar pull. In this case the static tire load is 2000 lb. and, from the chart, we would say the drawbar pull would be around 2200 lb. The weight transfer at this pull would be 440 lb. or 220 lb. per tire, making the true tire load 2220 lb. at which figure the drawbar pull is actually found to be 2300 lb.

Effect of Inflation Pressure. – The effect of inflation pressure is shown for three conditions – first sand, Fig. 26. Note the advantages of lower inflation pressure on this soil. Now consider the dry clay condition, Fig. 27, where the effect of inflation pressure is very small and also the gravel road condition, Fig. 28, where a higher inflation pressure appears



Fig. 18 – Close-up of Alfalfa Mat Left by Tire

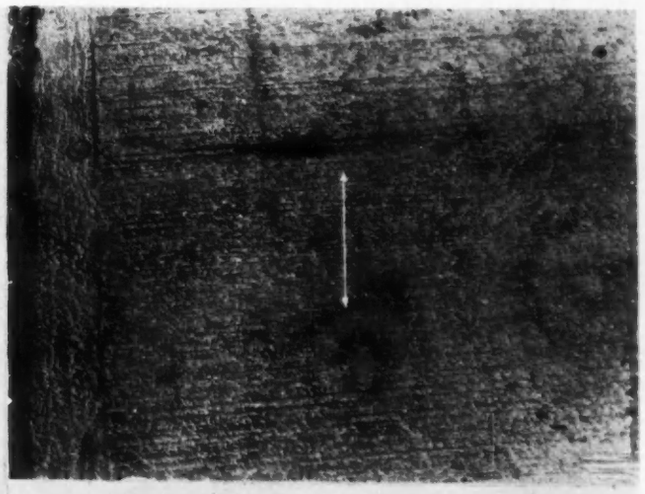


Fig. 19 – Close-up of Concrete Road

advantageous. The table in Fig. 29 shows the effect of inflation pressure on all soils as shown by the traction coefficients.

Traction Coefficients. – In this report the following terms are used in discussing traction:

Tire Thrust (two tires)	$= T$, lb.
Tractor Pull	$= P$, lb.
Rolling Resistance of Tractor	$= R = T - P$, lb.
True Tire Load (including weight transfer)	$= W$, lb.
Coefficient of Tire Thrust	$= \frac{T}{W}$
Coefficient of Traction	$= \frac{P}{W}$

FRACTION	SOIL		
	SANDY LOAM (STERLING)	FINE SAND (PHOENIX)	HEAVY CLAY (PHOENIX)
FINE GRAVEL	1.28	1.26	0.79
COARSE SAND	5.32	4.37	0.75
MEDIUM SAND	23.63	20.01	1.02
FINE SAND	22.76	44.43	4.63
VERY FINE SAND	1.79	10.79	10.06
SILT	25.02	7.64	35.75
CLAY	20.00	11.50	47.00

Fig. 20 – Mechanical Analysis of Test Soils

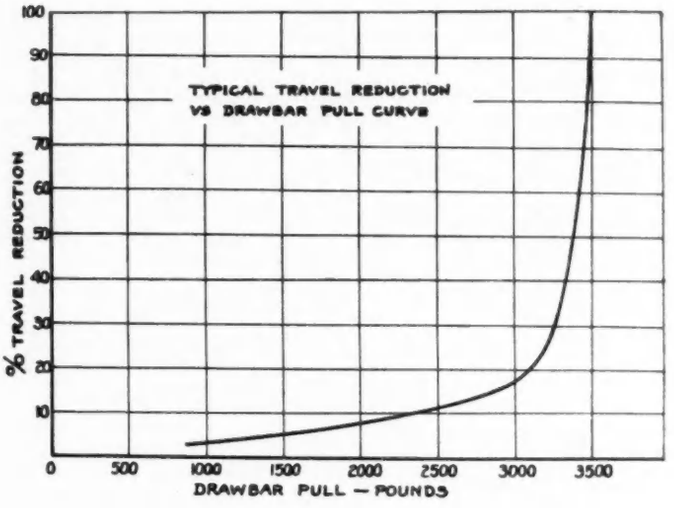


Fig. 21 – Typical Travel Reduction in Drawbar Pull Curve

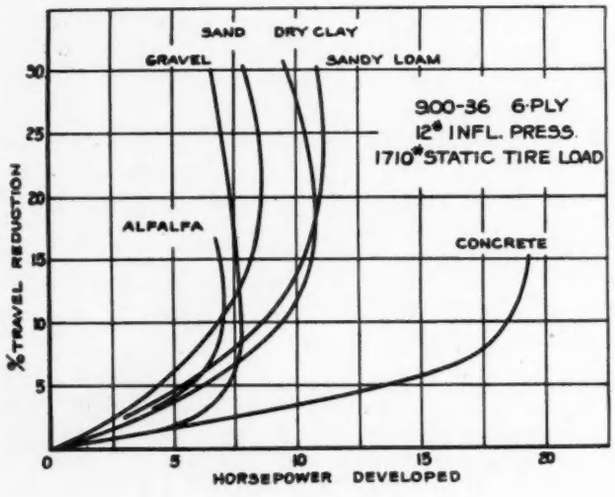


Fig. 22 – Sample Drawbar-Horsepower Curves

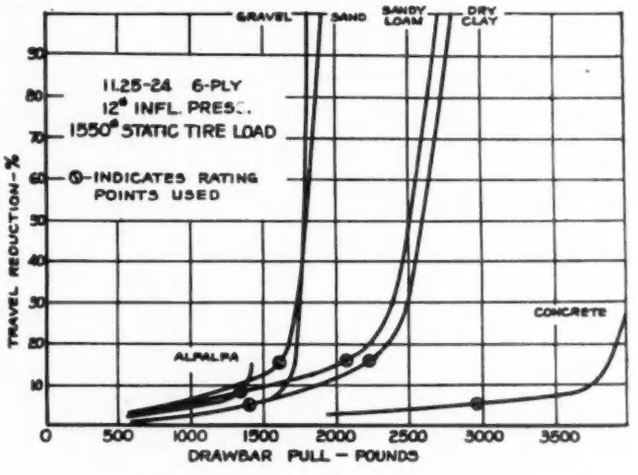


Fig. 23 – Travel-Reduction Curves on Different Test Conditions

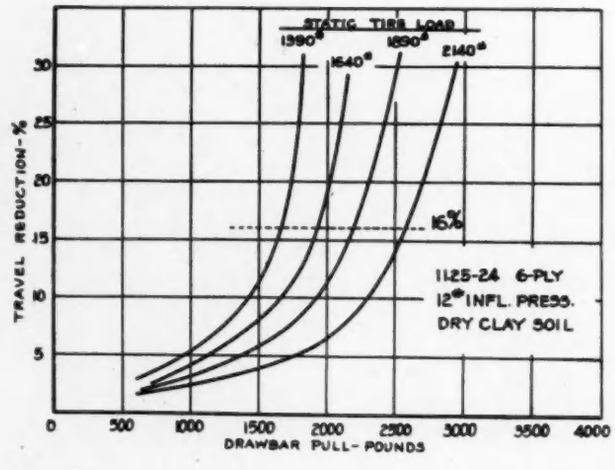


Fig. 24 – Effect of Adding Weight

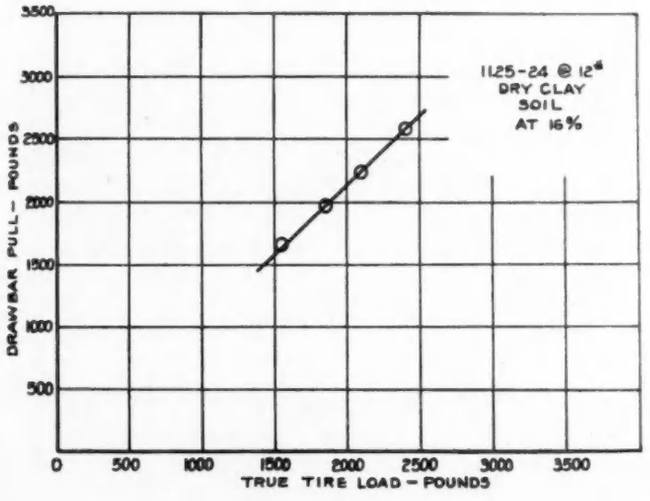


Fig. 25 – Effect of Tire Load on Drawbar Pull

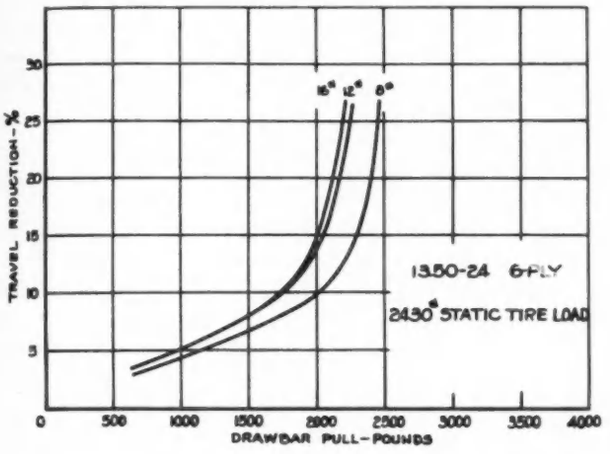


Fig. 26 – Effect of Inflation Pressure on Sandy Soil

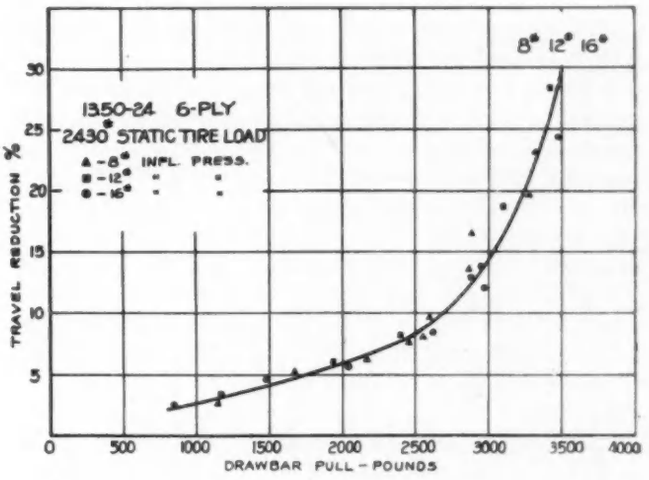


Fig. 27 – Effect of Inflation Pressure on Dry Clay Soil

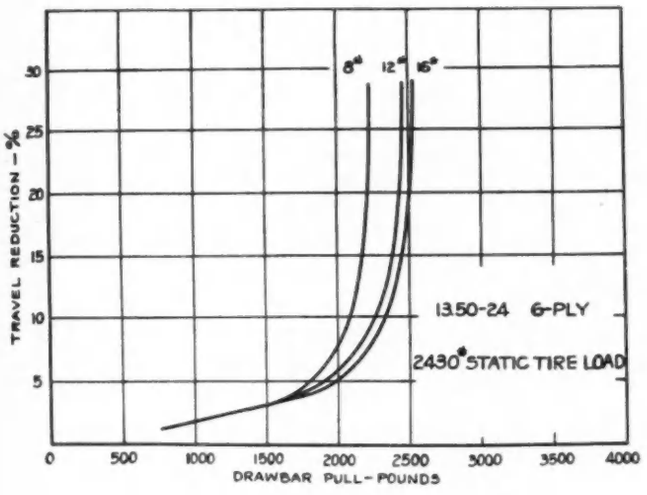


Fig. 28 – Effect of Inflation Pressure on Gravel Road

SOIL CONDITION	AVERAGE TRACTION COEFFICIENT			RATING		
	8#	12#	16#	8#	12#	16#
CONCRETE ROAD	.71	.66	.61	108%	100%	92%
DRY CLAY	.56	.55	.53	102%	100%	96%
SANDY LOAM	.53	.50	.49	106%	100%	98%
DRY FINE SAND	.39	.36	.33	108%	100%	91%
GRAVEL ROAD	.35	.37	.37	95%	100%	100%
ALFALFA	.39	.36	.35	108%	100%	97%

Fig. 29 – Effect of Inflation Pressure on All Test Conditions

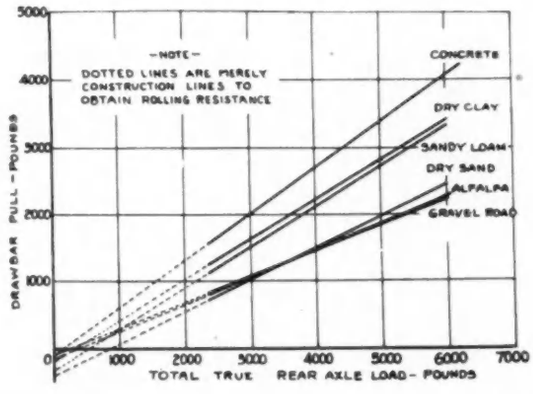


Fig. 30 – Comparison of Drawbar Pull for Different Test Conditions

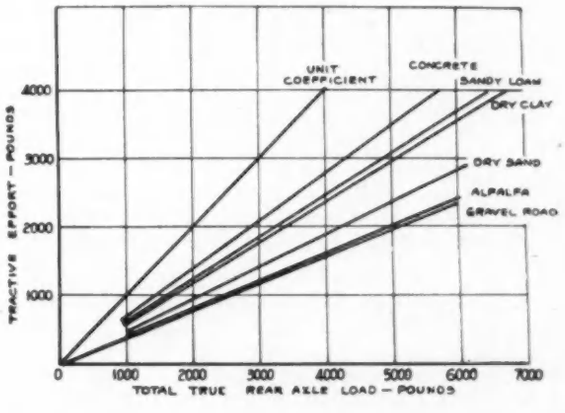


Fig. 31 – Comparison of Tractive Effort for Different Test Conditions

As a convenient figure and one which will not introduce any serious errors it can be stated that, if the traction coefficients are based on static tire loads rather than on true tire loads, they will average 12 per cent higher. The coefficient based on static load is a much easier figure for practical use and the actual errors involved due to differences in hitch, wheel base, and so on, will not be very large. However, in this discussion all traction coefficients are based on true tire load unless otherwise noted.

It should be borne in mind at all times that these traction coefficients are at the arbitrary rating point and are not the maximum available. The maximum traction available for emergency or reserve is on the average 25 to 50 per cent above the figures given at the rating point. This statement does not apply to the alfalfa condition where very little reserve is available.

Comparison of Soils. – Fig. 30 shows average curves of true tire load vs. drawbar pull for each of the soil conditions tested. Note how all curves cross the abscissa showing a negative drawbar pull at zero tire load. This condition again is ac-

tually a measure of the average rolling resistance of the tractor at the test condition, and it will be noted how well it agrees with expected values. If we plot tractive effort (pull exerted by tires to overcome rolling resistance plus drawbar pull) against true tire load we get the curves of Fig. 31. It is very important that no attempt be made to extrapolate these or similar curves beyond the range of actual test loads as the rolling resistance is, of course, lower at the lower loads and an exaggerated low figure of drawbar pull may be obtained. If necessary to extrapolate below or above the test range, it would probably be best to use the average coefficient for the tire. The table in Fig. 32 gives the average traction coefficients for all soils tested together with the range over which they varied.

Effect of Tire Size. – If we reduce all our results to traction coefficients we can then study the effect of tire size on traction in a rather simple and convenient way. Fig. 33 shows average coefficients for all tires tested arranged by sections and diameters for all four soil conditions, also for the two road conditions in Fig. 34. It will be noticed that the coefficient increases both with increasing section and with increasing diameter, but it also must be kept in mind that the coefficient also increases with increasing tire load. Since the tire loads also increase with section and diameter it is necessary to plot our values in another way to see whether the increases shown are due to sections, diameter, or load. In Fig. 35 some of the foregoing data are plotted as tire load against drawbar pull, and we find tires having one rim diameter lying along one line whereas tires of another diameter lie along another. From this finding we can say that, at the same tire load, the drawbar pull apparently increases with increasing rim diameter, the increase being definite though not large. It is not claimed that the rim diameter in itself affects the drawbar pull as shown; but rather that it is a good index of a complicated set of variables.

In general similar relationships hold for all conditions tested, namely, that traction increases with increasing tire load and with increasing rim diameter, except that the differences are apparently more definite on the light soils. It is possible that the differences shown are due to rolling resistance since they are greater in those soils where the rolling resistance is greater.

Rolling Resistance. – Among the other data of interest discovered in this investigation was that of the rolling resistance of the tractor in soft sand – the results of which undoubtedly apply in a degree to all soft soils where the tractor sinks in so that the tires always are climbing a hill, so to speak. Fig. 36 shows the average relation between tractor load and rolling resistance. It will be seen that the effort required to roll the tractor in such soft soil is quite high initially but that added load does not increase this amount proportionately but actually adds but a small amount. For instance, a 4000-lb. trac-

SOIL CONDITION	RATING – % TRAVEL REDUCTION	TRACTION COEFFICIENT AT 12#		
		LOW	HIGH	AVERAGE
CONCRETE ROAD	5%	.57	.75	.66
DRY CLAY	16%	.52	.66	.55
SANDY LOAM	16%	.45	.58	.50
DRY FINE SAND	16%	.29	.42	.36
GRAVEL ROAD	5%	.32	.41	.36
ALFALFA	8%	.31	.41	.36

Fig. 32 – Average Traction Coefficient for All Test Conditions

TRACTION COEFFICIENTS AT 12# INFL.								
RIM DIA.	SOIL CONDITION	SECTION DIAMETER						
		7.50	8.25	9.00		11.25		13.50
24	CONCRETE ROAD	.60		.61		.68		
	GRAVEL ROAD	.33		.34		.36		.38
28	CONCRETE ROAD			.65				
	GRAVEL ROAD			.34				
32	CONCRETE ROAD							
	GRAVEL ROAD							.36
36	CONCRETE ROAD	.72		.69				
	GRAVEL ROAD	.37		.36		.37		
40	CONCRETE ROAD							
	GRAVEL ROAD		.38	.38				

Fig. 34 – Effect of Tire Size on Traction Coefficient for Road Conditions

TRACTION COEFFICIENTS AT 12# INFL.								
RIM DIA.	SOIL CONDITION	SECTION DIAMETER						
		7.50	8.25	9.00		11.25		13.50
24	DRY CLAY	.52		.55		.54		.59
	SANDY LOAM	.49		.49		.52		.55
	DRY SAND	.31		.33		.35		.39
	ALFALFA	.33		.35		.33		.37
28	DRY CLAY			.49				
	SANDY LOAM			.49				
	DRY SAND			.33				
	ALFALFA			.36				
32	DRY CLAY							.63
	SANDY LOAM							--
	DRY SAND							.41
	ALFALFA							.39
36	DRY CLAY	.55		.56		.56		
	SANDY LOAM	.50		.53		.54		
	DRY SAND	.36		.39		.39		
	ALFALFA	.38		.34		.39		
40	DRY CLAY		.57	.56				
	SANDY LOAM		.54	.55				
	DRY SAND		.35	.37				
	ALFALFA		.36	.39				

Fig. 33 – Effect of Tire Size on Traction Coefficient for Soils

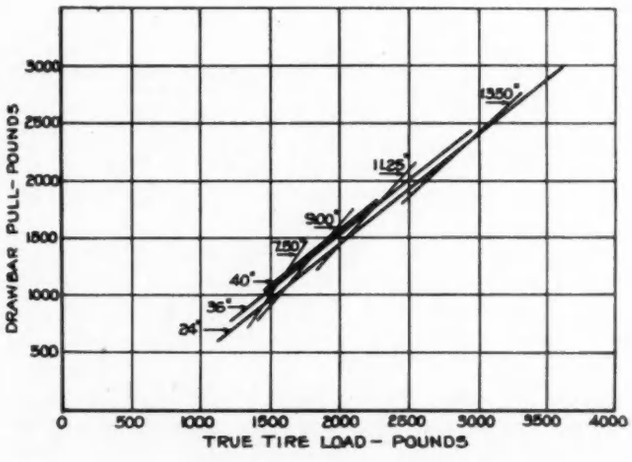


Fig. 35 – Effect of Section, Rim Diameter, and Tire Load on Drawbar Pull in Sandy Soil

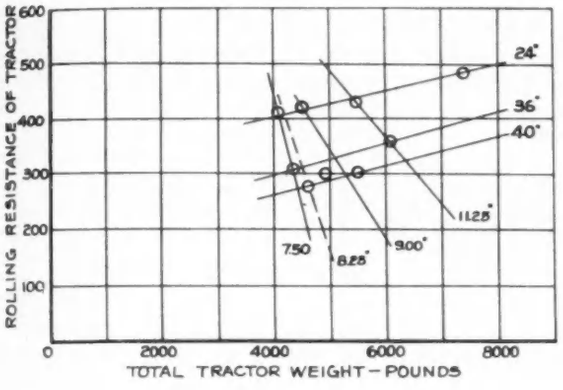


Fig. 36 – Rolling Resistance of Test Tractor in Sandy Soil

RELATION OF FRONT TO REAR LOADS

	TIRE SIZE										
SECT. RIM	7.50 24	7.50 36	8.25 40	9.00 24	9.00 28	9.00 36	9.00 40	11.25 24	11.25 36	13.50 24	13.50 32
TOTAL FRONT AXLE LOADS	1280 1390 1500 1810	1280 1460 1610 1790	1380 1560 1740 1920	1360 1560 1740 1920	1380 1610 1820 2030	1380 1640 1880 2140	1460 1710 1960 2210	1580 1830 2100 2370	1490 1740 2070 2370	1640 2030 2430 2790	1710 2000 2410 2820
TOTAL REAR AXLE LOADS	2320 2520 2720 2920	2480 2800 3080 3400	2620 2860 3200 3540	2380 2700 3040 3360	2440 2840 3220 3620	2520 2980 3440 3900	2720 3180 3640 4100	2780 3220 3760 4280	3340 3820 4420 5020	4120 4860 5860 6240	4740 5460 6200 6980
TOTAL TRACTOR LOADS	3600 3910 4220 4530	3760 4260 4670 5190	3900 4440 4940 5460	3760 4260 4780 5280	3820 4480 5040 5650	3900 4620 5330 6040	4180 4830 5600 6310	4340 5110 5880 6660	4850 5560 6490 7390	5760 6890 8010 9060	6460 7860 8610 9800
AVERAGE % LOAD ON FRONT	36%	34%	35%	36%	36%	35%	35%	36%	32%	30%	28%

Fig. 37 – Front Axle, Rear Axle, and Total Tractor Weight with Proportion of Weight on Front

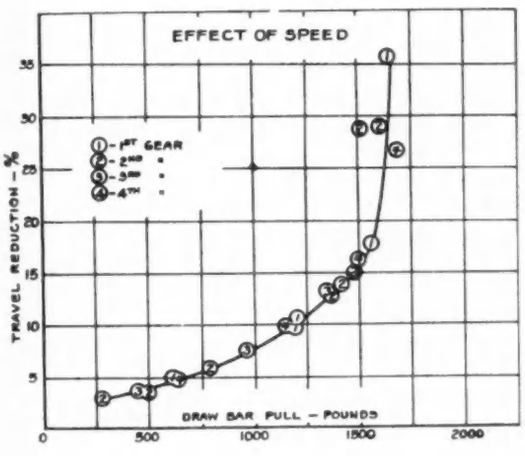


Fig. 38 – Effect of Speed on Drawbar Pull and Travel Reduction

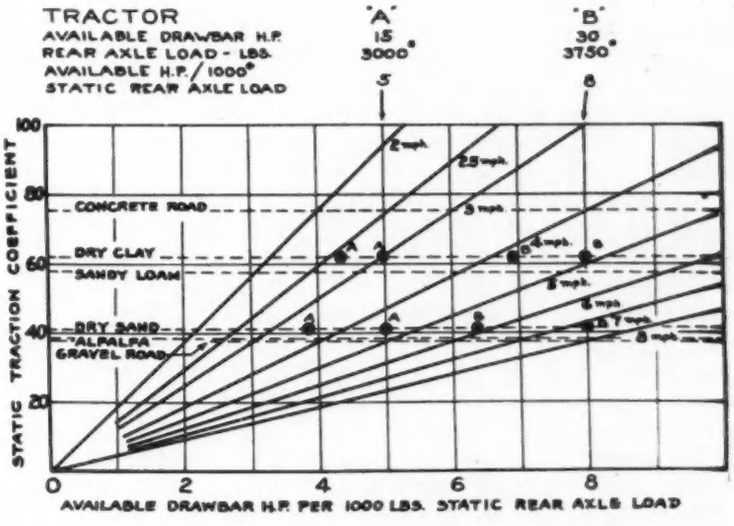


Fig. 39 – Effect of Soil Conditions and Tractor Characteristics on Speed Required to Utilize Available Drawbar Horsepower

tor on 24-in. tires has a rolling resistance of 400 lb., whereas an 8000-lb. tractor has a rolling resistance of only 500 lb. Thus it will be seen that adding weight to get traction does not always increase the rolling resistance so much as to make it uneconomical, in fact, it may so reduce the slippage of the drive wheels that the speed and efficiency is increased. (Adding 1000-lb. wheel weights increases rolling resistance 25 lb. and traction, say 400 lb.) Fig. 37 shows the front axle and rear axle as well as total load on the tractor. The proportion of total load on the front axle is also given.

Effect of Speed on Traction. – Another factor investigated was the effect of different speeds on the relation between drawbar pull and travel reduction. Fig. 38 shows the curve as obtained at the lowest speed (2 m.p.h.) together with the points obtained at three higher speeds (2.5, 3, and 4 m.p.h.). It is immediately evident that the points at higher speeds fit the initial curve equally well so that we can say that, up to 4 m.p.h., speed has no effect on the basic curve of drawbar pull against travel reduction.

Efficiency of Water-Filled Tires. – A second item investigated was the efficiency of water used for weight as compared with the usual cast-iron wheel weights. This investigation was conducted on both a loose loam soil and a hard dry clay with the same result, namely, that an equal weight of water or cast iron produced the same additional traction.

Discussion

Load Vs. Speed. – The major problem with rubber-tired tractors is that of securing traction for heavy jobs such as plowing. There are, we believe, two solutions to the problem. First we can load down the drive wheels of our rubber-tired tractor until, even at the lower traction coefficient, enough traction is developed to handle the job. In some cases this method is prohibitive, in which case the only solution is to lighten the drawbar pull by reducing the number of bottoms. However in some cases it may be desirable to add some load since, as we have seen, the penalty in added rolling resistance is not great.

The other and perhaps more logical solution of the problem lies in increasing the speed and reducing the drawbar pull required, thus utilizing the full horsepower and obtaining a greater amount of work per hour. Stated simply, it means that a steel-wheel tractor is a low-speed, high-pull machine, whereas the rubber-tired tractor is a high-speed, moderate-pull

tractor. With the trend to light-weight tractors this second alternative seems the only answer especially in view of the fact that the light-weight tractor has an initial disadvantage in rolling resistance on soft soil.

Fig. 39 is a chart that summarizes the entire situation, including horsepower, weight, speed and traction coefficient. From this chart we can see what speed is required at any traction coefficient to utilize the full horsepower of a given tractor. For example, let us take two widely different tractors, *A* and *B*. Tractor *A* develops a maximum drawbar hp. of 15 and the weight on the drive wheels is 3000 lb. which gives us 5 hp. to dissipate for every 1000-lb. load. Assuming at first that there is no rolling resistance from our chart we see that, on the dry clay soil, a speed of 3 m.p.h. will utilize the full horsepower whereas, on sand, it would require a speed of 4.6 m.p.h. This condition means that, if we apply a drawbar such that the drive wheels slip 16 per cent in both soils, the speeds given will just handle the power. Now let us consider Tractor *B*, at the other end of the scale. Tractor *B* develops a maximum drawbar hp. of 30 with the weight on the drive wheels 3750 lb., or 8 hp. per 1000-lb. load. Again consulting the chart, we find 4.8 m.p.h. required on dry clay and approximately 7 m.p.h. required on sand to utilize the available horsepower. It is quite obvious that, with the present gearing used, most of which was designed for steel wheels, Tractor *A* will be operating far more efficiently than Tractor *B*.

But let us also see the effect of rolling resistance. Introducing this factor reduces the available drawbar horsepower, thus shifting the points previously found to the left and reducing the speed required to utilize the smaller amount of power. Thus we find for Tractor *A*, instead of 3 m.p.h. on clay, 2.6 m.p.h. and instead of 4.6 m.p.h. on sand, 3.5 m.p.h. Also, with Tractor *B* we find the clay speeds reduced from 4.8 to 4.2 m.p.h. and the sand speed, from 7 to 5.9 m.p.h.

However, the basic difference between Tractor *A* and Tractor *B* remains – one requiring approximately twice the speed of the other to utilize its full available horsepower.

Before closing we should like to call attention to the fact that most of the conditions tested represented problem conditions and were, by no means, chosen as being advantageous to the rubber tire.

Conclusions

(1) The most important factor affecting the coefficient of traction or tire thrust of rubber-tired tractors is the nature of the operating surface or soil.

(2) For a given soil, the most important factor affecting the drawbar pull of a rubber tire is the weight that it carries.

(3) Inflation pressure has a relatively small effect, lower pressures being advantageous on loose sandy soils. This advantage disappears on firmer soils.

(4) For any given tire size, the coefficient of traction increases slightly with increasing tire load.

(5) For a given load and section diameter, tires having a larger rim diameter have a slightly higher coefficient of traction.

(6) For a given load and rim diameter, the change in coefficient of traction with increased section diameter is negligible.

(7) In soft soils rolling resistance is quite high – building up very rapidly with light loads, but increasing only a little over the range of normal tractor weights.

(8) Speed in itself has no effect on the traction developed by a rubber tire – up to at least 4 m.p.h.

(9) Equal weights applied to the wheel or as water in the tires produce equal additional traction.

(10) Tractors with high horsepower-to-weight ratios have to travel faster to utilize their available horsepower, or use added weights to operate at slower speeds.

The Supercharger – Its Progress and Prospects

ACCORDING to Webster, a supercharger is defined as "a device such as a blower, compressor, or pump for increasing the volume air charge of an internal-combustion engine over that which would normally be drawn in through the pumping action of the pistons. It is used to compensate for the lower density of air in altitude operation of aircraft engines or the deficiency of air charge in high-speed automotive operation." Because of the present widespread interest in such devices, I will attempt to give you a brief résumé of their past developments, and some idea of the problems connected with their design and application.

The idea of a supercharger or blower is, of course, not new. As far back as 1904 the eminent British scientist, Sir Dougald Clerk, reported his method of obtaining higher mean pressures without higher wall temperatures by the induction of a further charge at the end of the suction stroke.

During the Great War the general subject of supercharging came in for considerable attention because airplanes were playing an important role as fighting machines and their combat value depended largely on the capacity of their engines. Under such circumstances, it was only natural that blowers should have been used to improve their horsepower-per-pound ratio.

The development of the airplane as a freight and passenger transporting medium has focused even more attention on the supercharger as an engine-power- and efficiency-increasing instrument. It is safe to say that the advancement of general commercial aviation has been greatly accelerated by the use of supercharged engines and such service has naturally done much to take the supercharger out of the accessory class and establish it as an accepted principle of engine design.

In connection with the supercharging of aviation engines, there are some interesting facts worth mentioning. The case of the famous China Clipper is an outstanding example and it has been said that there would have been no room for payload if the engines had not employed positive or pressure induction.

In addition to its impressive record in the aircraft industry, the supercharger has proved very useful in several other fields. The fact that Gar Wood's "Miss America" holds the world's speed record on water clearly illustrates the ability of blowers to improve the performance of her inherently powerful engines.

Racing car powerplants have successfully employed positive induction for a considerable period. The most recent evidence is contained in the record of the great Columbus Day race at the Roosevelt Speedway on October 12th of 1936. It is significant that the first five places were won by cars which were driven by supercharged engines.

In the passenger car fields, although several very high-priced jobs with positive-induction equipment have been available for some time, the entrance of the supercharger into the popular-priced field is comparatively recent.

Excerpts from the paper by the same title by Ralph A. Plumb, Graham-Paige Motors Corp., presented before the Detroit Section Meeting of the Society, Detroit, Mich., April 26, 1937.

Report of the Volunteer Group for Compression-Ignition Fuel Research

By C. H. Baxley and T. B. Rendel
Secretary *Chairman*

FOLLOWING the adoption of a suitable design of engine and a tentative procedure for operating this engine, the work of the Volunteer Group has covered the investigation of other methods of measuring cetane number of Diesel fuels looking towards a simplification and improvement of reproducibility of the procedure.

Results of a second series of cooperative tests are given, using the procedure adopted in the Group's last report together with a series of tests on the same fuel using the critical-compression-ratio method with an interval timing-control device. Results of the first series do not show such good agreement, the grand average deviation on twelve samples being of the order of ±1.9. Results of the critical-compression-ratio tests show improved agreement due to better standardization.

Tests on three alternative methods based on the delay method, but using different instruments for recording the delay, are given. Results on two different full-scale engines also are presented.

It is concluded from the results of the past 18 months' work that a direct matching method on the basis of ignition delay is the best from the point of view of reproducibility and validity, even though this method will involve some sacrifice in simplicity and speed of testing. The progress of methods of instrumentation is advanced considerably, and it appears that the bouncing-pin type should be discarded in favor of a balanced-diaphragm or magnetic-pickup type.

DURING the past few years the development of the high-speed Diesel engine has been assisted materially through increasing knowledge of the various properties of Diesel fuels, particularly in relation to a property known as its ignition quality. Standardization of the measurement of this quality has been in the hands of a small group known as the Volunteer Group for Compression-Ignition Fuel Research, and this group now presents its third report dealing with this subject. In its first report delivered to this society in January, 1935, a tentative design of an engine suitable for measuring ignition qualities was proposed. In its second report, a tentative procedure for operating this engine, a recommendation as to the use of primary and secondary reference fuels, and the results of a series of cooperative tests using 9 different types of fuels, were presented.

In this, its third report, the results of a second series of cooperative tests using two different methods of operation are given, together with comments on suitability of either method. Other proposed methods are discussed, and some results of tests carried out by these methods are given. Results on two different types of full-scale engines also are presented as an indication of the validity of the laboratory test methods.

The present membership of the Volunteer Group for Compression-Ignition Fuel Research is as follows:

T. B. Rendel, Chairman	Shell Petroleum Corp.
C. H. Baxley, Secretary	International Aviation Associates
W. G. Ainsley	Sinclair Refining Co.
Dr. C. Banta	E. I. du Pont de Nemours
Dr. D. P. Barnard	Standard Oil Co. (Ind.)
Dr. A. E. Becker	Standard Oil Development Co.
T. A. Boyd	General Motors Corp.
H. K. Cummings	National Bureau of Standards
Com. R. F. Good	U. S. Naval Experiment Station
W. H. Hubner	Universal Oil Products Co.
Capt. F. D. Klein	U. S. Army Air Corps
Prof. L. C. Lichty	Yale University
Neil MacCoull	The Texas Co.
J. R. MacGregor	Standard Oil Co. of Calif.
A. W. Pope	Waukesha Motor Co.
J. R. Sabina	Atlantic Refining Co.
C. H. Schlesman	Socony-Vacuum Oil Co., Inc.
H. G. Vesper	Standard Oil Co. of Calif.

In addition to the foregoing the Committee has cooperated with others in the United States interested in this type of research and also with the Institute of Petroleum Technologists Standardization Committee No. 9 in England.

Program of Work

At the beginning of 1936, the situation in regard to ignition-quality measurements can be summed up as follows:

Cooperative tests using the delay method, recording the

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, May 8, 1937; the Annual Meeting of the American Society for Testing Materials, New York, N. Y., June 28 to July 2, 1937; and the Second World Petroleum Congress, Paris, France, June 14-19, 1937.]

Table 1 – Volunteer Group Cooperative Diesel Fuel Tests – Bouncing-Pin Delay Method* – Per Cent High-Cetane-Number Reference Fuel

Laboratory	1	2	3	4	5	6	7	8	9	10	11	12
B	73.9	73.7	88.9	55.9	73.2	63.5	57.2	56.2	45.1	....	....	66.7
D	76.0	77.0	99.0	53.0	78.0	66.0	57.0	57.0	43.0	90.0	93.0	71.0
E	70.1	64.9	90.1	57.0	75.2	61.7	57.3	58.0	44.8	75.1	82.3	53.1**
G	69.9	74.6	90.0	56.0	76.1	65.0	58.1	60.0	46.1	80.0	84.8	68.4
H	75.0	83.0	96.0	60.0	75.0	65.0	56.0	61.0	48.0	86.0	86.0	70.0
I	73.4	77.1	98.5	55.3	72.8	62.5	60.0	56.5	47.6	84.6	91.8	69.5
K	73.0	69.0	88.0	52.0	71.0	63.0	60.0	54.0	48.0	81.0	85.0	65.0
L	75.0	77.0	91.0	53.0	72.0	66.0	59.0	56.0	44.0	85.0	81.0	68.0
N	75.0	71.0	86.5	59.0	74.0	64.5	55.0	61.5	52.0	81.0	81.0	70.0
P	74.3	69.3	88.7	52.6	70.0	63.0	65.2	61.3	47.2	84.2	81.4	69.2
Q	74.0	78.0	90.0	56.0	73.0	64.0	58.0	58.0	47.0	85.0	82.0	68.0
Average	73.6	74.1	91.5	55.4	73.7	64.0	58.4	58.1	46.6	83.2	84.8	68.6
Average Deviation	1.5	4.2	3.4	2.1	1.8	1.2	1.9	2.0	1.8	3.1	3.3	1.4
Maximum Deviation	+2.4	+8.9	+7.5	+4.6	+4.3	+2.0	+6.8	+3.4	+5.4	+6.8	+8.2	+2.4
	−3.7	−9.2	−5.0	−3.4	−3.7	−2.3	−3.4	−4.1	−3.6	−8.1	−3.8	−3.6

Table 2 – Volunteer Group Cooperative Diesel Fuel Tests – Bouncing-Pin Delay Method* – Cetane Number

Laboratory	1	2	3	4	5	6	7	8	9	10	11	12
B	55	55	66	42	54	47	42	42	33	..	..	49
D	56	57	73	39	58	49	42	42	32	67	69	53
E	52	47	69	40	56	44	41	41	30	56	62	37**
G	52	55	67	41	56	48	43	44	34	59	63	51
H	55	61	71	44	55	48	41	45	35	63	63	51
I	55	58	74	42	55	47	45	42	36	63	69	52
K	54	51	65	38	52	46	44	40	35	60	63	48
L	56	57	67	39	53	49	44	41	33	63	60	50
N	55	52	63	43	54	47	40	45	38	59	59	51
P	55	51	66	39	52	47	41	45	35	62	60	51
Q	55	58	67	42	54	47	43	43	35	63	61	50
Average	55	55	68	41	54	47	43	43	34	62	63	51
Average Deviation	0.8	3.2	2.7	1.6	1.4	0.9	1.3	1.5	1.6	2.3	2.5	0.9
Maximum Deviation	+1	+6	+6	+3	+4	+2	+3	+2	+4	+5	+6	+2
	−3	−8	−5	−3	−2	−3	−2	−3	−4	−6	−4	−3

* For test procedure see Appendix A.
** This sample is apparently contaminated and the results are not included in the average.

delay by the bouncing pin and mechanical contacts on the fuel-injection valve, indicated that this method gave somewhat more reproducible results than were obtained with the critical-compression-ratio method. However, it was agreed generally that the bouncing pin was very difficult to set and, in view of the greater simplicity of the critical-compression-ratio method, more attention should be given to the standardization of details of this method in an endeavor to improve its reproducibility. It also was desired to investigate other methods of recording the delay in an endeavor to simplify the delay method of determining cetane number.

Accordingly, a Program Committee with W. H. Hubner as chairman was appointed to arrange a second series of cooperative tests using the delay method and the C.C.R. (critical-compression-ratio) method. Details of both methods were standardized somewhat more closely, and arrangements made for investigating other methods of rating fuels proposed by members of the Group, such as a method using a balanced diaphragm in place of the bouncing pin proposed by the Socony-Vacuum Oil Co., Inc., or that proposed by Schweitzer and Hetzel of Penn State in their paper presented to the Society of Automotive Engineers, January, 1936.[1]

[1] See S.A.E. TRANSACTIONS, May, 1936, pp. 206-216: "Cetane Rating of Diesel Fuels," by P. H. Schweitzer and T. B. Hetzel.
[2] See S.A.E. TRANSACTIONS, June, 1936, pp. 225-233: "Report of the Volunteer Group for Compression-Ignition Fuel Research," by T. B. Rendel.

Test Methods and Fuels Used

Appendixes A and B give details of the bouncing-pin delay method and the C.C.R. methods as finally adopted by the Group and used in the latest series of experiments. Other methods tried are described in the sections dealing with this point.

The widest possible range of fuels was selected for this second group of tests and, in order to investigate any possible effects of viscosity, two fuels of different viscosity but of the same crude source were chosen where possible. Appendix C gives the physical properties of the fuels used, together with a brief description of their source. In order to avoid calibration difficulties, it was agreed that results should be reported in terms of the secondary reference fuels.

The secondary high-cetane-number reference fuel is a straight-run gas oil obtained from a small field in southern Illinois. The low-cetane-number secondary reference fuel is commercial methylnaphthalene (a mixture of alpha and beta methylnaphthalene). The reference fuels used are those recommended in the previous report.[2]

Results of Tests

Table 1 shows the results obtained by the bouncing-pin delay method expressed in terms of secondary reference fuels. Fig. 1 shows the average deviation for each laboratory in graphical form. For the sake of convenience, the results were

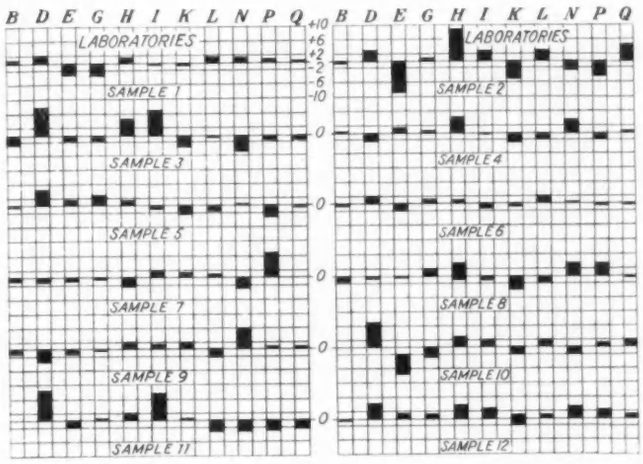


Fig. 1 – Volunteer Group Cooperative Tests – Delay Method – Deviations from Average Results – Per Cent Shell Reference Fuel

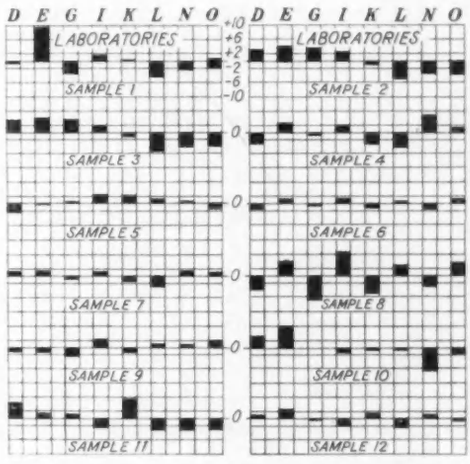


Fig. 2 – Volunteer Group Cooperative Tests – C.C.R. Method – Deviations from Average Results – Per Cent Shell Reference Fuel

also expressed in terms of cetane numbers reported to the nearest whole number (Table 2). These values were obtained by the average calibration curve for the delay method published in last year's report. (See Appendix D.) A composite calibration curve is not available for the secondary reference fuels on the basis of the critical-compression method.

The deviations are somewhat larger than those found in the previous series of tests and indicate that a year's experience with the bouncing pin has not shown any great improvement in its reproducibility. No explanation for this deterioration in reproducibility is evident. The maximum variations are perhaps somewhat larger than are desirable and are probably due to difficulties occurring in the adjustment of the bouncing pin or to unsteady fuel-injection timing. Except in the case of the California gas oil, there is no definite evidence of the effect of viscosity on either the cetane number or the variability of the results. In the California gas oil the relatively high cetane number of the low-viscosity but wide-boiling-range fuel (Fuel 6) is interesting when compared with the other straight-run California fuels (Fuels 7 and 8).

Tables 3 and 4 show results obtained by the critical-compression-ratio method in terms of secondary reference fuels and cetane numbers. The cetane numbers in this case were obtained from the individual calibration curves used by the laboratories reporting. Fig. 2 shows deviations in graphical form by this method.

Table 3 – Volunteer Group Cooperative Diesel Fuel Tests – Critical-Compression-Ratio Method* – Per Cent High-Cetane-Number Reference Fuel

Laboratory	1	2	3	4	5	6	7	8	9	10	11	12
D	77.0	77.0	84.0	57.0	64.0	67.0	61.0	61.0	46.0	89.0	90.0	68.0
E	87.1	78.1	....	62.8	66.6	70.1	61.2	68.8	45.6	92.1	87.0	69.8
G	74.0	77.7	84.3	60.0	67.0	68.2	58.8	58.0	44.6	86.2	86.1	67.1
I	79.1	76.0	85.5	62.5	68.8	70.5	60.8	71.4	49.5	85.0	83.1	65.3
K	77.8	72.7	84.4	57.2	66.3	67.5	58.4	59.8	45.4	85.8	90.8	68.8
L	73.0	69.0	86.0	56.0	68.0	69.0	57.0	68.0	48.0	86.0	82.0	65.0
N	75.0	70.0	75.0	65.5	67.0	67.0	61.5	61.5	48.0	80.0	82.0	68.5
O	78.0	70.0	85.0	62.0	65.0	70.0	61.0	69.0	49.0	85.0	82.0	67.0
Average	77.6	73.8	83.5	60.4	66.6	68.7	60.0	64.7	47.0	86.1	85.4	67.4
Average Deviation	2.9	3.4	2.4	2.8	1.1	1.2	1.4	4.6	1.6	2.2	3.1	1.3
Maximum Deviation	+9.5	+4.3	+2.5	+5.1	+2.2	+1.8	+1.5	+6.7	+2.5	+6.0	+5.4	+2.4
	−4.6	−4.8	−8.5	−4.4	−2.6	−1.7	−3.0	−6.7	−2.4	−6.1	−3.4	−2.4

* For test procedure see Appendix B.

Table 4 – C.C.R. Method* – Cetane Number

Laboratory	1	2	3	4	5	6	7	8	9	10	11	12
E	67.9	61.2	....	48.7	52.1	54.7	47.2	53.8	33.7	71.6	67.8	54.2
G	55.4	58.2	63.3	45.0	50.2	51.3	44.1	43.4	33.4	65.0	64.9	50.3
I	62.0	59.5	67.3	48.4	53.5	55.0	47.1	55.7	37.7	66.8	65.3	50.7
K	60.0	56.0	65.0	44.0	51.0	52.0	45.0	46.0	35.0	66.0	70.0	53.0
N	55.0	51.0	55.0	48.0	49.0	49.0	45.0	45.0	35.0	58.0	60.0	50.0
Average	62.5	58.8	65.5	47.9	52.6	53.6	46.1	49.2	35.1	68.4	68.0	53.0
Average Deviation	5.1	3.8	5.6	2.2	2.7	2.9	1.4	4.4	1.1	5.9	4.7	2.7
Maximum Deviation	+12.5	+8.2	+11.5	+5.1	+7.4	+6.1	+1.9	+6.5	+2.6	+14.6	+12.0	+7.0
	−7.5	−7.8	−11.5	−3.9	−3.6	−4.6	−2.0	−5.8	−1.7	−10.4	−8.0	−3.0

* For test procedure see Appendix B.

Table 5 – Volunteer Group Cooperative Diesel Fuel Tests – Results by Various Test Methods – Per Cent High Cetane Reference Fuel

Laboratory	Method	1	2	3	4	5	6	7	8	9	10	11	12
I	Socony—Vacuum	74.1	77.6	95.5	58.3	77.1	64.5	62.5	56.0	38.2	86.6	90.6	68.1
M	Delay Method	81.8	79.5	92.0	60.0	81.8	71.0	58.0	62.0	53.0	84.8	82.0	69.8
Q		75.0	77.0	91.0	57.0	74.0	63.0	60.0	60.0	46.0	84.0	87.0	70.0
	Average of above 3 Laboratories	77.0	78.0	92.8	58.4	77.6	66.1	60.1	59.3	45.7	85.2	86.5	69.0
D	Penn State	75.0	73.0	95.0	56.0	73.0	66.0	55.0	62.0	45.0	82.0	81.0	68.0
G	Method—see Penn	68.2	63.5	94.0	57.6	73.2	63.8	51.6	61.2	46.0	76.5	73.2	68.8
I	State Bulletin	77.5	76.6	89.8	56.8	77.0	68.3	58.8	60.3	44.5	86.3	84.8	68.5
L	No. 45 by T. B.	76.0	74.0	88.0	54.0	72.0	65.0	54.0	59.0	40.0	84.0	81.0	67.0
R	Hetzel	74.0	75.5	94.0	58.0	76.0	64.0	57.0	61.0	45.5	85.5	79.5	68.5
	Average of above 5 Laboratories	74.1	72.5	92.2	56.5	74.2	65.4	55.3	60.7	44.2	82.9	79.9	68.2
G	Modified Magnetic	71.4	70.0	93.1	56.2	76.0	62.8	56.7	57.9	45.2	85.0	80.9	65.0
H	Pickup Methods	77.0	74.0	90.0	55.0	77.0	64.0	60.0	63.0	42.0	85.0	85.0	70.0
O		74.0	71.0	89.0	56.0	75.0	65.0	62.0	59.0	47.0	85.0	81.0	68.0
	Average of above 3 Laboratories	74.1	71.7	90.7	55.7	76.0	63.9	59.6	60.0	44.7	85.0	82.3	67.7
	Average of Bouncing-Pin Delay	73.6	74.1	91.5	55.4	73.7	64.0	58.4	58.1	46.6	83.2	84.8	67.2
	Average of Critical-Compression-Ratio Method	77.6	73.8	83.5	60.4	66.6	68.7	60.0	64.7	47.0	86.1	85.4	67.4

The deviations by this method are an improvement over those obtained last year and are now about the same as those obtained by the delay method. Standardization of details probably has accounted for this improvement over last year, and the method is apparently a promising one provided the validity of the results in terms of service engines is not questioned. This point is discussed later.

Other Methods

In addition to the preceding two methods the Committee investigated other possible methods. Data are available on three methods, using the samples reported in Appendix C, by more than one laboratory. All these methods involve the measurement of ignition delay, comparing this delay to known blends of reference fuels. Operating conditions are generally similar to those used in the bouncing-pin delay method, the differences being mainly those involved in measuring the delay. Table 5 shows the results of a series of tests on three different methods.

The method devised by Socony-Vacuum involves the use of a balanced pressure diaphragm in place of the bouncing pin, but otherwise uses the same engine-operating conditions as the bouncing-pin-delay method.

An insulated contact is placed on the tip of the injector feeler pin. A slight rise of the valve makes a contact which starts an electric current flow. A balanced pressure diaphragm is placed in the bouncing-pin hole in the engine head. Pressure on top of the diaphragm is arbitrarily maintained at 50 lb. per sq. in. above compression pressure by connection to a CO_2 bottle. Combustion pressure causes contact to be made which stops the before-mentioned electric current flow. The successive delay times or current times are averaged by means of a special-type condenser and a vacuum-tube voltmeter. The start of injection is indicated visibly by a neon flasher and flywheel pointer which shows the injection time in degrees on a flywheel protractor for each cycle.

Details of the apparatus may be obtained from the Socony-Vacuum Oil Corp. Comments of those who have used the apparatus are favorable in regard to ease of handling, and the results compare well among themselves and also with the averages by the bouncing-pin-delay method as shown in Fig. 3.

The Penn State method is fully described in Pennsylvania State College Engineering Bulletin No. 45. In general, the results compared reasonably well with the other delay methods. (See Fig. 4.) Although the speed of testing was increased, it is believed that a modification involving direct matching is desirable.

Two modifications of this method involving direct matching have been devised, one by W. G. Ainsley of Sinclair Refining Co. and the other by W. H. Hubner of the Universal Oil Products Co. Broadly speaking, these modifications include changes in design of the pickups and the addition of a second neon light to the flywheel.

In the Sinclair set-up, instead of the thyratron, a radio-type amplifier is used to flash a neon tube on the flywheel. The amplifier is standard, except that the final tube is operated at such a grid bias that no current flows until the output of the pickup exceeds a certain value. By varying this bias, the neon tube can be made to flash on firing but not on compression. Pickups are mounted on the injector and on the cylinder.

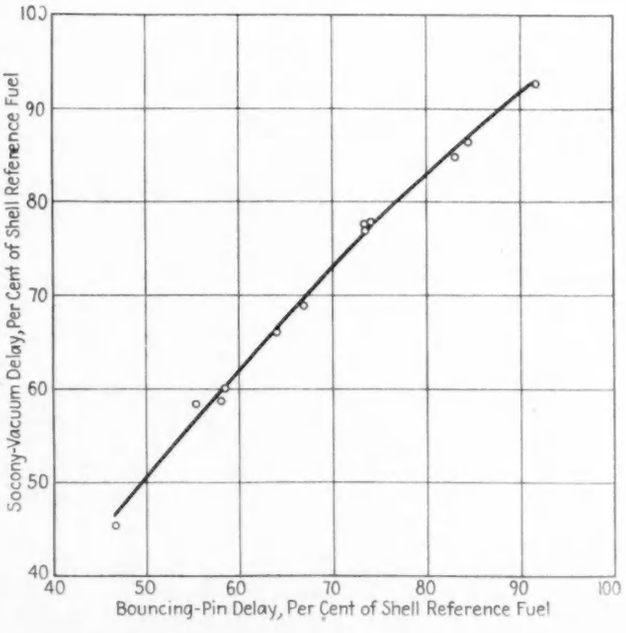


Fig. 3 – Volunteer Group Cooperative Tests – Correlation Between Socony-Vacuum and Bouncing-Pin Delay Methods

Neon tubes are mounted 15 deg. apart on the flywheel, one being flashed by the firing-point pickup, and the other by the injection-angle pickup. When the flashes on the two neons apparently coincide, the delay angle is 15 deg., and fuels are rated by setting the sample so that the two flashes coincide and bracketing between reference blends with which the firing flash appeared after or before the injection flash.

In the Universal Oil Products set-up, specially built magnetic pickups and a four-tube rectifier and amplifier are used in conjunction with two neon lights on the flywheel. One of the lights is operated through a specially built contact breaker which is driven from the tachometer driveshaft; the movement of the contact breaker is measured by a standard C.F.R. micrometer gage. With this arrangement, three flashes are shown on the flywheel circumference; injection and combustion on one neon light and the flash made by the contact breaker on the second light. The method of operation is simple:

With the compression ratio adjusted so that combustion occurs at top dead-center, the breaker is adjusted until its neon flash is in line with the combustion flash. The micrometer setting is noted. The micrometer settings for two reference fuels, one better and one worse than the test sample, are noted also, without changing the compression ratio. It is a simple matter then to arrive at the rating of the test fuel.

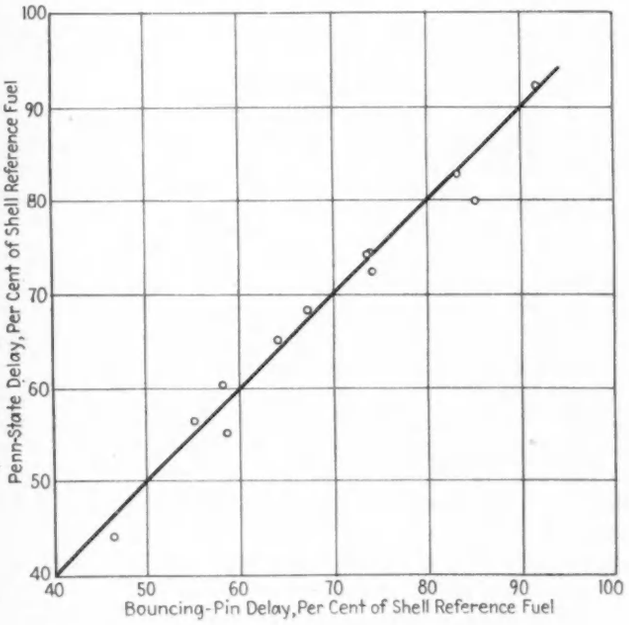


Fig. 4 – Volunteer Group Cooperative Tests – Correlation Between Penn State and Bouncing-Pin Delay Methods

The results of these modifications of the Penn State method also compare well with the bouncing-pin-delay method results as shown in Fig. 5.

Correlation With Full-Scale Engines

An indication of the validity of the testing procedures is obtained from two sets of results on full-scale engines on this series of fuels, one by the U. S. Naval Experiment Station at Annapolis, Md., and the other from the Standard Oil Co. of Calif. The U. S. Navy results, together with their method of rating, have been previously given in a report by Lieut-Com. R. F. Good in his paper "Cetane Numbers, Life Size," presented before the Society last January.[3]

[3] See S.A.E. TRANSACTIONS, June, 1937, pp. 232-242: "Cetane Numbers – Life Size," by R. F. Good.

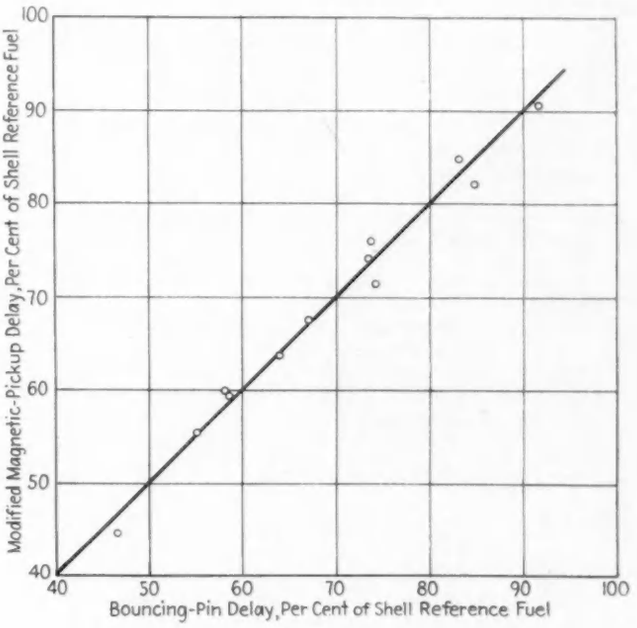


Fig. 5 – Volunteer Group Cooperative Tests – Correlation Between Magnetic-Pickup and Bouncing-Pin Delay Methods

The Standard Oil Co. of Calif., using a Fairbanks-Morse Model 36A engine, rated the fuels by two methods.

Method No. 1 consists of idling the 4¼- by 6-in. single-cylinder Fairbanks-Morse engine at 1200 r.p.m. and throttling the intake air until misfire occurs, as observed through a glass window in the combustion-chamber. The equivalent altitude is recorded and, by comparison of this value with daily calibration curves of critical altitude vs. per cent reference fuel, the matching reference fuel blend is obtained.

In Method No. 2, a cathode-ray oscillograph is used in conjunction with a new engine indicator developed in their laboratory. By this method the injection advance is adjusted at 35 per cent load and 1200 r.p.m. with each fuel until firing occurs at top-center as observed on the oscillograph screen. Contacts on the pintle actuate a thyratron relay which lights up a neon tube on the flywheel, thus indicating stroboscopically the timing of injection directly on the flywheel quadrant.

These results, together with the average cetane numbers as obtained by the delay method and C.C.R. method, are given in Table 6.

Fig. 6 shows graphically the correlation between the U. S. Navy results on the Winton engine by both the delay and computed-combustion-knock methods and the Standard Oil results on the Fairbanks-Morse engine by the delay method. The greater deviations of these data are probably caused by the fact that only single short-time tests were made on each fuel in the Winton engine as compared with average data from one or more check tests on the Fairbanks-Morse engine. It is interesting to note that the agreement is fairly close with the delay method, regardless of the type of instrumentation used.

Fig. 7 shows that the critical altitude on the Fairbanks-Morse engine correlates somewhat more closely with the critical compression method. This correlation, however, is not unexpected in view of the similarity of the two methods of test, and the difference in correlation with either method is not great.

Further full-scale laboratory and field tests on a larger number of engines are necessary before general conclusions can be drawn.

Table 6 – Volunteer Group Cooperative Diesel Fuel Tests – Correlation With Full-Scale Engines

Laboratory	Engine	Rating Based on	1	2	3	4	5	6	7	8	9	10	11	12
U. S. Navy	Winton Submarine	Ignition Delay	56	48	70	42	62	45	46	40	36	65	67	47
		Combustion Shock	55	59	65	37	62	49	43	45	33	55	59	55
Standard Oil of Calif.	Fairbanks-Morse	Critical Altitude	64	64	71	49	64	54	49	50	39	71	70	55
		Ignition Delay	55	58	69	41	52	47	44	40	34	69	68	51
Average of all laboratories	CFR High-Turbulence	Delay	55	55	68	41	54	47	42	43	34	62	63	49
Average of all laboratories	C. F. R.	C. C. R.	63	59	66	48	53	54	46	49	35	68	68	53

Standard I.P.T. Method

Before concluding, attention should be drawn to the new tentative standardization of the Institute of Petroleum Technologists for determination of ignition quality of Diesel fuels. Three sets of Volunteer Group samples were sent over to England for correlation with the I.P.T. Group, though the results unfortunately were not received in time to be included in this report.

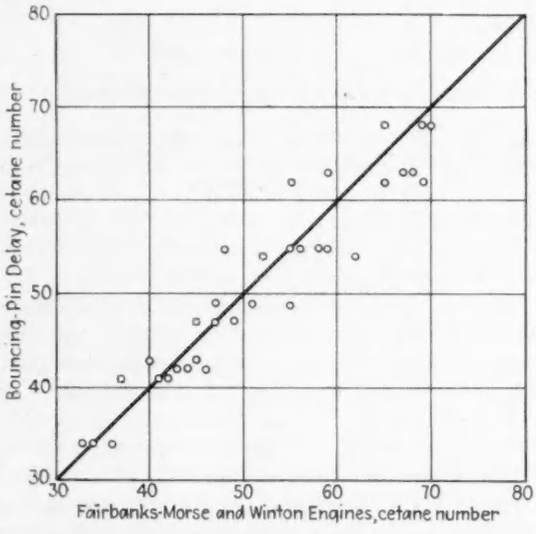


Fig. 6 – Volunteer Group Cooperative Tests – Full-Scale Engine Ignition Delay and Combustion Knock Correlation with Bouncing-Pin Delay

In this connection it is to be noted that, according to the I.P.T. plan, results are to be reported in ignition numbers. These numbers are the percentage of high ignition quality reference fuel in the low ignition quality reference fuel divided by ten and reported to the nearest half number. On this basis

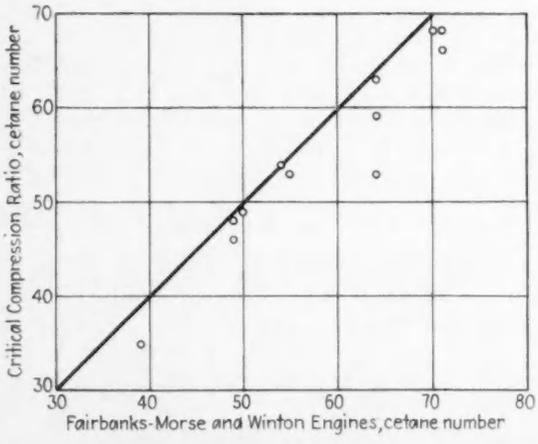


Fig. 7 – Volunteer Group Cooperative Tests – Correlation Between Fairbanks-Morse Critical Altitude and C.F.R. Critical Compression Ratio

the results of the Volunteer Group tests of all methods are in excellent agreement with one another, and it is suggested that some consideration be given to this plan both for the sake of international standardization and to avoid the use of too fine a distinction in commercial practice since such practice undoubtedly will demand reports to the nearest whole number.

It should be noted, however, that the ignition number is based on a secondary reference fuel equivalent. In this country it is not practical over a long period of time to adopt secondary reference fuels, and primary reference fuels, such as cetane and alpha methylnaphthalene, must be used.

Conclusion

In conclusion it can be said that the results of the past eighteen months' work indicate fairly definitely that a direct matching method on the basis of ignition delay is the best from the point of view of reproducibility and validity and, therefore, some sacrifice in simplicity and speed of testing must be made. In this connection it is to be remembered that the octane-number test, after several years' development and with a far greater commercial incentive behind it, sometimes requires about 45 min. for determination and is still liable to errors of one or two octane numbers.

Progress on methods of instrumentation has advanced considerably, and it is recommended that the bouncing-pin type of instrumentation should be discarded definitely in favor of the balanced-diaphragm or the magnetic-pickup type. Further work in a larger number of laboratories is desirable before definitely standardizing on this point. The cetane number is, however, not apparently affected outside the limit of error of the determination.

It is, therefore, recommended that Diesel fuels be rated for ignition quality on the basis of their cetane number as determined by an ignition-delay method on the high-turbulent Diesel conversion of the C.F.R. engine, the exact type of instrumentation for recording the delay to be left open at the option of the user pending further work of the Group.

Appendix A

Tentative Standard Operating Conditions and Procedure for Delay-Period Method of Rating Diesel Fuels.

OPERATING CONDITIONS

(1) Engine Speed ..... 900, ±3 r.p.m.
(2) Cylinder ..... High-turbulence variable-compression Diesel cylinder
(3) Jacket Temperature ... Constant within ±1 deg. fahr., limits 205-212 deg. fahr.
(4) Cooling Liquid ..... Distilled water
(5) Inlet-Air Temperature .. 150, ±2 deg. fahr.
(6) Crankcase Lubricating Oil ..... S.A.E. 30
(7) Oil Pressure ..... 25-30 lb. per sq. in.
(8) Valve Clearance ..... Intake 0.008 in. cold, Exhaust 0.010 in. cold

(9) Injection Advance	10 deg. before top dead-center (constant)
(10) Injection Pressure	1500, ±50 lb. per sq. in. (opening pressure)
(11) Fuel Quantity	13.0, ±0.5 cc. per min.
(12) Injector Cooling-Water Temperature	100 ±5 deg. fahr.
(13) Injector Specifications	Bosch DN30S3
(14) Injection-Pump Specifications	Bosch PE1B50A302/3S97, port closing at 0.075 in. ±0.005 in. lift from base circle
(15) Fuel Line – Tank to Pump	⅜-in. copper tubing
(16) Fuel Line – Pump to Injector	¼-in. outside diameter, ⅛-in. inside diameter, length, 36 in.
(17) Fuel Tank Height	25 in., ±1 in. from bottom of tank to pump inlet
(18) Knockmeter Generator Voltage	120, ±1

PROCEDURE

(A) Starting and Stopping the Engine

While the engine is being turned over by the electric motor, the fuel bypass valve on the injector is closed, and the compression ratio is increased until the engine begins to fire.

To stop the engine, the fuel bypass valve on the injector is opened and the electric motor then switched off.

(B) Checking Injection Pump for Port Closing

The pump plunger port should close when the plunger has traveled up 0.075 in. ±0.005 in. from the base circle of the cam. This setting is important as it influences the injection rate. To check the port closing see paragraph No. 14 under "Installation Instructions." This adjustment is made in the factory and should not require resetting unless it has been tampered with.

(C) Injection Pressure Setting

Remove injection pump cover and, with injection line pressure gage connected and injector arranged to spray into the air, operate the pump plunger with a screwdriver used as a lever. With the pressure gage set at 1500 lb. per sq. in., adjust the pressure on the injector spring until equal quantities of fuel spray from the gage and injector. The opening pressure of the injector will then be the same as indicated on the gage.

(D) Injector Indicator Setting

(1) Loosen the contact spring carrier clamp nuts and adjust until the spring leaf just touches the injector pin. Then set the clamp nuts to provide ½ turn initial tension on the spring.

(2) Adjust the gap between the contact points to 0.004 in.

(E) Bouncing-Pin Preliminary Static Setting

Make static bouncing-pin setting as follows:

(1) Set gap between pin and arm at 0.005 in. with gap-adjusting screw.

(2) Bear down lightly on the end of the contact-arm spring so that the arm is held on its seat. Adjust the spring-tension screw until the screw just touches the spring. Then increase the tension by turning the screw down five notches.

(F) Final Compression-Ratio and Bouncing-Pin Adjustment

After the engine has reached equilibrium, the compression-ratio and bouncing-pin setting at which an unknown fuel is rated are determined as follows:

(1) Adjust the compression ratio about two compression ratios above that at which definite misfiring occurs.

(2) With the engine firing, close the bouncing-pin gap between pin and arm by turning the adjustment screw up until two distinct lines appear ahead of the "bump" on the optical indicator diagram. (This setting indicates that the bouncing-pin arm is deflected by the compression pressure before combustion.)

(3) With the engine firing, increase the bouncing-pin gap between pin and arm by turning the adjustment screw down until the double line on the optical indicator just coincides with the base line. (This setting indicates that the bouncing-pin arm is not moved by compression pressure, but is deflected the moment compression pressure is exceeded by combustion.)

(4) Observe the angle at which combustion starts. The correct angle of combustion for making a rating is 1 deg. after top dead-center. Readjust the compression ratio until this condition is obtained.

(5) After a change in compression ratio, readjust the bouncing pin as outlined in (2) and (3) preceding.

(6) If the indicated angle of injection after the final bouncing-pin setting has shifted more than ½ deg., readjust the compression ratio and pin as outlined previously.

(7) Check the regularity of the bouncing pin on the neon-tube indicator. If the angle of combustion fluctuates more than ±1 deg., adjust the bouncing-pin tension screw by trial until steady readings are obtained.

(G) Adjustment of Contact Breaker

The make-and-break points in the knockmeter circuit should be adjusted for an 8-deg. contact period, as determined on the neon-tube indicator. This period also can be indicated on the knockmeter and should produce a reading of 80 to 100 on the scale when the generator voltage is 120 and the engine is not firing. The breaker timing should be adjusted to make contact approximately 2 deg. before top dead-center as indicated on the neon-tube indicator. Check this setting on the knockmeter with 120 generator voltage. A knockmeter reading of 50 should be obtained with the engine firing when combustion occurs at 1 deg. after top dead-center. Advance or retard the breaker until such a knockmeter reading is obtained.

(H) Cetane-Number Determination

The cetane number of a fuel is ascertained by comparing the delay (as measured with the knockmeter) for the fuel with those for various blends of the reference fuels until two blends differing in delay by not more than the equivalent of 8 cetane numbers are found, one of which has a longer delay and the other a shorter delay period than the sample. The reference fuel which would exactly match the sample is computed by interpolation from the knockmeter scale readings of the fuels.

An alternate series of knockmeter readings is taken on the test fuel and reference-fuel blends. After changing from one fuel to the other, 5 min. must be allowed to insure the complete change-over since there is a comparatively large volume of fuel in the pump and line.

At least three alternate series of readings should be taken on each fuel and, if the average knockmeter reading of the fuel sample is higher than that of the reference-fuel blend, the test should be repeated with a blend containing decreased proportion of the high-cetane-number reference fuel. The test is continued in this manner until the knockmeter reading for the sample is definitely higher than one blend and lower than another blend of the reference fuels.

(I) Precision of Results

The cetane number should be reported in the nearest whole number to the exact rating as computed by interpolation from the knockmeter readings.

(J) Miscellaneous Notes

(1) *Clean* fuel must be used. It is suggested that the fuel be filtered through thin chamois leather into the fuel tanks.

(2) The fuel lines and fuel pump must be flushed thoroughly of air before starting the engine. After the engine is running, better results are obtained by switching quickly from one fuel to another *without flushing the fuel pump,* and allowing 5 min. for the change-over.

(3) When changing fuels in the tanks, it is very necessary to flush thoroughly the line to the switch valve until a solid fuel stream is obtained from the bleed drain.

(4) The fuel injection timing should be shown continually on the spark quadrant, and any deviation from 10 deg. before top dead-center must be corrected before each knockmeter reading is taken.

Appendix B

Tentative Standard Operating Conditions and Procedure for Critical-Compression-Ratio Method of Rating of Diesel Fuels.

OPERATING CONDITIONS

(1) Engine Speed	900, ±3 r.p.m.
(2) Cylinder	Standard motor-method variable-compression with shrouded inlet valve
(3) Piston	Cupped Diesel piston
(4) Jacket Temperature	Constant within ±1 deg. fahr., limits 205-212 deg. fahr.
(5) Cooling Liquid	Distilled water
(6) Inlet-Air Temperature	150 ±2 deg. fahr.
(7) Crankcase Lubricating Oil	S.A.E. 30
(8) Oil Pressure	25-30 lb. per sq. in.
(9) Valve Clearances	Intake 0.008 in. cold, Exhaust 0.010 in. cold
(10) Injection Advance	12 deg. before top dead-center (constant)
(11) Injection Pressure	1500, ±50 lb. per sq. in. (opening pressure)
(12) Fuel Quantity	18.0, ±0.5 cc. per min.
(13) Injection Period	2 injections per 60 engine revolutions
(14) Injector Specifications	Bosch DN30S3
(15) Injection-Pump Specifications	Bosch PE1B50A302/3S97, port closing at 0.075 in. ±0.005 in. lift from base circle
(16) Fuel Line – Tank to Pump	⅜-in. copper tubing
(17) Fuel Line – Pump to Injector	¼-in. outside diameter, ⅛-in. inside diameter
(18) Fuel Tank Height	25 in., ±1 in. from bottom of tank to pump inlet

PROCEDURE

(A) C.C.R. Determination

With the engine in equilibrium and operating conditions as specified previously, set the injection control lever to cause two injections per cycle. Open a quick-acting valve in the exhaust line and listen for audible combustion during the injection period. Reduce the compression ratio until no audible combustion occurs. After a 1-min. interval raise the compression ratio until combustion is again audible. Alternate compression-ratio settings are made at 1-min. intervals until fire and no-fire settings occur consistently 3 or 4 times within a range of 2 turns of the compression-ratio crank.

The critical compression ratio is considered to be half way between the ratio for audible combustion and the ratio for inaudible combustion.

(B) Cetane-Number Determination

To determine the cetane number of the fuel, find a pair of reference-fuel blends differing from each other by not more than 8 cetane numbers which will have critical compression ratios above and below the sample.

Calculate the cetane number of the sample by interpolation between the critical compression ratios of the reference fuels and sample.

Report the cetane number in terms of the nearest full number.

(C) Cetane Number from Calibrated Curve

A more rapid approximate method of obtaining cetane number is to determine the critical compression ratio of the sample. Then from a reference-fuel calibration curve select a reference fuel blend which should match the sample. Determine the critical compression ratio of this reference-fuel blend and, if it deviates from the value on the calibration curve, correct the cetane number of the sample, as determined from the curve, a corresponding amount.

(D) Precision of Results

The cetane number should be reported in the nearest whole number to the exact rating as computed by interpolation.

MISCELLANEOUS NOTES

(1) *Clean* fuel must be used. It is suggested that the fuel be filtered through thin chamois leather into the fuel tanks.

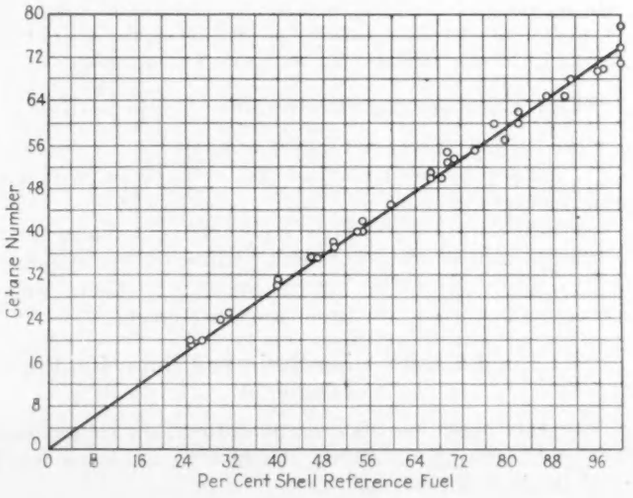
(2) The fuel lines and fuel pump must be flushed thoroughly of air before starting the engine. After the engine is running, better results are obtained by switching quickly from one fuel to another *without flushing the fuel pump,* and allowing 5 min. for the change-over.

(3) When changing fuels in the tanks, it is very necessary to flush thoroughly the line to the switch valve until a solid fuel stream is obtained from the bleed drain.

(4) See Part 4 of general C.F.R. instruction manual for other detailed information.

Appendix D

Calibration of Shell Secondary Reference Fuel Against Mixtures of Cetane and Methylnaphthalene.



Appendix C

Composition, Source, and Supplier of Cooperative Test Fuels

Sample No.	Composition	Source	Supplier
1	Straight-Run Gas Oil	Mid-Continent	Sinclair Refining Co.
2	Straight-Run Gas Oil	Mid-Continent	Sinclair Refining Co.
3	Sample No. 1 + 1 per cent Ethyl Nitrate	Mid-Continent	Standard Oil Co. (Ind.)
4	Cracked Distillate	Mid-Continent	Shell Petroleum Corp.
5	Sample No. 4 + 1 per cent Ethyl Nitrate	Mid-Continent	Shell Petroleum Corp.
6	Straight-Run Gas Oil	California	Standard Oil Co. of Calif.
7	Straight-Run Gas Oil	California	Standard Oil Co. of Calif.
8	Straight-Run Distillate	California	Standard Oil Co. of Calif.
9	Cracked Distillate	California	Standard Oil Co. of Calif.
10	Straight-Run Gas Oil	Pennsylvania	Socony-Vacuum Oil Co.
11	Straight-Run Gas Oil	Pennsylvania	Socony-Vacuum Oil Co.
12	Reference-Fuel Blend (67.4 per cent Shell High Reference; 32.6 per cent Methylnaphthalene)		Shell Petroleum Corp.

Average Physical Properties of Volunteer-Group Diesel Fuel Samples

	1	2	3	4	5	6	7	8	9	10	11	12
Gravity, deg. A.P.I.	38.5	32.8	38.2	33.7	33.3	35.9	28.2	38.9	30.6	38.3	35.6	29.5
Saybolt Universal Viscosity at 100 deg. fahr., sec.	36.4	50.4	36.2	34.7	34.6	36.3	51.6	32.7	35.2	43.0	51.3	34.7
Aniline Cloud Point, deg. fahr.	163	172	163	126	124	151	147	140	110	181	188	123
Flash (P.M.), deg. fahr.	168	213	130	185	141	176	218	156	189	151	240	149
Pour Point, deg. fahr.	0	+10	0	B-O	B-O	+15	+20	B-O	B-O	+15	+15	0
Sulphur, per cent	0.15	0.23	0.14	0.29	0.26	0.21	0.54	0.12	0.43	0.10	0.10	0.43
Initial Boiling Point, deg. fahr.	369	415	221	405	217	386	435	367	412	323	477	355
10 per cent Boiling Point, deg. fahr.	429	528	430	441	439	433	495	404	438	511	569	443
20 per cent Boiling Point, deg. fahr.	461	567	460	454	453	452	524	416	444	538	594	456
50 per cent Boiling Point, deg. fahr.	524	632	524	486	487	503	602	447	471	582	634	482
70 per cent Boiling Point, deg. fahr.	556	667	556	513	515	544	663	469	498	611	660	514
90 per cent Boiling Point, deg. fahr.	597	701	597	565	570	625	737	500	562	655	690	602
Final Boiling Point, deg. fahr.	640	725	642	637	643	708	741	539	655	708	721	689
Residue, per cent	1.3	1.5	1.3	1.4	1.3	1.4	6.2	1.1	1.4	1.4	1.5	1.5

Discussion

Calculated Ignition-Quality Factor Introduced

–W. H. Hubner

Automotive Engineer, Universal Oil Products Co.

IN the progress report of the Volunteer Group for Compression-Ignition Fuel Research, laboratory analyses and engine ratings are given for 12 cooperative test samples. The data represent the average of 8 to 14 laboratories.

With this information available, it was thought interesting to establish the relation between cetane number, as determined by both the delay and the critical-compression-ratio methods, and other expressions of ignition quality which are based entirely on the physical properties of the fuels, namely:

(1) Diesel index number.
(2) Viscosity-gravity number.
(3) Boiling-point-gravity number.
(4) UOP characterization factor.
(5) Ignition-quality number.

Although the first four of the preceding expressions have been described in detail in the literature,[a] they may be reviewed briefly as follows:

(1) Diesel index number: $D.I. = \frac{G \times A}{100}$

where: $D.I.$ = Diesel index number.
G = deg. A.P.I. gravity at 60 deg. fahr.
A = Aniline point in deg. fahr.

(2) Viscosity-gravity number:
$G = 1.082A - 0.0887 + (0.776 - 0.72A)$ (log log $(KV-4)$)
where: A = Viscosity-gravity number.
G = Specific gravity at 60 deg. fahr.
KV = Kinematic viscosity in millistokes at 100 deg. fahr.

(3) Boiling-point-gravity number:
$G = A + (68 - 0.703A) \log B.P.$
where: A = Boiling-point-gravity number.
G = deg. A.P.I. gravity at 60 deg. fahr.
$B.P.$ = 50 per cent distillation point in deg. cent.

(4) UOP characterization factor:
$K = \frac{(T_b)^{1/3}}{S}$
where: K = UOP characterization factor.
S = Specific gravity at 60 deg. fahr.
T_b = Molal average boiling point in deg. Rankine (in the case of Diesel fuels, the 50 per cent distillation point in degrees Rankine may be used with accuracy).

(5) Ignition-quality number:
$Q = \frac{G \times A \times B.P.}{100{,}000}$
where: Q = Ignition-quality number.
G = deg. A.P.I. gravity at 60 deg. fahr.
A = Aniline point in deg. fahr.
$B.P.$ = 50 per cent distillation point in deg. fahr.

The cetane ratings, calculated ignition-quality characteristics, and pertinent laboratory inspection data for each of the 12 cooperative test samples are listed in Table *A*.

The relation between cetane number and the various calculated expressions of ignition quality is shown graphically in Figs. *A*, *B*, and *C*. In the case of the high-viscosity Mid-Continent and the low-viscosity (highly volatile) California straight-run Diesel fuels, it will be noted that Diesel index number does not correlate so closely with cetane number as do the other calculated ignition-quality factors. In an attempt to allow for the apparent effect of volatility (boiling range and viscosity) and at the same time to develop a formula which would be simple to use (absence of cube

[a] See *National Petroleum News*, Vol. 28, Jan. 22, 1936, pp. 22, 24-26, and Jan. 29, 1936, pp. 25-28: "The Effect of Crude Source on Diesel Fuel Quality," by W. H. Hubner and G. B. Murphy.

Table *A* – Laboratory Inspection Data and Ignition-Quality Characteristics

Sample No.	1	2	3	4	5	6	7	8	9	10	11	12
Source	Mid-Cont.	Mid-Cont.	Mid-Cont.	Mid-Cont.	Mid-Cont.	Calif-ornia	Calif-ornia	Calif-ornia	Calif-ornia	Penn-sylvania	Penn-sylvania	Ref. fuel blend
Composition	Str.-Run	Str.-Run	No. 1 + 1% (a)	Crkd.	No. 4 + 1% (a)	Str.-Run	Str.-Run	Str.-Run	Crkd.	Str.-Run	Str.-Run	(b)
Gravity:												
a) °A.P.I.	38.5	32.8	38.2	33.7	33.3	35.9	28.2	38.9	30.6	38.3	35.6	29.5
b) Specific	.8324	.8612	.8338	.8565	.8586	.8453	.8860	.8304	.8729	.8333	.8468	.8789
Viscosity:												
a) S.S.U. @ 100°F.	36.4	50.4	36.2	34.7	34.6	36.3	51.6	32.7	35.2	43.0	51.3	34.7
b) Centistokes @ 100°F.*	2.87	7.52	2.79	2.22	2.19	2.83	7.88	1.43	2.42	5.19	7.79	2.22
50% Distillation Pt.:												
a) °F.	524	632	524	486	487	503	602	447	471	582	634	482
b) °C.	273	333	273	252	253	262	316	231	244	306	334	250
Aniline Pt., °F.	163	172	163	126	124	151	147	140	110	181	188	123
Diesel-Index No.	52.8	56.4	62.3	42.5	41.3	54.2	41.5	54.5	33.7	69.3	66.9	36.3
Viscosity-Gravity No.	.828	.835	.829	.858	.861	.841	.862	.850	.873	.813	.818	.881
Boiling-Point-Gravity No.	179.0	179.5	179.6	188.2	189.0	184.2	187.2	184.0	194.2	174.8	175.5	194.3
U.O.P. Characterization Factor	11.95	11.96	11.93	11.45	11.42	11.68	11.50	11.68	11.19	12.17	12.18	11.16
Ignition-Quality No.	32.9	35.7	32.6	20.6	20.1	27.3	25.0	24.3	15.9	40.4	42.4	17.5
Cetane Number**:												
a) Delay (Bouncing Pin)	54.5	55.0	68.0	41.0	54.5	47.5	43.0	43.0	34.5	62.0	63.0	50.0
b) Delay (All Methods)	55.0	55.0	68.0	41.5	55.5	48.0	43.0	44.0	34.0	62.0	62.0	50.0
c) C.C.R. (900 R.P.M.)	57.5	54.5	62.0	45.0	49.0	51.0	44.5	48.0	35.0	63.5	63.0	50.0

* Converted from S.S.U. @ 100°F.

** To nearest half number from Volunteer Group's average calibration curve for bouncing-pin delay method.

(a) Ethyl nitrate; (b) 67.4% Shell high reference fuel and 32.6% alpha-methylnaphthalene.

roots, logarithms, and so on), it was found that the Diesel index number multiplied by the 50 per cent distillation point gave surprisingly good correlation with cetane number, particularly delay-method cetane ratings. For lack of a more appropriate name, this combination of aniline point, gravity, and 50 per cent distillation point has been called ignition-quality number. Strangely enough, a straight-line relationship exists between it and cetane number.

We are not suggesting the substitution of ignition-quality number for any of the other calculated ignition-quality expressions; we are merely presenting the data for what they may be worth. The results do indicate, however, that better correlation between engine ratings and calculated ignition-quality expressions may be obtained by the proper selection and combination of physical properties.

Although a general relation is found to exist between cetane number and the various calculated ignition-quality expressions (except in the cases of doped fuels and reference-fuel blends), we do not advocate the use of calculated ignition-quality as a substitute for engine tests.

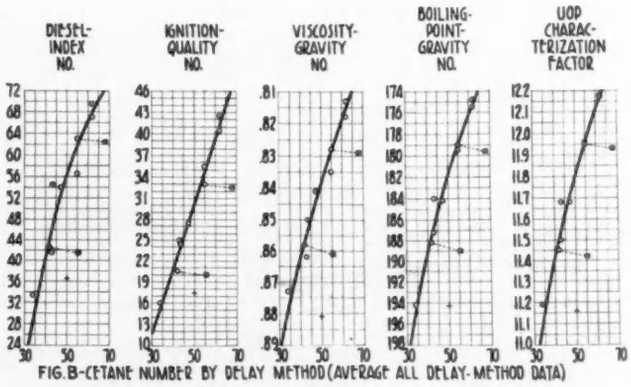


FIG. B-CETANE NUMBER BY DELAY METHOD (AVERAGE ALL DELAY-METHOD DATA)

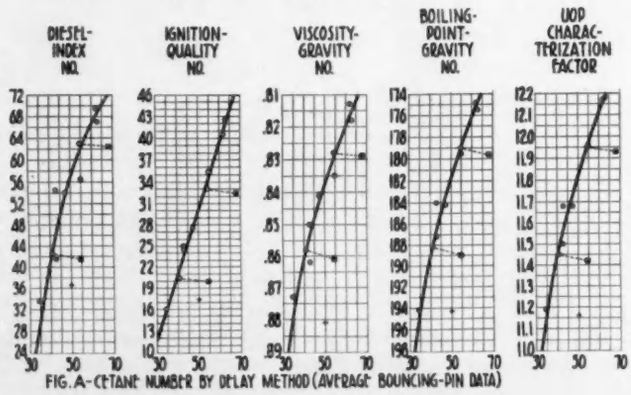


FIG. A-CETANE NUMBER BY DELAY METHOD (AVERAGE BOUNCING-PIN DATA)

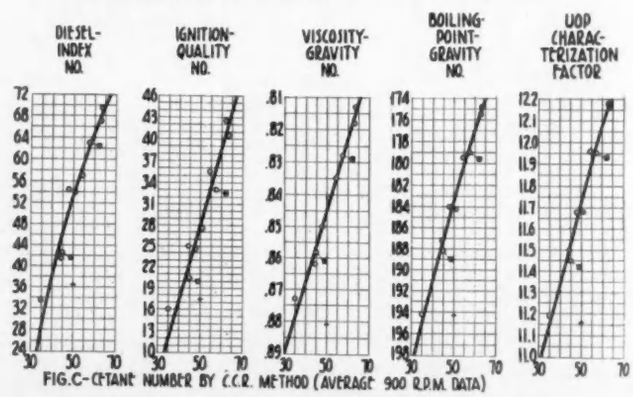


FIG. C-CETANE NUMBER BY C.C.R. METHOD (AVERAGE 900 R.P.M. DATA)

Figs. *A*, *B*, and *C* (Hubner Discussion) – Relation between Cetane Number and Calculated Ignition-Quality Factors

The Economics of Truck Selection

By Frederick K. Glynn
American Telephone and Telegraph Co.

THE truck manufacturers have made available to fleet operators a wealth of truck chassis with sufficient models and interchangeability of units to create special models to meet any transportation job requirements no matter how particular or peculiar.

Within reasonable limits, the first costs of these chassis are indicative of relative chassis strength, durability, and ability.

The selection of a chassis cannot be made from first cost or from operating cost expectancy alone, for these two go hand-in-hand to form the total cost and either may be increased with impunity if the overall cost of transporting the product is thereby decreased.

One of the most important requirements in the selection of a chassis is a thorough operation and transportation job-analysis.

Other considerations include availability of service, inherent safety, legal limitations, and appearance.

IN discussing our subject "Economics of Truck Selection" we will start with the chassis and then work into the principles governing selection and the economics involved.

Chassis Available

As to the chassis themselves, we find listed as of October, 1937, a total of 796 models manufactured by some 40-odd manufacturers. Of these models, 568 are four-wheel chassis with two wheels driven, 96 are of the four-wheel-driven type, and 132 are six-wheelers. The 796 models are labeled "basic." However, with the options offered, there are probably double or triple 796 models in current manufacture. This gives us, say, 2000 models available from the printed specifications.

Even beyond the printed specifications tremendously greater diversification of the unit make-up of truck chassis is possible and is used daily to produce the exact chassis for the particular job. This unorthodox chassis not found in specifications nor in advertising is largely the result of interchangeability of units. For example, one truck manufacturer advertises 20 chassis models based on 7 different engines, 8 rear-ends, 11 front-ends, 7 transmissions, 53 distinct wheelbases, 7 frames, and 33 gear ratios, and can make an almost unlimited number of combinations of these units. Another manufacturer shows 21 models made up from 9 engines, 12 rear-ends, 5 fronts, 6 transmissions, 10 wheelbases, 6 frames and 29 gear ratios.

Therefore, considering all of the combinations available, we find that practically any truck chassis manufacturer who makes a full line of capacities could produce all of the chassis which he has sold this year with no two chassis identical in every respect. Likewise, the total annual production of chassis this year, even including those manufacturers who produce large quantities in few sizes, could have been accomplished without any two truck chassis being identical in all respects. And remember, we have not discussed tires and springs as affecting the divergence in chassis make-up!

Therefore – with this wealth of standard models, options offered, and interchangeability of units – *I believe that we can take it for granted that chassis can be produced to meet any hauling or transportation job of any operator anywhere in the country. In fact, it is logical to believe that, of the 42 manufacturers shown for 1937, at least five of these manufacturers can compete in any price range with the exact specifications to meet any truck transportation job no matter how peculiar, particular, or specialized.*

Furthermore, with continuing cooperation between manufacturers and operators and continuing development by the manufacturers' engineers, it is reasonable to expect that tomorrow's chassis will better fit tomorrow's transportation problems.

Price Considerations

Having indicated an availability of product to meet any and all fleet operators' needs, let us bring "first cost" or price of the chassis into the discussion with an investigation of just what we get for the dollar used in purchasing a chassis, and attempt to discover the reasons for the difference in first cost of the so-called low-priced, medium-priced, and high-priced truck chassis. This attempt can best be illustrated by using the nominal 2½-ton, 14,000-16,000 lb. gross vehicle weight chassis as our example. This size of chassis seems to be offered the most universally as it is the upper limit for some light truck manufacturers and is the minimum of the strictly heavy truck builders. But, before referring to charts and specific data, it is essential to review critically the materials and operations necessary to manufacture a truck chassis.

As you know truck chassis are made from iron, steel, brass, copper, aluminum, rubber, and so on. These raw and semi-raw materials have a fairly small spread in cost, but undoubtedly these costs are affected by the quantities purchased. This difference, it should be noted, is between quantity manufacturers and small manufacturers and not as between low-, medium-, and high-priced chassis manufacturers. Any benefits in manufacturing cost to be derived from using a poorer quality of raw materials instead of the best have no place in the truck industry, for I firmly believe that all manufacturers

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try honestly to secure the best raw materials, regardless of the price range in which they place their chassis.

These raw materials, of course, must be cast, forged, drawn, pressed, machined, and so on, into parts, and here perhaps starts the major difference between the high-, medium-, and low-priced chassis. One manufacturing process may be cheaper than another, but not quite so productive of ultimate life in the part, for example, stamping in place of forging. Another difference could include the type and extent of heat-treatments, surface-finishing, and so on. Furthermore, in the assembly of the parts into units – fronts, engines, transmissions, rears, and so on – differing practices such as closer tolerances, balancing, and severity of inspection, can enter into the cost of the finished chassis, and it is conceivable that the foregoing points constitute some of the major differences in mechanical production costs among manufacturers in the different price ranges.

Final assembly cost should not be a major difference in the costs of manufacturers of the same quantity production but, of course, may favor large-quantity producers to some degree. On the other hand, I am told that it is less costly for the small-quantity manufacturers to make changes from standard to meet individual-fleet-job requirements, for their final assembly lines are the more flexible. Incidentally, these changes may involve "out-of-proportion" expense to any chassis manufacturer!

There is, however, one point missed in the foregoing discussion and it concerns relative size of the important load-carrying and wearing parts in the average chassis in one price range as compared with a chassis in the other price ranges. Size within reasonable limits means strength, wearing ability, and so on, and naturally represents weight, in the assembled chassis.

Another item of vital importance but of an entirely different character affecting the price of a chassis is that of the distribution of the chassis manufacturer's "overhead" cost both direct and indirect. The direct overhead, of course, comprises that necessary allocation to the price of each chassis of all fixed charges pertaining to the operation of the business of producing the chassis. The indirect overhead is composed essentially of the burden which must necessarily be included in the charges of purveying companies for the parts and units which they sell to the chassis manufacturer. For example, although the specialized quantity manufacturers of engines, transmissions, rears, frames, and so on, may have a normally lower overhead requirement per unit, this burden is transferred to the chassis manufacturer in addition to the profit charged by the specialty manufacturer. In addition to lower overhead per unit, it is also reasonable to expect that specialty manufacturers can produce at lower overall cost per unit than can the average chassis manufacturer. This result is due to higher production from a like investment in patterns, jigs, dies, fixtures, and machinery.

With all of the foregoing points in mind, and referring particularly to the chart of Fig. 1, we find that the 2½-ton chassis available on the market divide into three price ranges with distinct stratification when list prices are plotted against the cost per pound of chassis weight.

In studying this chart, note the clustering of the chassis in the medium-price range and the regular slope to a line drawn through the chassis on the left hand or minimum cents-per-pound side in this price range. A line drawn through the average point for each price range (marked *X* in Fig. 1) would just about parallel this line. Note also that the chassis in the high price range are scattered widely, but let us look at the underlying data of the chart before drawing any conclusions:

For 2½-Ton 14,000-16,000 Lb. G.V.W. Chassis	Low-Priced Chassis	Medium-Priced Chassis	High-Priced Chassis
Number of manufacturers	7	19	8
Number of models included	7	23	8
Quantity-production manufacturers[a]	7	9	4
Low-production manufacturers[b]	0	10	4
Manufacturers included in other price class[c]	3	3	2
Models — "own" engine	2	8	1
Models — "purchased" engine	5	15	7
Price range — total dollars	900/1150	1320/1890	2000/2860
Average price of chassis, total dollars	1043	1584	2317
Price range per lb. of chassis weight, cents	22.7/26.2	25.7/39.9	33.1/48.4
Average price per lb. of chassis, cents	24.0	32.6	40.5
Average gross vehicle weight, lb.	14,080	14,250	15,300
Average weight of chassis, lb.	4,350	4,860	5,735
Average net carrying capacity, lb.	9,730	10,390	9,565
Load carried per pound of chassis, lb.	2.24	2.14	1.68

[a] Listed by make in trade journals' compilations of "truck registrations."
[b] Production included in "miscellaneous registration" in trade journals.
[c] Manufacturers with "competitive" lines.

The salient facts from the chart and the data as they appeal to me are:

(1) The low-price range includes "quantity manufacturers" only. Is this condition because quantity production produces the low price or is it because purchasers demand and buy more low-price chassis, or both?

(2) Quantity and small-production manufacturers are about equally divided in numbers in the medium- and high-priced ranges.

(3) The wealth of models from which to select are in the medium-price range, and this condition seems logical.

(4) The majority of manufacturers of nominal 2½-ton chassis in each price range are buying their engines.

(5) The price per pound of chassis weight increases as the list price increases indicating that more than mere material costs are involved. This increase may involve, among other factors, differing price and trading structure but, for the purposes of this discussion, we must use list prices.

(6) Comparing the low- and medium-price ranges, the average chassis in the low-price range carries 660 lb. less and the chassis is 510 lb. lighter, making a gross difference of 1170 lb.; yet these chassis are rated to carry more per pound of chassis weight.

(7) Apparently, conservatism in the rating of a chassis increases with the price but, of course, gross-vehicle-weight data are the least reliable of those shown in the table as will be discussed later on.

(8) There is such a wide spread in the list prices, the price per pound, and the chassis weight for the chassis in one capacity group that the utmost care must be exercised to select the chassis best fitting the conditions, all things considered, and most economical in overall results.

Conclusion: *If weight of chassis is any criterion – and within reasonable limits it is – you get just what you pay for* (strength, durability, and perhaps ability) *in the present-day chassis,* and this view seems to be universal with fleet operators.

Operating-Cost Considerations

As to the economics of selection between price ranges practically all day-to-day costs may be affected to some degree.

Oil, vehicle service, supervision, and garaging are least, if any, affected by the price range of the chassis and are, therefore, eliminated from our consideration.

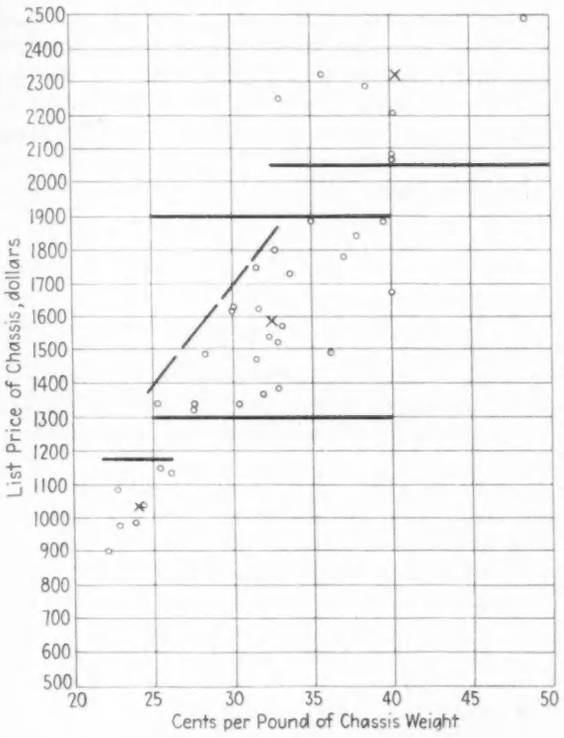


Fig. 1 – Chassis Price Vs. Chassis Weight – 2½-Ton, 14,000-16,000 Lb. G.V.W. Chassis

Slightly affected as we increase our investment in the chassis are insurance and interest, also license fees, gasoline and tires for, according to the increase in chassis weight going from one price range to another, so are we faced with paying the costs for supporting and moving a greater overall weight. This difference remains slight only when we have selected the right chassis and can be very important if we have "heavied-up" our chassis beyond reasonable needs.

For example, if we find upon careful analysis of the job requirements that a low-price-range chassis is sufficient except for an occasional moderate overload, it would seem uneconomical to select for the overload condition as we would be moving the added chassis weight 100 per cent of the time.

License fees vary in some states according to unladen weight and, in others, according to gross vehicle weight. However, in the final reckoning, these items are not controlling but rather in the nature of contributing to the overall economy.

The important items to be considered are the depreciation and repair-expense expectancies for each price range. Depreciation expense starts when we sign for the delivery of the chassis and continues inexorably until the chassis leaves our possession. The net charge for depreciation must keep pace with the service life, be it reckoned by years or miles, so that, when the chassis is spent, there will be funds available to replace it.

For the purpose of immediate discussion, we will consider depreciation on a yearly basis, and reference to the chart of Fig. 2 will indicate what we may expect in the three price ranges over the usual span of service life. These data are not from any one fleet but are the application of the experience of chassis manufacturers covering the age of trade-ins and the percentages of trade-in value to original cost in each of the three price ranges for 2½-ton chassis.

It will be noted that the low-price-range chassis starts for the second owner in the third year of life and that the majority have been traded by the end of the fifth year. The medium-price range chassis steps up a year with majority trades in the fourth, fifth and sixth year of service life, whereas the high-price chassis do not usually start for the second owner until the fifth year. For our purposes we are carrying the last range through the same span of years as for the two lower price ranges although trading experience of the chassis manufacturers indicates we might go further.

As to the depreciation dollars per year, each price range overlaps the next higher, but the lowest and the highest price ranges do not overlap.

In the foregoing comparison, general averages and experience with regard to 2½-ton chassis were used. A widely different size of chassis might possibly work out differently. *But,* when we jump from one price range to another in the 2½-ton size, we find that *depreciation increases with the first cost* and this relation will not be altered unless some abnormal condition exists between range of life, first cost, and salvage.

Repairs, on the other hand, can increase at a higher rate than the service life and, conversely, can be proportionately less for equal service the more we pay for a chassis. Whether or not the expected lower repair cost will offset the increased annual depreciation charges is a question which cannot be answered offhand but which can and must be answered in each specific case where the selection of a chassis from differing price ranges is involved.

Concerning the repair-cost expectancies for the chassis of each price range, it now seems impossible to find a fleet operator using chassis from each of the three price ranges on the same general class of work. This condition, in itself, indicates that chassis manufacturers and fleet operators are doing a much better job of fitting the chassis to the work. Furthermore, if I had been able to secure data for 1937 operation for all ages of chassis in each of the three price ranges, I doubt if it would prove very useful due to the constantly accelerated rate of development and improvement in truck chassis of the present decade.

At this point we might take time out to list hurriedly a few of these developments and improvements:

The move from four- to six-cylinder engines.
Three to four to five multiple-speed transmissions.
The wide front axle.
The full-floating rear axle.
The four-wheel brakes.
Solids to high pressure to balloon tires.
Interchangeability of units.

And – not strange to relate – the so-called "assembled chassis," of late condemned but now heartily accepted, led the way with practically all of these changes.

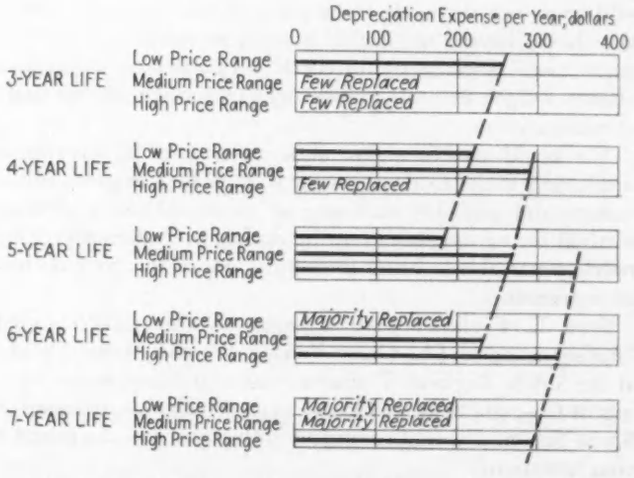


Fig. 2 – Depreciation Expense Per Year – 2½-Ton, 14,000-16,000 Lb. G.V.W. Chassis

Manufacturer's Service

Closely associated with the matter of repairs and an important factor in the selection of chassis is the availability and efficiency of manufacturers', distributors' or dealers' service shops. In fact, if two to five chassis manufacturers were competing on a proposition and bids were relatively close but not similar, it is quite conceivable that the highest bidder might take the business and properly so, if his service facilities were outstanding in the territory. Fewer fleets operate extensive shops nowadays, and all fleets need parts and units quickly available to minimize "out-of-service" for repairs.

Other Manufacturers' Qualifications

Assured financial stability of the chassis manufacturer is essential to guard against the difficulty of securing parts and units for an orphan chassis.

Uniformity of product is an important consideration in the selection of the make of chassis and is directly proportional to the severity of inspection of raw materials, parts, units, and assemblies both in the chassis plant and in the specialty manufacturers' plants. A rigid inspection system together with the financial courage to junk any assembly that is off color should go far in recommending a manufacturer.

A well-balanced, qualified, up-to-the-minute engineering department with a reputation for careful development of product and a willingness to cooperate is also a recommendation in the selection of the manufacturer.

Ability to deliver chassis as and when promised is a further desirable qualification in the selection of the manufacturer.

Chassis Specifications

In the selection of a chassis to fit a particular job, consideration must be given to the specific specifications and their effect on original, day-to-day, and overall job costs.

Tire sizes should be adequate and perhaps here is the place to err slightly on the heavy side, for the curve of tire wear vs. load carried is in favor of underloading of tires unless the mileage is abnormally low.

The load-carrying parts of the chassis such as frame, springs, front, rear, and so on, and the braking ability should be checked for strength equal to the job. Along these lines it has been suggested that the simplest method is to take the size of parts such as front-wheel spindle diameter at the inside bearing, frame section modulus, rear-axle shaft diameter – full floating type, clutch size, effective braking area, and so on, and express these sizes in terms of gross vehicle weight and price, comparing one chassis against another. This method will bring out definitely the weak spots and quickly establish the chassis having the largest number of maximums at minimum cost. Another useful check is to compute the ratio of chassis weight to gross vehicle weight for the chassis under consideration.

We could use formulas, slide rules, and all manner of measuring sticks to classify chassis according to gross vehicle weight and probably each one of us would use a different method for no one, not even the Society of Automotive Engineers, seems able to bring uniformity out of the present chaos of argument.

Fred L. Faulkner gave a paper on this subject called: "Rating a 14,000-Lb. Gross Vehicle Weight Motor Truck," at the S.A.E. Regional Transportation and Maintenance Meeting in Chicago, Sept. 29-Oct. 1, 1937, which was discussed by Pierre Schon. I would commend both paper and discussion to your attention.

Wheelbase – C.O.E. vs. orthodox design – has a job importance from a maneuvering and parking standpoint also certain inherent disadvantages. The selection of the wheelbase of either design determines to some degree the load distribution in service which, in turn, has a direct effect on the chassis maintenance costs – habitually too much weight on the front construction or on the rear construction and higher repair costs result. This consideration also involves the balance which should be preserved as between driving and steering traction.

Ability Factor

Engine power, gear ratios, and tire sizes bring in the factor of ability and, taken by and large, it seems ultimate economy to purchase ability in the chassis up to the point of diminishing returns on the overall transportation job. In other words, if chassis ability can be made to pay dividends in work produced at lower cost, do not penalize the job by underpowering the chassis.

For example, let us take two 2½-ton chassis from our previous chart (Fig. 1) and underlying data for convenience sake, even though it exaggerates the illustration, the one at our average investment for the high-price range and the other at the average for the medium-price range. The first chassis cost $2317 and the latter $1584 or, roughly, $2300 and $1600. Let us assume a life before obsolescence in this case of six years for both chassis, and a job requirement of 240,000 miles total for both or 20,000 miles per truck per year. This mileage we will consider to be in open territory, and we have the high-price truck costing us for depreciation six years times about $325 per year from our depreciation chart (Fig. 2), or $1950 and, in like manner, $1440 for the medium-price chassis. If we assume $3.00 and $3.25, respectively, per 100 miles for maintenance, and each travels a total of 120,000 miles, we have

	High-Price Chassis	Medium-Price Chassis
Depreciation	$1950	$1440
Maintenance	3600	3900
Total	$5550	$5340

Let us further assume that we purchased and paid for an ability in the high-price chassis to average 25 m.p.h. in open territory against an ability of 20 m.p.h. for the medium-price chassis. This ability affects work accomplished by the driver whose wages we will figure at $1 per hr.

The high-price truck of greater ability will require 4800 hr. to do its mileage, whereas the medium-price truck will require 6000 hr. This time equated into driver's wages is $4800 and $6000 and, when added to depreciation and maintenance, brings the total comparable cost to $10,350 for the high-price chassis and $11,340 for our medium-price chassis and we, therefore, should buy the greater ability for the open-territory job.

If these chassis were to be used in congested territory where ability could not produce a higher mileage in a given time, then the medium-price chassis should be purchased.

These figures, as mentioned heretofore, exaggerate the illustration but, regardless of this exaggeration, the fleet operator must use the same arithmetic to assure himself that he is purchasing an ability in the chassis which will result in ultimate overall economy in the particular transportation job.

Ability factor, as you know, is under consideration from a legislative standpoint and, if you are not entirely familiar with this situation, I would suggest reading the survey currently appearing in *Commercial Car Journal*.

Analysis of the Specific Job

In the foregoing we have hinted time and again that the selection of a chassis depends upon the specific transportation

job and that each territory, job, or class of work might require differences in the chassis. Now let us go on and consider this as an item in itself – in fact, it is one of the most important items in the entire discussion.

In many scattered fleets a given operating center will be found to contain a heterogeneous collection of miscellaneous types and classes of chassis performing the same job, whereas the districting of the trucks by make, perhaps, and certainly by the type best suited to the job is desirable. In this matter the approach to the problem, whether the fleet is scattered or concentrated, is that of determining proper size of chassis in relation to predetermined work factors as found from job-analysis.

To establish a definite job-analysis, operating statistics which will inform the fleet operator in usable detail as to the intensity of use of the equipment, that is, work factor, are vital. A thorough knowledge of the geography of the territory, traffic movement, road surfaces, points of congestion, type, size and frequency of loads, and so on, is also necessary in making a job analysis.

Let us illustrate these important matters of proper assignment, of intensity of use, and so on, by using the petroleum industry which is the country's largest truck-owning industry.

One of the fundamental objectives of these fleet operators, and this is true, of course, in all industries, should be that of reducing the size of the fleet, that is, capital investment, to such absolute minimum as will be measured by the point of diminishing the quality of service to the company and its customers. This consideration naturally implies the necessity of extracting the last ounce of economic usefulness from each piece of equipment. Each vehicle must work as long and as effectively per day and per year as is possible. Only by the establishment of a technique of job-analysis befitting the individual company within a given industry can the work factor be improved through proper selection of chassis and the number of vehicles reduced.

Thus in the transportation of gasoline, for example, statistics should be maintained for each vehicle which will permit usable analysis of the intensity of use – gallons delivered per mile run, gallons delivered per hour, total hours worked per day, each day of the week, total hours which could be worked, average miles per hour running time which is translated into driver cost, time consumed in loading and unloading, size of average delivery, number of non-productive stops, load factor or relationship between size of body or tank and average load returned, and so on.

Suppose, for illustration, that there are two small distributing centers. In Station *A* there are two 1000-gal. trucks with an average total of 120,000 gal. per month to deliver, 60,000 gal. for each truck. The load factor averages 70 per cent or, in other words, each truck delivers only 700 gal. out of the potential 1000-gal. load per trip. Analysis of customer buying habits, customer locations, and so on, disclose that a 95 per cent load factor is attainable. Then, in assuming that the present average round trip is 30 miles at an average speed of 20 m.p.h., we derive a monthly mileage at present of about 5150 total for two trucks and drivers' driving time of about 260 hr. Assuming truck cost including both fixed and variable charges to be 10 cents per mile and driver $1 per hr., we, therefore, establish a total monthly trucking cost of about $770, or about 6½ mills per gal. delivered. On the basis, however, of the 95 per cent load factor, the cost per month becomes, roughly, $570 and the cost per gal. about 4¾ mills, a reduction of about 26 per cent.

In Station *B* let us assume one 1500-gal. truck with a total average monthly gallonage of about 70,000. For the sake of simplicity, it will be assumed that, as the case of Station *A*, the round trip mileage is 30; the average speed, 20 m.p.h.; the driver wage, $1; the load factor, 95 per cent; and truck cost per mile, 12 cents. Thus, the cost per month will be about $250 for the truck and driver, again excluding driver cost during loading and unloading operations, or just over 3½ mills per gal.

However, the preceding cost per gallon in Station *B* is not a true cost as the 1500-gal. truck runs only 1475 miles per month. Assuming the purchase price of the new vehicle was $6500 complete and that it is depreciated on a mileage basis of 175,000 total miles before the eighth year to take care of the obsolescence factor, the depreciation per mile is, therefore, $0.037 and the necessary minimum number of miles per month 2085. However, as stated previously, monthly mileage is only 1475 or 610 miles less than minimum depreciation mileage. Thus, 610 times $0.037 or about $22.60 represents a charge-off of depreciation for unused mileage. Therefore, the true cost per gal. amounts to almost 4 mills, an increase of over 10 per cent.

The preceding illustrates a problem frequently mishandled. There are, of course, several possible solutions. In this example it can be shown that the 1500-gal. Station *B* truck can do the whole job for Station *A*, exceeding its depreciation quota of monthly miles by 440 and showing a final cost per gallon of just over 3½ mills or a saving to Station *A* of about 25 per cent, based on Station *A* increasing its load factor to 95 per cent. Analysis of customer buying habits disclosed a maximum of 7½-hr. operation on peak days using the two 1000-gal. trucks and 10 hr. using the one 1500-gal. truck. Total possible time was 11 hr.

I will not elaborate this example endlessly by discussing appropriate action to take in Station *B*, and so on, but I have cited these figures by way of illustration of the effect of operating job-analysis studies upon overall transportation cost as concerned in the selection of the size and number of chassis. Let it suffice to call particular attention to the steps which were taken in these illustrations to increase the intensity of use of equipment and, at the same time, to derive the greatest return from capital investment as illustrated by depreciation in connection with Station *B*.

Experimental Selections

As mentioned previously, the operator, while making job studies followed by "playing checkers" with the existing fleet in the interests of increased efficiency, must not lose sight of the importance of districting vehicles by similar types as far as possible. Likewise, his purchase policies should keep this vital point in mind. Despite the reasonableness of this practice, the diversity of make and type of chassis in some fleets might lead to the belief that all were purchased for experimental purposes to determine the right chassis. On the other hand, some fleets adhere so tightly to standardization and their fleets are so uniform that the question arises whether they are right in their selection.

Neither fleet would seem to be on solid ground. The former needs a degree of standardization and should be in a position to do so, whereas the latter is vulnerable and should buy "border-line chassis" of more or less strength, durability, and ability as the case may be, from time to time for experimental operation to prove its standardization.

Selection for Replacement

It follows from the foregoing that chassis seldom should be selected and purchased blindly like kind for like on a replacement basis, for operation and job analysis frequently will lead to the purchase of a unit more ideally fitted to the work to be performed than was the worn-out unit to be discarded.

The size and other characteristics of the new unit may or may not be the same as the old unit; in fact, such studies frequently will indicate the necessity for a "reshuffling of the deck" which will obviate the necessity of purchase altogether.

First Cost Vs. Operating Cost

There are some fleet operators who are prone to close their eyes to first cost but who are highly interested in operating costs, maintenance labor, material, tires, gasoline and oil consumption. Conversely, some fleet operators seem to be more vitally concerned with first cost than with the operating expense. The accountant divides his books into fixed and controllable expenses. The wise operator, however, will never be literal in his interpretation of the word "fixed." Also some operators are too prone to create elaborate budgeting supervision over controllable expenses and to block out a capital budget solely upon the usual and obvious considerations of life expectancy and replacement in kind of worn-out vehicles, and so on.

Furthermore, "fixed" expenses are *not* fixed and perhaps some illustrations will help to bear out this point. The fixed expense, depreciation, taxes, interest, insurance, licenses, and overhead allocations of a newly purchased vehicle which is wrapped in cellophane and left in the garage may cost from $500 to $1000 per year or more depending, of course, on investment. If, because of poor planning, lack of operation and job studies, this truck only works half its proper mileage based on obsolescence calculations, there still remains a highly expensive annual or monthly charge-off for non-productive capital investment.

Normal operating expenses per unit of product hauled or work done may be less expensive on larger higher priced vehicles. Not always, however. The intelligent approach to this problem is obviously that of making detailed operating studies resulting in the setting up of operating and investment standards. Such studies must be followed by an "out-of-pocket" conception which usually will result in the fleet operator lumping capital outlay with estimated operating expenses based on past experience, thereby deriving a figure from which may be computed *the total cost of transportation over the life of the vehicle.* This is what really counts, and so it also follows that the collection of genuinely useful operating cost and investment statistics is of extreme importance – not "statistics for statistics' sake" as is so often the case.

In your studies of this "out-of-pocket" cost do not lose sight of the fact, as stressed before, that the service rendered by the vehicle is the all-important consideration for, in some cases, higher motor vehicle costs may produce a cheaper overall job.

Economies of Bulk Purchases

In previous paragraphs we have considered price structure, price range, quantity production, and so on as affecting our purchasing dollar; we have advised a full knowledge of what, when, and how to select; we have discussed job-analysis; and lastly we have prepared an estimate of capital and expense. Therefore, if it can be anticipated that the fleet will need 5, 10, 15, or 20 of XYZ chassis during the year, why not give the benefit of advance information of this quantity to the chassis manufacturer who, in turn, can return some of his quantity purchase savings to the fleet even though the delivery of the chassis may be singly throughout the entire year to eliminate peak loads of preparation for purchase, of preparation for operation after delivery and peak loads of financing.

Miscellaneous Considerations

Safety on the road begins with the selection of the chassis and involves items discussed in the foregoing such as adequate power, efficient braking, sufficient load-carrying ability, and so on.

Legal requirements must be known and observed in the selection of a chassis, or the job may be penalized by legal limitation of the use of the truck.

Appearance is important as a truck is a moving signboard with the firm's name advertised for all to see. Also, good appearance has a beneficial psychological effect on the employees handling the equipment.

Effect on public sentiment enters into selection, that is, poking up a hill, poor brakes, speeding down hill, and so on.

Likelihood of premature obsolescence must be given consideration particularly with respect to unusual design, changing legal requirements, changing job requirements, and so on.

There are times when it is uneconomical to purchase a chassis at all and when a truck should be rented to do a certain job, particularly if the work is of short duration or seasonal. On the other hand, it sometimes pays dividends to buy a chassis, use it intensely, and sell it at the completion of the job.

Summary

(1) There are a wealth of chassis available and adaptable to meet today's transportation problems.

(2) Within reasonable limits, the price paid for a chassis seems indicative of its relative strength and durability.

(3) First cost and operating costs go hand-in-hand and pay dividends in work produced only when the proper chassis is selected.

(4) Chassis abilities both tractive and load-carrying are important considerations in the selection of a chassis.

(5) Complete operation and transportation job-analysis is essential to correct selection of chassis.

(6) The transportation job determines the chassis specifications.

(7) Other important considerations in the selection of chassis include: availability of parts and service, qualifications of the manufacturer, inherent safety of design, likelihood of early obsolescence, and legal requirements.

Grateful acknowledgment for assistance is hereby given to T. L. Preble, O. A. Axelson, W. F. Banks, Clinton Brettell, Fred L. Faulkner, J. Willard Lord, and John M. Orr.

Developments in Diesel Engines

THE necessity for the Diesel engine was the need for a prime mover which could utilize efficiently the wide range of oils lying between gasoline and the heavier residual oils, such as road oil and asphalt. Now, in the constant effort of designers to get more and more out of their engines per cubic inch of displacement, a trend toward higher speeds is a perfectly natural development.

However, high-speed Diesel engines as we know them today are still in their infancy and, if such developments are to achieve ultimate success, the original reason for the development of the Diesel engine must not be forgotten, that is, the design of the engine must be such that it will burn cleanly and completely, without objectionable exhaust smell, the majority fuels lying within the range of gasoline and reasonably heavy residual oils.

This point becomes more and more important as the developments advance, since higher-speed engines are unavoidably more sensitive to abuse by unsuitable fuels.

Excerpts from the paper: "Fuels for High-Speed Diesel Engines," by T. B. Rendel, Shell Petroleum Corp., presented at the Kansas City Section Meeting of the Society, Kansas City, Mo., April 23, 1937.

Manufacture, Characteristics and Uses of Magnesium Castings

By W. G. Harvey
American Magnesium Corp.

OUTSTANDING characteristics of magnesium castings are light weight and excellent machinability. Production has increased from about 10,000 lb. in 1925 to approximately 800,000 lb. in 1936. Differences in foundry methods from those practiced for other common metals are pointed out. The mechanical properties are specified, and inspection methods are described. Suggestions are given for proper design of castings.

Machining methods and tool design suitable for obtaining the smooth finish, the speeds, and economies that are possible for magnesium-alloy castings are described. Careful painting under approved methods is recommended. Applications of magnesium castings for many parts of aircraft engines, for parts in the fuselage, and in such accessories as aircraft landing wheels and starters are listed, along with other typical applications.

THE commercial manufacture of magnesium sand castings may be said to have begun in 1925, in which year approximately 10,000 lb. were made. Development work had been in progress for about five years previous to this date, during which period methods of casting were perfected and the possible market investigated. The problems were not easy, and disappointments in the early years were many. Although casting methods for other metals were well known, they were of little value, for magnesium differed from other metals in two important characteristics – density and chemical activity. The lighter weight of the metal required the development of gating and pouring methods which would insure successful casting of commercially thin sections, and its reaction with water vapor required special sand addition agents. Proper methods of melting and pouring had to be worked out to produce castings free from flux and oxidation products if satisfactory stability and mechanical properties were to be obtained.

The inherent lightness of magnesium was believed to be of value to industry and, based on the correctness of this assumption, the problems were attacked and solved. As an indication of the success of the casting processes and also of the value of this light metal in industry, the following figures will be of interest:

Production of Magnesium Alloy Castings in United States by Years[1]

Year	Lb.
1928	55,861
1929	116,350
1930	99,443
1931	127,398
1932	132,049
1933	165,599
1934	284,419
1935	375,625
1936	791,859

Manufacture

Foundry methods for magnesium sand castings have been described adequately in current literature[2] and will be only briefly mentioned. The metal generally is melted in gas- or oil-fired steel pots of 100- to 400-lb. capacity. A charge consists of approximately 97 per cent of metal and 3 per cent of a salt flux which, by excluding the air, prevents the oxidation of the metal and also serves the very important function of refining and cleansing the molten metal from non-metallic impurities. The main portion lies at the bottom of the pot under the molten metal but a thin film floats on the metal surface. This flux is of such a nature that it gradually thickens and can be pushed aside easily when the metal is to be poured.

An inspection of a magnesium sand foundry would seem to indicate procedures very much like those in use for aluminum and other nonferrous metals. There are, however, certain important differences which are necessary because of the lighter weight and chemical characteristics of the metal. Magnesium, at around its melting point of 1204 deg. fahr., reacts with water vapor, and castings cannot be made successfully in the green sand molds employed by most foundries. Inhibitors or sand addition agents must be incorporated in the sand to prevent this reaction. These agents are of various types and comprise such materials as powdered sulphur, boric acid, ethylene glycol, ammonium fluoride, or combinations of these. Some are more effective with certain types of castings than others. Only a small amount of the inhibitor is necessary, and its use allows sufficient water to be added to the sand to furnish satisfactory molding qualities. These chemicals, in effect, produce inert atmospheres in the mold and protect the metal while it is in the molten condition. Because of the gases produced, a molding sand of high permeability is required. For

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 8, 1937.]

[1] U. S. Bureau of Mines.

[2] See Transactions of the American Foundrymen's Association, June, 1936, pp. 591-614; "Founding Magnesium Alloys," by J. A. Gann and M. E. Brooks.

the best results this sand is prepared synthetically by mixing a properly graded silica sand with bentonite. Core sand is of the same general character containing, in addition, the requisite binder.

Another difference in foundry procedure has to do with the lightness and low specific heat of magnesium alloy and the lower hydrostatic pressure which is available to force the molten metal into the thinnest cavities of the mold. This deficiency is overcome by the employment of a larger number of gates and risers, making all of these higher and of greater cross-section than in ordinary foundry practice. These provisions do produce the required result and furnish a flexible means of making even thin-sectioned and intricate castings, but they have their effect on cost through increase in the ratio of metal melted and poured to casting weight. In addition, the trimming cost is proportionally increased. These differences in foundry technique, together with alloy considerations, will indicate the importance of making magnesium castings in a separate foundry and the difficulty of combining these operations with the casting of aluminum alloys.

Alloys and Mechanical Properties

Pure metals seldom make good castings, and magnesium is no exception. The requirements of a good casting alloy are: (1) good casting characteristics, (2) suitable mechanical properties, and (3) satisfactory stability. To these requirements might be added (4) the ability to have its mechanical properties improved through heat-treatment. Although magnesium alloys freely with most of the common metals, it has been found that minor amounts of aluminum and zinc or combinations of these metals in magnesium meet all the preceding requirements to a satisfactory degree and, consequently, the standard casting alloys in use today contain 85 per cent or more of magnesium with varying amounts of aluminum or aluminum and zinc. All these standard alloys also contain approximately 0.1 to 0.3 per cent of manganese. This component improves the stability and appears to benefit the yield strength. The following table gives the nominal composition of these standard alloys:

Nominal Composition of Magnesium Casting Alloys, per cent

Alloy Number	Al.	Zn.	Mn.	Mg.
AM240	10.0		0.2	Balance
AM265	6.0	3.0	0.2	Balance
AM246	12.0		0.1	Balance
AM241	8.0		0.3	Balance

All of the alloys listed are amenable to heat-treatment with improvement in mechanical properties, and all except AM241 also can be aged. The mechanism of heat-treatment, properly called solution heat-treatment, consists of causing the inter-metallic compounds to go into solid solution by holding the castings for a period of time at an elevated temperature followed by quenching. This corresponds to the T4 condition. The times and temperatures vary for each composition. A further change in the mechanical properties is obtained by precipitating the solute constituents to a predetermined extent. This is termed "aging" and is generally indicated by the symbol T6. It is accomplished by holding the heat-treated castings at about 350 deg. fahr. for a period of time. In general, the tensile strength and elongation are improved by heat-treatment and the yield strength and hardness by aging.

Typical and specified minimum mechanical properties for the standard magnesium casting alloys are given in Table 1 as determined on separately cast test bars.

Patterns

The shrinkage of the casting alloys is about $^{11}/_{64}$ in per ft. This value compares so closely to the $^{5}/_{32}$ in. per ft. used for aluminum alloys that pattern equipment built for the latter can be used satisfactorily. Cases will arise where slight changes of either a temporary or permanent nature may be necessary to allow for the different foundry characteristics of the lighter metal. Compared generally, magnesium alloy cannot be cast in as light sections as can the aluminum alloys. As a guide in design it can be stated that $^{5}/_{32}$ in. is the thinnest section which can be cast commercially in magnesium unless the areas of thinner sections are small or so positioned as to allow adequate gating of these thin sections. The commercial casting tolerance on dimensions is similar to that maintained for aluminum castings.

Design Considerations

On the last pages of this paper applications for magnesium castings will be mentioned. From these applications it will be evident that, in the majority of cases, the reason for their use is that of weight-saving. In other words, the lighter metal has been adopted only after further reduction in section thickness has not been possible in castings made from the heavier metals. Although the mechanical properties of magnesium castings compare favorably with those of aluminum, it is of great importance that the design characteristics of the magnesium alloys be understood fully. Magnesium cast alloys, especially in the heat-treated form, exhibit excellent tensile properties and fatigue resistance but are sensitive to conditions producing local stress concentrations. They are satisfactory for applications demanding high fatigue strength if care is taken to avoid sharp corners and abrupt changes in section thicknesses. Portions of a casting that are highly stressed should be provided with liberal fillets blended into the body of the casting. The edges of cored holes and webs should be beaded. Stud bosses should be of ample size, and good radii should be provided at their bases. The length of stud threads should be 2½ to 3 times the diameter for those carrying high stresses. When magnesium-alloy castings replace aluminum, increases in wall sections seldom are required. On the other hand, when they replace malleable iron or steel, a redesign of the important sections is generally necessary. The redesign may vary from a thickening of ribs and the increasing of fillets to a general thickening of all sections. Even if the volume be doubled, which would be an extreme case, the resultant magnesium casting would have only one-half the weight of the original. The elastic modulus of magnesium is but two-thirds that of aluminum. In castings, this characteristic is seldom of importance for the wall sections are generally of ample thickness for foundry reasons.

Although magnesium alloy is not recommended as a bearing metal, there are many situations in which it will operate satisfactorily. For instance, steel shafts may be run directly in magnesium castings where ample lubrication is provided and where the bearing loads and speeds are not excessive. Where this is not feasible, iron or steel inserts can be cast in the magnesium body. Magnesium does not alloy with iron or steel, and many castings are made in which iron inserts or steel pipes are cast in place. Fig. 1 shows these inserts in pneumatic tool castings.

Soundness and freedom from harmful porosity can be obtained in the castings to a marked degree. It is seldom necessary to impregnate a casting to seal it, and pressure tests of 100 lb. per sq. in. are a part of many specifications. It follows from this discussion that rejections during the machining operation should be low, and experience proves this to be the case.

Although minimum specified properties often are used in the design of a particular casting, it must be remembered that these minimum values are based on the properties obtained from separately cast test bars. A sand casting with its many variations in wall thickness, different cooling rates, and other considerations of this nature, cannot be expected to duplicate in all of its sections the conditions obtaining in separately cast bars. However, magnesium sand castings do differ from castings of most metals in that there are means of controlling the grain size and eliminating, to a large degree, internal porosity. Pin-hole porosity, at times so troublesome in castings showing its presence on machined surfaces, does not occur. Microshrinkage affects the mechanical properties in proportion to its occurrence. Fortunately its amount and location in a casting are subject to some control and, when the amount is held within normal limits, the properties are but little affected.

It has become standard practice with some concerns to mark on their blueprints the magnitude and direction of maximum nominal stress to which the casting is subjected in service. These stress analyses are of value in that they make possible the choice of a foundry practice which will produce the optimum properties in those sections which will be subjected to maximum stresses in service. Static breakdown tests can be made on castings through the section or sections involved, thus checking the effectiveness of the foundry methods used.

The strength of engineering materials at both low and high temperatures is important for certain applications. In aircraft, for instance, it is necessary to know what effect the low temperatures of high altitudes will have on the strength of magnesium castings. The results on alloys AM241T4 and AM265C at —112 deg. fahr. are given in the following table with properties at room temperature for comparison.

Alloy	Temperature, deg. fahr.	Tensile Strength, lb. per sq. in.	Elongation, per cent in 2 in.
AM241T4	75	31,890	10.0
AM241T4	—112	30,880	6.25
AM265C	75	26,500	5.0
AM265C	—112	25,550	4.5

It is evident from the preceding table that these alloys are not affected seriously by low temperatures and, inasmuch as these compositions are typical of the casting alloys in general, it can be assumed that magnesium castings are satisfactory for any low temperatures which might be encountered in aircraft travel.

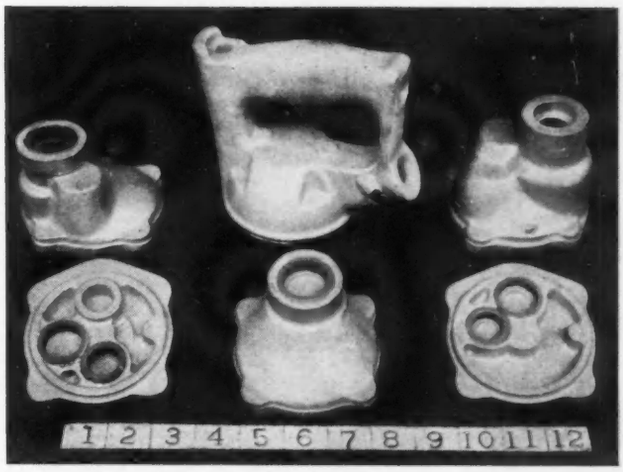


Fig. 1 – Portable Pneumatic-Tool Parts Showing Cast-In Steel Inserts – Alloy AM265C

Higher temperatures than normal may also be encountered. Investigations indicate that the tensile and yield strengths decrease and the elongation increases with rise in temperature. These changes are not serious up to 400 deg. fahr.; for use at higher temperatures magnesium castings are not recommended. Our present information indicates that the following factors modifying the standard properties should be used for design.

Design Stress Factors for Magnesium Castings for High Temperatures

Condition	Designation	Factor 200 deg. fahr.	Factor 300 deg. fahr.	Factor 400 deg. fahr.
As Cast	C	0.92	0.80	0.65
Heat-Treated	T4	0.92	0.80	0.65
Heat-Treated and Aged	T6	0.90	0.60	0.40

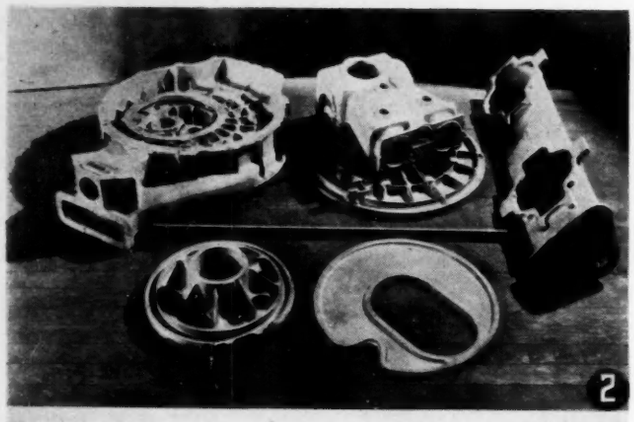
Inspection Methods

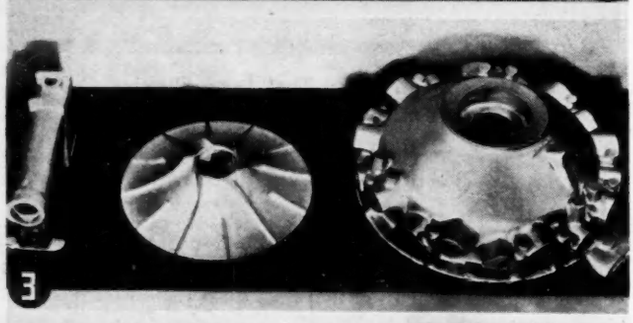
Although it is evident that an increasing number of magnesium castings are being used for other purposes, the per-

Table 1 – Mechanical Properties of Magnesium Cast Alloys as Determined on Separately Cast Test Bars

Alloy	Designation	Tensile Strength, lb. per sq. in. Typical	Tensile Strength, lb. per sq. in. Specified Minimum	Yield Strength, lb. per sq. in. Typical	Yield Strength, lb. per sq. in. Specified Minimum	Elongation, per cent in 2 in. Typical	Elongation, per cent in 2 in. Specified Minimum	Charpy Impact Value (keyhole notch), ft.-lb.	Endurance Limit (500 Million Reversals), lb. per sq. in.	Brinell Hardness (500-Kg. Load 10-mm. ball)
AM240	C	20,000	18,000	11,000	10,000	2.0	1.0	0.6	8,000	52
	T4	35,000	29,000	12,000	10,000	9.0	6.0	2.0	11,000	52
	T6	35,000	29,000	16,000	15,000	4.0	2.0	1.0	8,000	60
AM265	C	27,000	24,000	11,000	10,000	6.0	4.0	1.0	11,000	48
	T4	35,000	30,000	12,000	10,000	9.0	6.0	2.5	11,000	51
	T6	37,000	32,000	18,000	16,000	4.0	2.5	1.1	9,000	69
AM246	C	19,000	17,000	14,000	11,000	0.5	none	0.4	6,000	65
	T6	32,000	27,000	20,000	17,000	0.5	none	0.5	7,000	85
AM241	C	24,000	21,000	11,000	9,000	5.0	3.5	0.6	7,500	45
	T4	33,000	29,000	11,000	9,000	10.0	6.0	2.2	7,500	48

C designates as cast; T4 designates heat-treated; T6 designates heat-treated and aged.
Yield Strength is taken as the stress which produces a permanent set of 0.2 per cent (A.S.T.M. Specification E8-36).





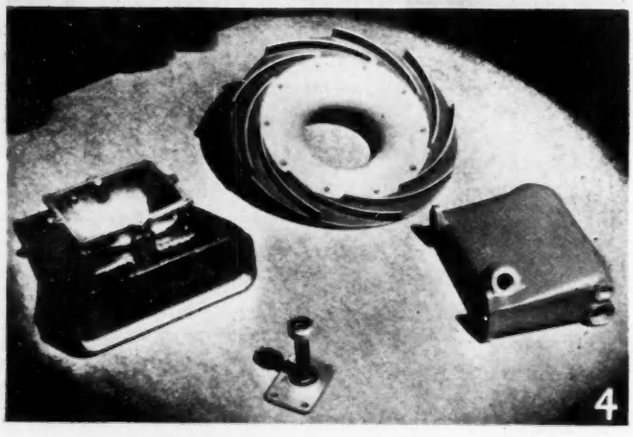


Fig. 2 – Aircraft-Engine Castings

Fig. 3 – Nose Section and Impeller for Aircraft Engine

Fig. 4 – Diffuser, Intake, and Oil Sump for Aircraft Engine

centage entering aircraft structures is still in the majority and this fact is responsible for the type and carefulness of the inspection methods employed. Routine foundry inspection involves an examination of the castings for visual defects such as shrink cracks, misruns, sand holes, and dimensional faults. All of these defects are fairly obvious, and rejections for these causes come within the normal scrap allowance of any foundry. It is of greater importance to guard against the production of castings which have internal defects and to change the foundry practice to correct the trouble. This type of inspection requires frequent fracturing of the castings in all sections and generally leads to a system of gating and chilling which will produce the maximum soundness. It is frequent practice to subject new castings to complete fracturing before production runs are made. Occasional static breakdown tests are made to determine the overall strength of the casting.

The examination of magnesium-alloy castings by means of X-radiographs is of value for detecting gross defects, such as blowholes and shrinkage cavities. The radiograph, however, is of little aid in showing the presence of the extremely fine dispersed shrinkage, sometimes termed micro-shrinkage. It is possible to reveal this micro-shrinkage in thin plates by means of X-rays but, because of the ease with which they penetrate the low-density alloys, the contrast on the photographic film between solid metal and slightly unsound areas is not very great.

Consequently, castings with heavy sections or of complicated design may contain areas of micro-shrinkage that do not show up on the radiographic film.

Machining

Magnesium castings have excellent machining characteristics. They do not drag or tear, can be cut at high speeds and feeds, and take a fine smooth finish. These features, together with their general freedom from internal porosity, lead to many economies. A saving of 15 to 35 per cent (including idle time for tool travel, mounting, and demounting) can, with proper tools, be obtained over the time for machining aluminum.

Compared with machining brass, cast iron, and steel, the economies are even greater. Magnesium cuts so easily that standard tools used for other metals may be employed. For maximum economy, however, these tools should be of slightly different design. The principles which affect these changes are the low cutting resistance, the low heat capacity of the metal, and the large volume of chips produced by the easy and rapid cutting. The low cutting resistance leads to high speeds and heavy feeds, and a tool design is necessary that will dispose of the chips rapidly and smoothly without jamming and developing frictional heat. Under any frictional conditions, the low heat capacity of the metal will produce higher temperatures in tool, chips, and work than with other metals. Several tool manufacturers are prepared to furnish the proper designs but, when these designs are not available, the following basic factors of tool design should be kept in mind: (1) smooth tool faces, (2) small areas of tool in contact with work, (3) large chip spaces (and, in milling cutters, small number of blades), and (4) small rake angles (undercuts).

It is important to maintain the cutting edges as sharp and smooth as possible and free from grinding-wheel scratches. In addition to rapid cutting this precaution aids in chip flow. These principles apply in general to all operations such as turning, milling, drilling, reaming, and tapping, and specific detailed directions are available from the producers of castings.

Because of the ease of cutting and the small amount of frictional heat generated with sharp and properly designed tools, magnesium castings usually can be machined without lubrication but, under certain conditions, the use of a lubricant may be desirable. This is the case, for instance, when a very smooth finish is required or when a temperature rise resulting from rapid cutting or to the shape of the work may cause distortion. Under the same conditions, a lubricant or coolant is advisable to prevent ignition of fine chips. In such cases, a lubricant with the lowest water content should be selected. Kerosene or mixtures of equal parts of kerosene and lard oil are recommended. Good standard cutting compounds are also suitable.

The fire hazard in machining magnesium is slight and, if proper tools and methods are used, ignition of the turnings can be avoided. To ignite the cast alloy, it is necessary to reach the melting point which varies between 1100 and 1200 deg. fahr. In rough-cutting operations it is seldom possible to attain these temperatures. In finishing operations, the turnings are finer and, under adverse conditions, the ignition temperature may be reached. These adverse conditions are the use of very fine feeds, dull or chipped tools, small clearances,

inadequate chip space, and high cutting speeds. Coarse feeds, deeper cuts, and lower speeds are advisable but, in those cases where fine feeds and cuts and the highest speeds are necessary for quality and economy, a lubricant should be used.

Chips which have become ignited may be extinguished easily by covering them with dry cast-iron turnings. It is always recommended that a supply of these turnings be placed at each piece of machining equipment. Dry sand is equally effective but its use is harmful to the equipment.

Stability

Magnesium alloys in general are resistant to alkalies, fluorides, chromic and hydrofluoric acid, but are attacked by mineral and organic acids and by salt solutions. They are attacked easily by chloride solutions. The rate of attack varies to some extent with the alloys in use today. Exposure to inland conditions causes only a darkening as a result of oxidation of the surface of the unprotected metal. This film, although porous, is adherent and retards further attack.

Iron, the most widely used of all the metals, exposed under the same conditions, experiences loss of section by scaling. That even this rapid rusting of iron and steel is a commonplace phenomenon and does not prevent their use, leads to the conclusion that both iron and magnesium require some protection against severe atmospheric conditions.

Experiments are indicating that, except in hot and humid atmospheres, the alloys do not corrode as seriously as formerly expected, even when exposed to sea air and spray. Painted metal panels exposed within 100 ft. of the Atlantic Ocean at Pt. Judith, R. I., show scarcely any attack after three years Actual service experience checks these results and indicates that the resistance is greater than might be expected from salt-spray tests. When the temperature and humidity increase, the rate of attack is accelerated. It is not often possible to predict where magnesium products will be used and, consequently, safe practice dictates that magnesium castings should be protected with properly applied paints of an approved type which will protect the surface even under the most severe conditions. From the standpoint of appearance it is often desirable to paint castings even for inland use.

A number of chemical coatings have been developed in recent years which afford considerable protection to the metal surfaces. They produce thin but adherent chemical films. Their resistance to attack is only a part of their value for, through a microscopic etching of the surface, they afford an excellent base for paints. The acid-dichromate coating is most extensively used. It is applied by dipping the casting for a minute or two in a solution containing nitric acid and sodium dichromate. It gives a characteristic yellow color to the metal surface. Other coatings applied either chemically or electrolytically may be employed.

Best protection will be obtained by carefully following an approved painting system. The castings should be freed from grease or oil by the use of some solvent such as gasoline, carbon tetrachloride, or one of the alkaline cleaning solutions. Following this cleaning they should be dipped in the dichromate-nitric acid solution, washed with hot water, dried and primed with a zinc-chromate primer. Primers of the type covered by Navy Specification P-27 give excellent results. At least two finish coats should then be applied. The type of finish paint depends on the service to be encountered. For the most severe service the oil-base phenolic-resin (bakelite) enamels or an oil-base phenolic-resin varnish pigmented with Albron paste is recommended. These types can be baked if considered necessary at temperatures up to 250 deg. fahr. and, for the best results, both the primer and finish coats should be baked for one hour at this temperature. Baking improves adhesion and resistance to moisture penetration. Inspection of the paint coats at intervals while in service is important if the best service is desired.

Uses

The magnesium alloys used in castings have a density of about 1.8. They have, therefore, but two-thirds the weight of aluminum and less than one-quarter the weight of brass or malleable iron. This characteristic is the basis for practically all of the present-day applications. To a greater extent than ever industry is striving to decrease the weight of portable equipment and to eliminate dead weight in parts of moving machinery. The aircraft industry early perceived the possibility of decreasing the weight of engine castings and of many parts in the fuselage through the use of light-weight castings. In fact, it would probably be a more accurate statement to say that the growth and demands of the aircraft industry are what led to the development of the magnesium industry as we have it today. It was the interest in magnesium castings by aircraft-engine manufacturers that justified the early development work.

Aircraft uses constitute approximately 70 per cent of the present production. On engines magnesium-alloy is used for crankcase sections, blowers, accessory housings, oil-pump housings, oil sumps, manifolds, supercharger casings, starter housings, and gear cases, to mention only the larger and more important parts. Some of these applications are shown in Figs. 2 to 5, inclusive. Although the castings used in the fuselage are of smaller size they make up for it in the variety of parts involved. These parts are comprised of cockpit fittings, pedals, air scoops, condulets, control boxes and levers, brackets of many kinds, handles, door hinges, landing-gear apparatus, foot rests, pulleys, quadrants, and many other parts of which

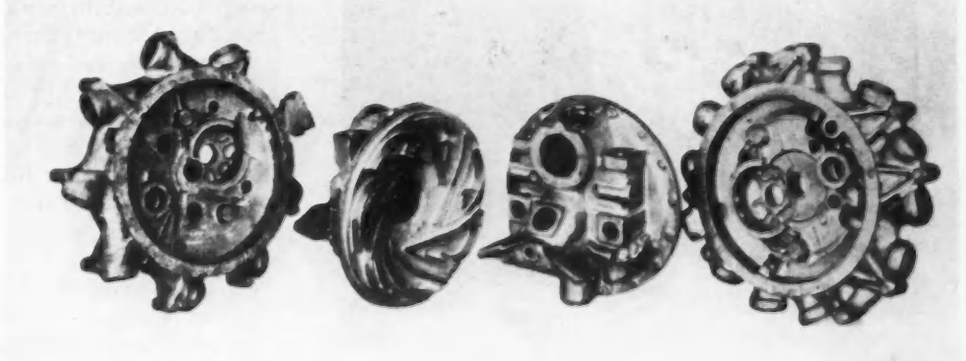


Fig. 5 – Blower and Rear Accessory Drive Case (Front and Rear) for Aircraft Radial Engine – Alloy AM240-T4 – Weights, 30 and 21 Lb. Respectively

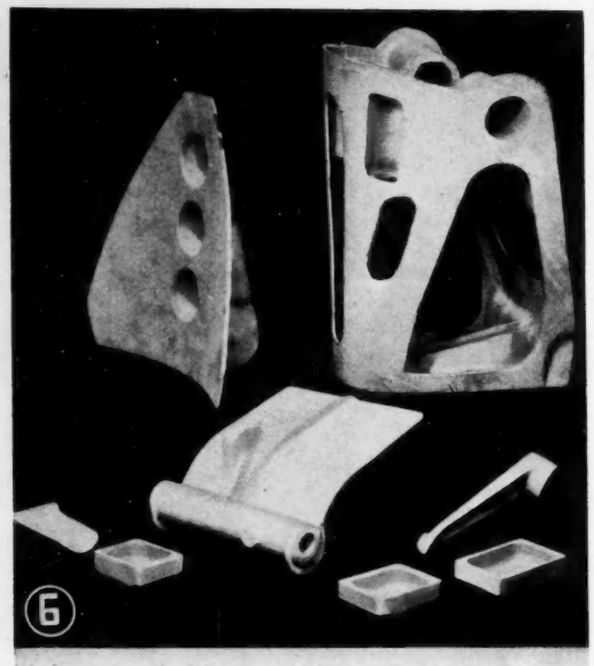


Fig. 6 – Castings for Airplane Fuselage Parts – Alloys AM265T6 and AM240T4

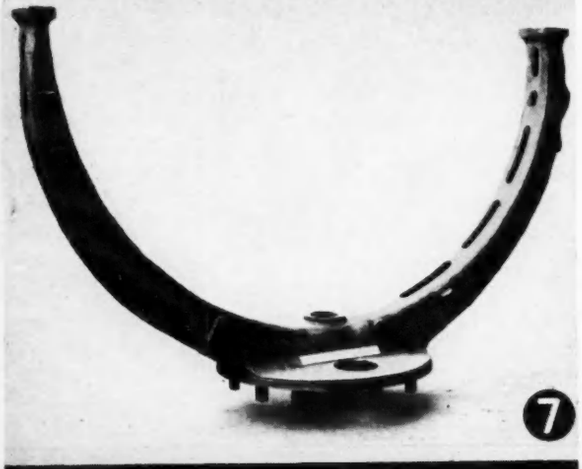


Fig. 7 – Trunnion for Searchlight – Alloy AM240T4 – Weight, 112 Lb.



Fig. 8 – Hub for 65-In. Aircraft Landing Wheel – Alloy AM240T4 – Weight, 188 Lb.

those mentioned will be representative. Several of these castings are illustrated in Figs. 6, 7 and 8. Aircraft landing and tail wheels are an important aircraft application. These magnesium wheel castings are used in many sizes from the small 7-in. up to and including some recently made in 65-in. size.

In industries outside of aircraft, magnesium castings are

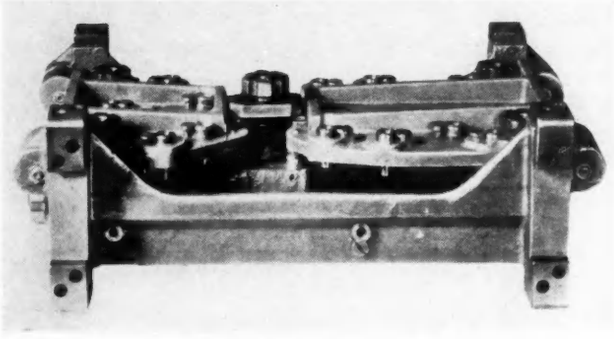


Fig. 9 – Assembled Machine-Shop Jig Employing Three Magnesium Castings – Alloy AM241T4 – Weights, 36, 14, and 11 Lb.

finding many uses where light weight is important. Examples are portable parts of pneumatic and electrical grinding and drilling tools, bread-slicing machine frames; parts of cigarette, envelope-folding, book-binding, and printing machinery; for motion-picture equipment and sound cabinets; for foundry flasks, patterns, match plates and core boxes; for jigs and fixtures in machining operations; and for certain parts of trucks, buses, and coaches. Fig. 9 shows a machine-shop jig assembly. It should not be necessary to burden this paper with other examples, as the foregoing should indicate the trend. Magnesium castings are more expensive to manufacture than castings of other metals, and their use must be justified by the weight-saving and machining characteristics. The cost trend, however, is in a downward direction, which should tend to expand their use.

Engine Installations

THE installation of an engine in an airplane, as practiced at present, is still somewhat distant from an exact science. Practically every branch of engineering is touched upon to a greater or lesser extent in the design of the powerplant installation, and the work involved for a complete mechanical analysis of the many problems faced would entail an excessive amount of time and expense. Also, most of the results of such computations would be of questionable value due to our lack of satisfactory information for the original assumptions. Consequently we are faced with the necessity of designing a great deal of the equipment from previous experience, for instance, as to cooling, vibration, local airflow, and so on, rather than a rational analysis such as would be employed in designing a wing spar or fitting where definite factors are known.

Excerpts from the paper of the same title by Ivar L. Shogran, Douglas Aircraft Co., presented at the Southern California Section Meeting of the Society, Los Angeles, Calif., April 30, 1937.

Flight-Testing with an Engine Torque Indicator

By A. Lewis MacClain and Richard S. Buck
Pratt & Whitney Aircraft, Division of United Aircraft Corp.

A TORQUE indicator, in which the reaction forces of the fixed gear in the propeller reduction gearing are measured, has been developed and flight-tested by Pratt & Whitney Aircraft Division of United Aircraft Corp.

In this device the reaction forces are balanced by two hydraulic pistons, thus producing an oil pressure under these pistons which is proportional to the engine torque. The pressure of the oil, measured by an ordinary pressure gage in the cockpit, in conjunction with the engine speed and a suitable factor, gives the actual brake horsepower developed by the engine under all operating conditions of positive torque. Using a constant-speed propeller, it has been possible to make complete engine calibrations of power versus manifold pressures at various engine speeds at several altitudes between sea level and 20,000 ft.

Fuel consumptions, induction-system temperatures and pressures at several points from the carbureter inlet to the intake port, cylinder temperatures, airplane speeds, miles per gallon, and other data were obtained in flight. The effect of mixture strength on power, airplane speed, cruising range, and on general engine operation has been evaluated. The differences in air speed and engine power between an uncowled engine with short exhaust stacks and the same engine with cowling and collector are shown. Very illuminating results are shown as to the effects of cowl flap opening on engine power, airplane speed, drag coefficient, and fuel consumption in miles per gallon although the data on the study of range under cruising conditions are meager at present.

The torque indicator has shown itself to be very free from trouble during 150 hr. of flying and is an aid to the pilot in regulating mixture strength, in checking ignition, adjusting carbureter preheat, and, primarily in controlling the engine power. The added weight is about 20 lb.

DETERMINATION of engine power in flight has been a much-debated subject. Due to variations in atmospheric conditions, the power of engines in flight as evaluated from altitude, manifold pressure, engine speed, and carburetor or manifold temperature by means of sea-level dynamometer calibrations, cannot always be correlated with day-to-day performances of carefully calibrated propellers and airplanes. The need for measuring engine power in flight has been felt for a great many years, and various means have been tried with more or less success. Many of these attempts were not successful because the apparatus used was complicated and the accuracy of readings obtained was doubtful.

There has been developed and flight-tested by the Pratt & Whitney Aircraft Division of United Aircraft Corp., a comparatively simple torque indicator that measures accurately the reaction forces of the fixed gear in the propeller reduction gearing. Using a constant-speed propeller, it has been possible to make complete engine power calibrations at several altitudes at various engine speeds and throttle openings in flight between sea-level and 20,000 ft. Also, a number of factors influencing airplane and engine performance have been investigated to a small extent. It is the object of this paper to describe this torque indicator, and to present results of tests made under flight conditions.

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 7, 1937.]

The Pratt & Whitney Torque Indicator

The type of reduction gearing used on Pratt & Whitney Aircraft engines provides an easy means of measuring engine torque. The reaction forces of the fixed gear are balanced by two hydraulic pistons, thus producing an oil pressure under these pistons which is proportional to the engine torque. The pressure of the oil, measured by an ordinary pressure gage in the cockpit, in conjunction with the engine speed and a suitable factor, gives the actual brake horsepower developed by the engine under all operating conditions of positive torque. Fig. 1 is a diagram showing the principle used in the torque indicator, the right-hand view showing a cross-section of a conventional Pratt & Whitney reduction gearing. The large internal gear (2) is splined to the front end of the crankshaft (1) and meshes with the pinions (3). These pinions

turn on shafts (4) which are fixed to the flange (5) of the propeller shaft (6). The pinions mesh with the fixed gear (7) and thus cause the propeller shaft to rotate at a slower speed than does the crankshaft. The fixed gear (7) is attached to a torque arm (8) instead of being fastened to the reduction-gear housing in the usual way. As shown in the left-hand view, this torque arm is connected through piston rods (9) to the torque-reaction pistons (10). These pistons float in cylinders (11), and the torque reaction is balanced by hydraulic pressure in the clearance space (12). Any convenient number of pistons may be used but, in the present model, two pistons are incorporated. The Pratt & Whitney Aircraft torque indicator uses the engine oil for its operating fluid. The small amount of oil lost through leakage is replenished by a small booster pump that maintains a higher oil pressure in the oil line (13) than is required in the cylinders to balance the maximum torque that the engine will develop. If, by leakage, the oil in the clearance space (12) escapes, the piston (10) and the valve (14) move to the left, uncovering port (16). This action allows oil to flow into the space (12) and through the oil passage (17), until the pistons push the torque arm (8) back to its normal position. In doing so one piston moves valve (14), automatically closing off the supply. Consequently the oil in the two cylinders is always at the proper volume and at equal pressure, and this pressure is developed by and proportional to the engine torque. A branch in the connecting oil passage is led to the pressure gage in the cockpit. In order to prevent air from accumulating inside the cylinders, adjustable bleeds (not shown) are provided, one in each cylinder, so that there is a positive oil flow through the mechanism at all times.

The first torque indicator (Fig. 2) was built up by attaching the torque cylinders to the outside of a stock reduction-gear housing by means of brackets. This device proved very successful, and it was used in the flight tests described herein. From the result of experience obtained on this first model a revised torque indicator was constructed and is now being used for flight tests on an R-1830 engine. The operating principle has not been changed, the main difference being that the revised model incorporates built-in cylinders and oil passages, and a small high-pressure oil pump with relief valve for maintaining the required amount of oil in the cylinders. As seen from the photographs (Figs. 3 and 4) the cylinders project very little from the external surface of the reduction-gear housing. Fig. 5 shows all of the internal parts of the torque indicator and illustrates the simplicity of the device. The torque indicator as built adds less than 20 lb. to the weight of the engine. Once the device has been assembled there is no need for further adjustment, and its accuracy can be maintained over a considerable period.

It was found, upon dynamometer calibration of the two torque indicators, that the torque obtained from the calculated constants agrees with that obtained from the dynamometer within a maximum error of ± 0.50 per cent for each of the two indicators. Fig. 6 shows the calibration of one torque indicator and illustrates their general accuracy. Due to the design of the device no calibration is actually required. The horsepower of an engine can be obtained easily from the formula:

$$\text{B.Hp} = (K)\ (P_Q)\ (\text{R.P.M.})$$

where P_Q = torque indicator gage reading in lbs. per sq. in., and K is a constant depending upon the reduction gear ratio and dimensions of the parts of the torque indicator.

The flight tests were made on a moderately supercharged Twin Wasp Junior engine equipped with the torque indicator and a Hamilton Standard constant-speed propeller and installed in a Vought Corsair airplane. No cowling was used for the major portion of the test as shown in Fig. 7, although the engine was equipped with pressure baffles. Short exhaust stacks, with one cylinder exhausting through the hot spot, were used to maintain back pressure as near as possible to atmospheric. The carbureter air intake was of generous dimensions and was carried well forward under the engine to keep the carbureter air temperature as near as possible to atmospheric to prevent undue heating of the air. A valve was installed in the air duct for the purpose of adjusting the carbureter inlet pressure to simulate the condition of no ram.

Measured and Predicted Performance

The present method of predicting engine power at altitude generally depends upon sea-level dynamometer calibrations

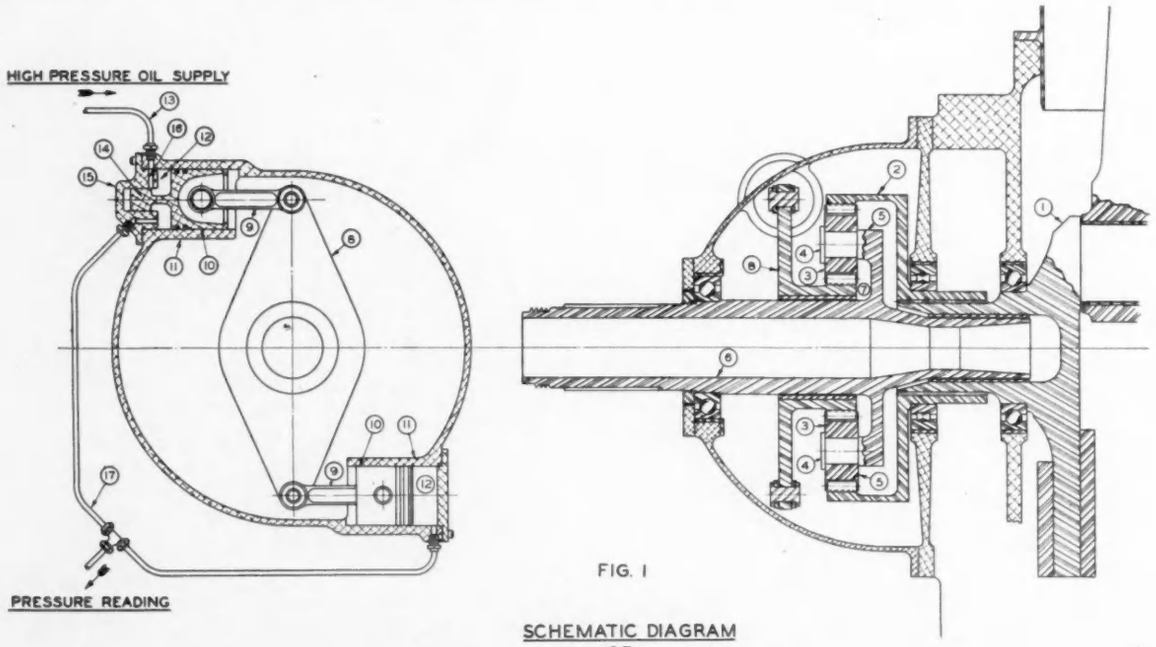


FIG. 1

SCHEMATIC DIAGRAM
OF
PRATT & WHITNEY TORQUE INDICATOR

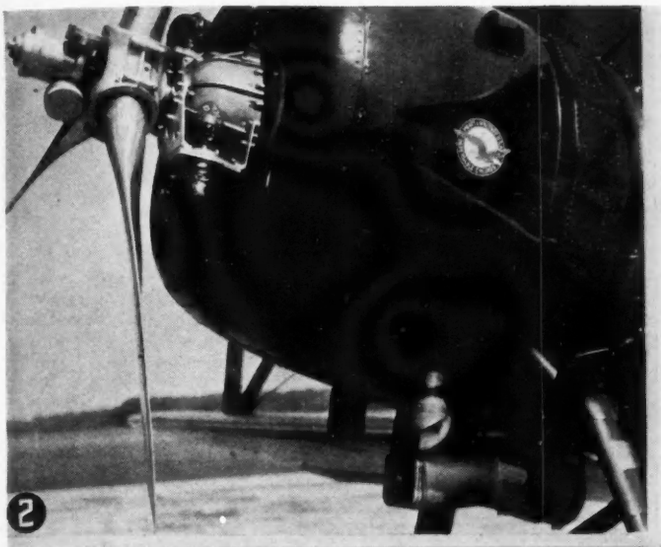


Fig. 2 – First Torque Indicator – Installed on a Twin Wasp Junior Engine

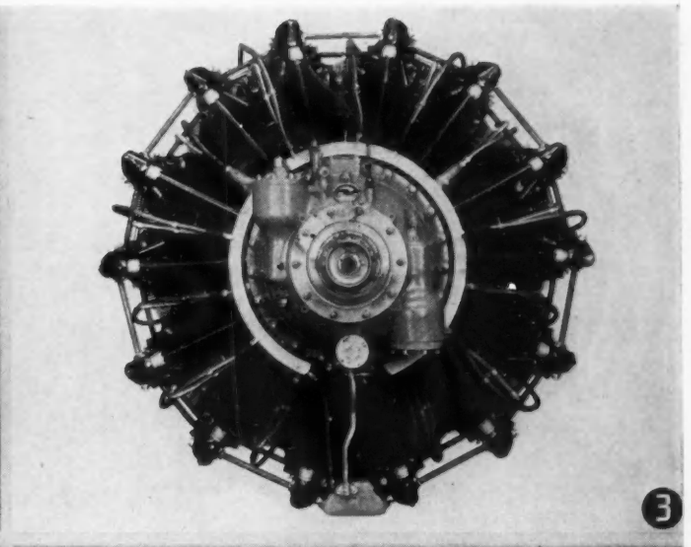


Fig. 3 – Twin Wasp Engine with a Torque Indicator Built Into the Reduction-Gear Housing

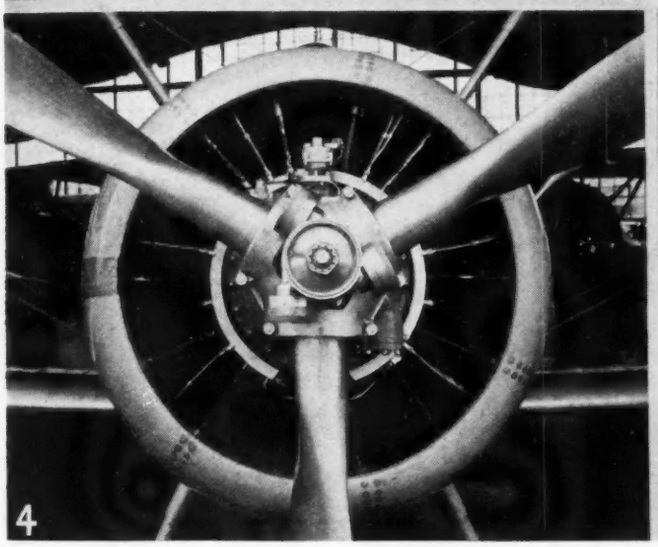


Fig. 4 – Twin Wasp Engine with Torque Indicator Built Into the Reduction-Gear Housing – The Propeller Speed Control Is More Prominent than the Indicator Cylinders Which Are Almost Hidden by the Propeller Hub

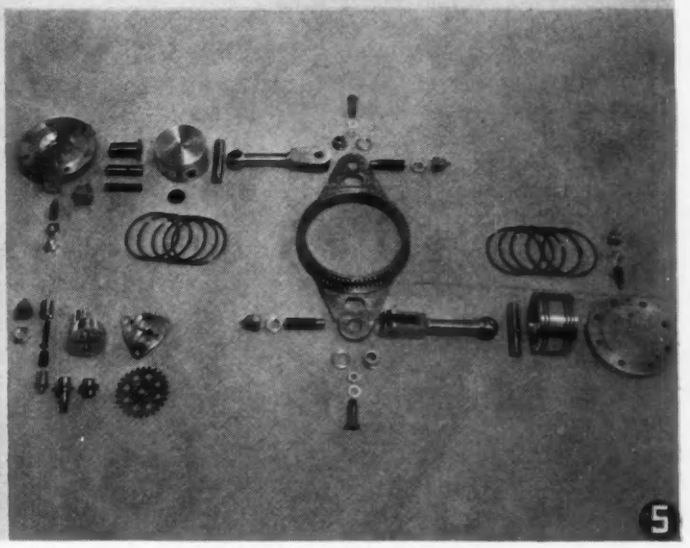


Fig. 5 – Internal Parts of the Pratt & Whitney Torque Indicator – The Booster Oil Pump Is the Lower Left Group of Parts

carefully corrected to the sea-level conditions of the standard atmosphere as established by the National Advisory Committee for Aeronautics – an ideal atmosphere that contains no moisture. Using this arbitrary sea-level power as a basis, the engine performance at altitude is calculated and predicted with suitable allowances for differences in the standard conditions between sea level and altitude. These allowances have been developed from experience in the various altitude laboratories and from a meager amount of flight-testing with power-measuring devices. The curves of power at altitude along with the sea-level calibrations are used in determining the actual power of an engine in flight from readings of several instruments that measure various factors that affect the power. The more factors that are taken into consideration, the closer can the true power be determined. It is customary to use altitude, manifold pressure, engine speed, and carbureter-air-temperature ratio as the determining factors for power. As a rule, no allowances are made for the effects of humidity, of cooling-air temperature, nor for changes in manifold pressure due to the change in carbureter-air-temperature, and numerous other minor factors.

In actual flight at a given pressure altitude, the atmospheric temperature may deviate considerably from standard and, for this reason, a determination of power will not agree with the predicted power for that altitude with standard conditions unless suitable corrections are applied. The correction usually made is based on the assumption that power varies inversely as the square root of the air-temperature ratio at the carbureter. This assumption was developed from experience with liquid-cooled engines maintaining substantially constant temperature in the cooling medium. It is doubtful if the power of an air-cooled engine, subjected to the vagaries of atmospheric temperatures, is adequately corrected by the carbureter-air-temperature ratio alone.

With the torque indicator, as with sea-level dynamometer tests, the observed power varies in some manner with atmospheric conditions of pressure, temperature, and humidity. The torque indicator in flight has an advantage over the ground dynamometer in that the pressure variable is controlled easily or eliminated by flying at a predetermined pressure altitude. This elimination leaves the effects of humidity and temperature for which corrections are necessary in order to obtain power for standard conditions. As is well known, the effect

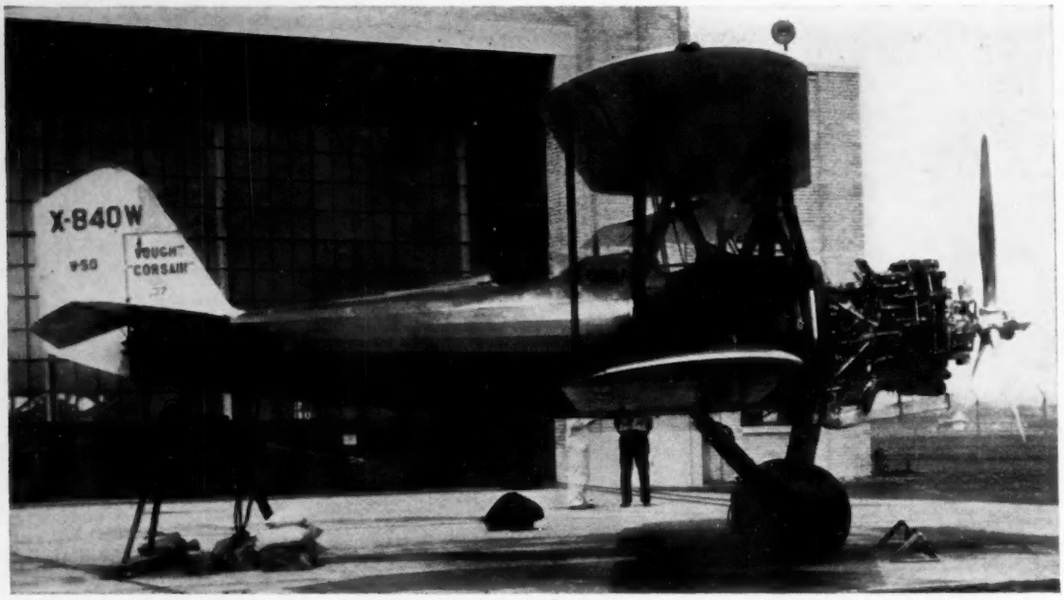


Fig. 7 – Twin Wasp Junior Engine with First Torque Indicator as Installed for the Tests Without Cowling – The Airplane Is Set Up for Static Thrust Measurements – The Cable Leads to Scales Some Distance Back from the Airplane

of humidity becomes negligible with temperature values approaching freezing.

The curves in Fig. 8 show a comparison of measured and predicted horsepower. The measured horsepower was obtained from the torque indicator during the flight tests, and is corrected to standard temperature conditions at altitude by the conventional correction factor for carbureter-air-temperature:

$$\text{B.Hp.}_{standard} = \text{B.Hp.}_{observed} \sqrt{\frac{(\text{Carbureter Air Temperature})_{observed}}{\text{Air Temperature}_{standard}}}$$

The scattering of the points, particularly at 10,000 ft., indicates a deficiency in the conventional method of correcting observed power to standard conditions. The altitude performance was predicted in the usual manner from sea-level dynamometer calibrations of the identical engine used in these tests.

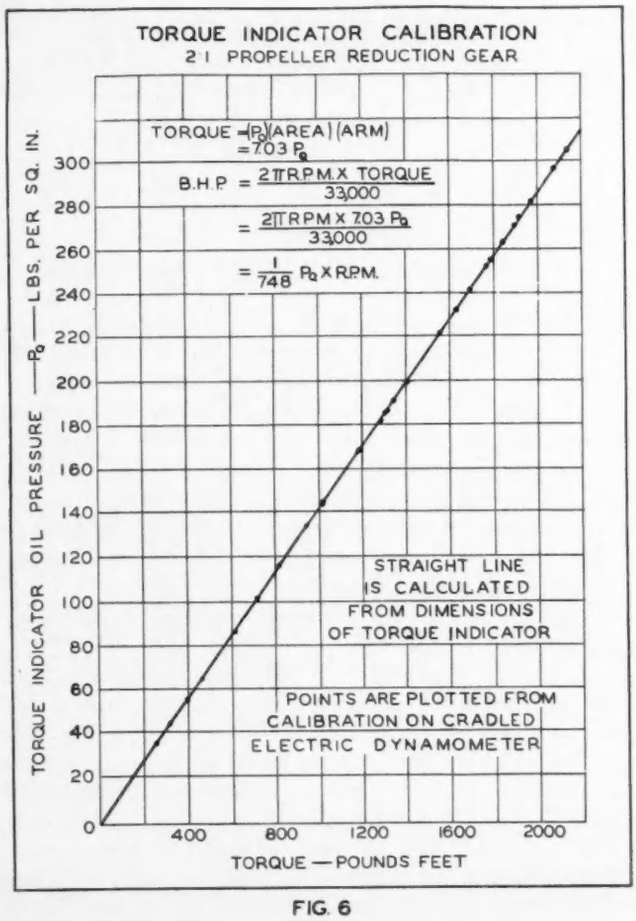


FIG. 6

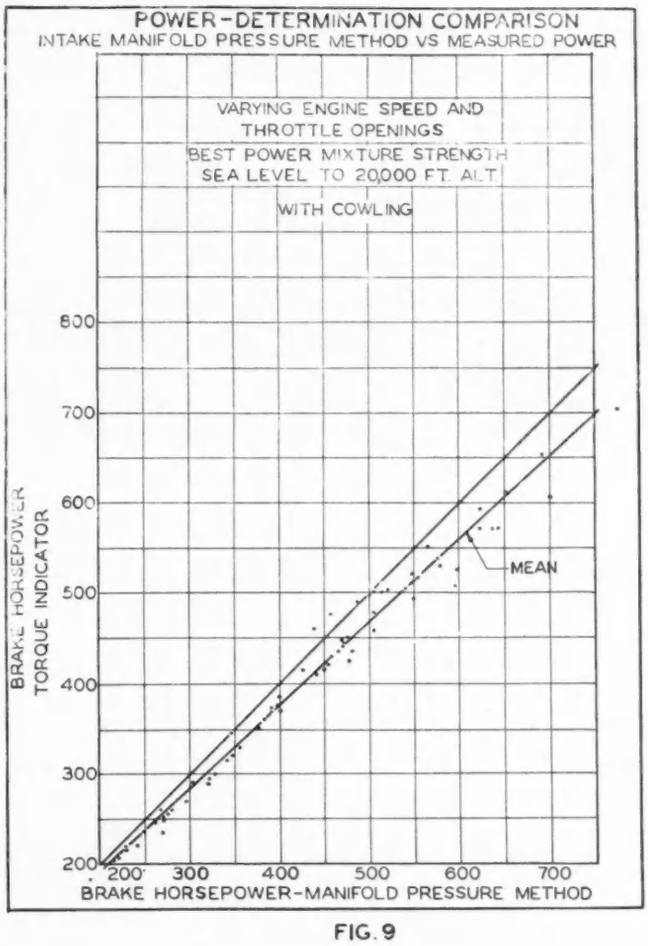


FIG. 9

A comparison between the powers measured by the torque indicator and those obtained by the intake-manifold-pressure method is made in Figs. 9 and 10 for all altitudes and throttle positions. Fig. 9 shows the results obtained with cowling. A mean line through the scattered points indicates that the

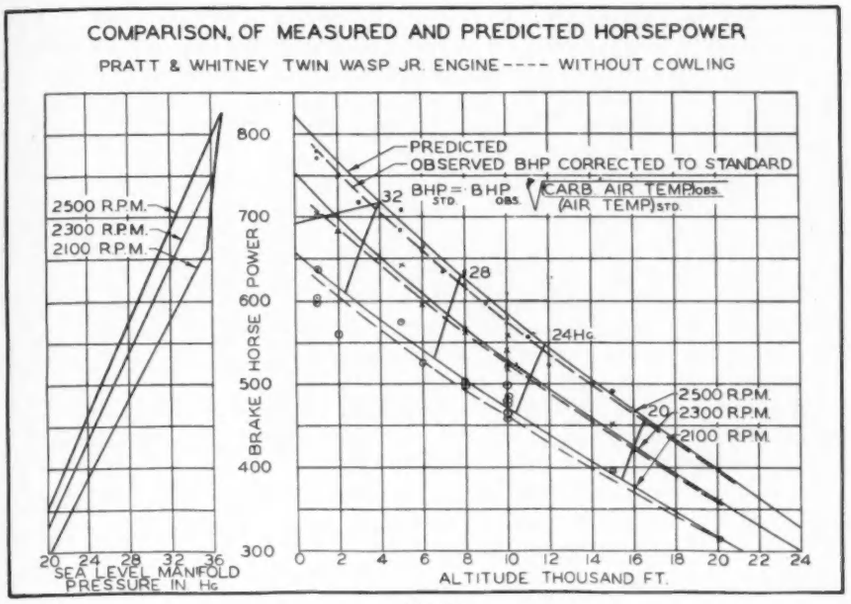


FIG. 8

power determined by the manifold-pressure method is about 8 per cent greater than the measured values.

In the comparison made without cowling, Fig. 10, the points are also scattered, but the powers by the manifold-pressure method and those read by the torque indicator are in much closer agreement. At the higher powers, however, the manifold-pressure method again indicates more power than the engine actually develops.

Other methods for correcting power at a given pressure altitude to standard atmospheric conditions may be used in addition to the carbureter-air-temperature method. Fig. 11 presents two additional methods of power correction for various free air temperatures. The solid lines represent the powers measured by the torque indicator on various flights at one pressure altitude, and the broken lines represent the values corrected to the standard temperature. A desirable correction would make these values constant for the complete temperature range, in other words, the broken lines would be made horizontal.

Figure 11-*A* (bottom) illustrates the inadequacy of the conventional carbureter-air-temperature correction when applied to an engine flying from day to day at the same pressure altitude but with carbureter-inlet temperature varying with the free air temperature. The correction factor does not take into account the effects of cooling-air temperature or density on engine power.

Figure 11-*B* (center) illustrates another

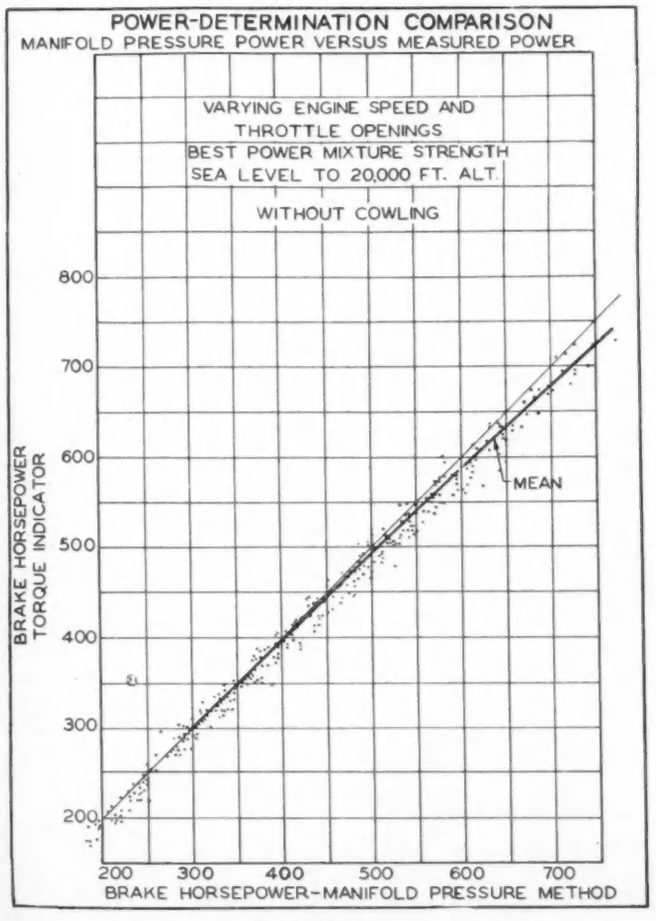


FIG. 10

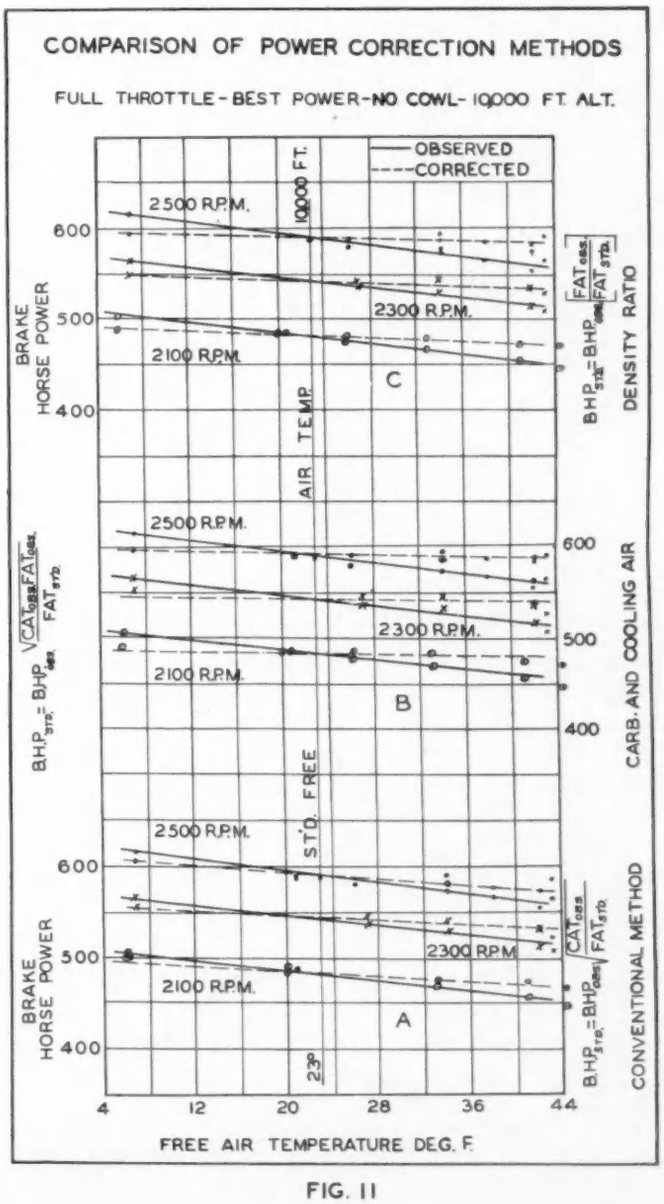


FIG. 11

method of correction in which the cooling-air-temperature is used.

$$\text{Cooling Factor} = \frac{}{\sqrt{(\text{Free Air Temperature})_{observed}\ (\text{Free Air Temperature})_{standard}}}$$

and, combined with the carbureter-air-temperature, the factor becomes

$$\text{Correction Factor} = \frac{\sqrt{(\text{Carbureter Air Temp.})_{observed}\ (\text{Free Air Temp.})_{observed}}}{\text{Free Air Temperature}_{standard}}$$

It is evident from the curves that the values obtained by applying this factor are much nearer the powers to be considered as standard than those with the correction for carbureter-air temperature alone.

Under flight conditions with cold air to the carbureter the free-air temperature and carbureter-air temperature are very nearly the same and, if assumed to be the same, the correction factor becomes the ratio of the density under standard conditions to the density at the observed conditions. Fig. 11-*C* (top) shows the effect of this correction which may be termed the density-altitude or density-ratio method. For the average range of free-air temperatures the processes of correction are simplified if the engine power can be plotted on a density-altitude basis, and would work hand-in-hand with airplane and propeller corrections.

With this torque indicator it is feasible to make complete engine calibrations at any altitude. Fig. 12 illustrates such calibrations made at 10,000 ft. Actually the points shown on these curves were obtained from a number of tests made at this altitude over a considerable range of temperatures. Data in this figure are plotted as constant-speed curves of manifold pressure versus power. The power values are corrected to standard temperature by the methods shown in Figs. 11-*A* and 11-*B*. Similar calibration tests have been made at several altitudes between sea level and 20,000 ft.

Flight data plotted at the observed density altitudes are shown in Fig. 13. This figure shows, on a density-ratio basis, the values of the observed full-throttle, best-power runs made during the series of tests. The measured powers were plotted against the equivalent density altitude prevailing at the time of the flight. For example, most of the points lying between 9,000 and 12,000 ft. were flown at 10,000-ft. pressure altitude. These small groups show the fairly reasonable corrective effect of plotting engine data on this basis and, as mentioned previously, this method would be very convenient in connection with airplane performance. Solid lines are faired through the points made without cowling, and the broken lines represent the cowled engine. These curves do not give perfect corrections but they seem to be reasonably consistent. It is felt that the method can be used with reasonable accuracy only when the carbureter is receiving air that is approximately the same temperature as the free air and that all cowling, cooling, and carbureter ram conditions remain un-

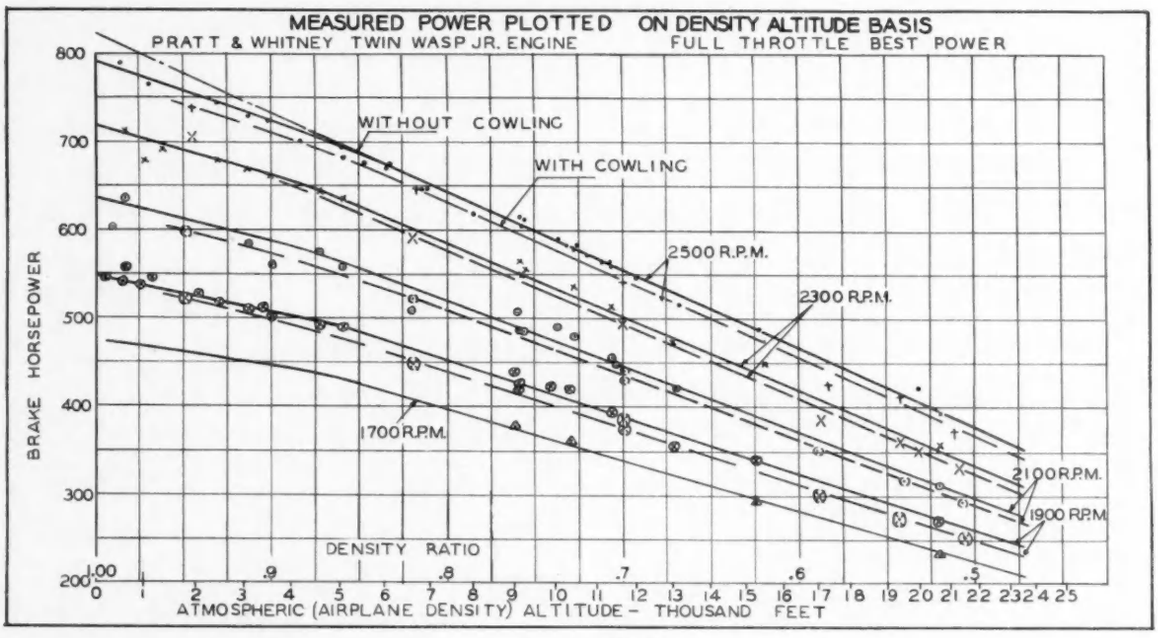


FIG. 13

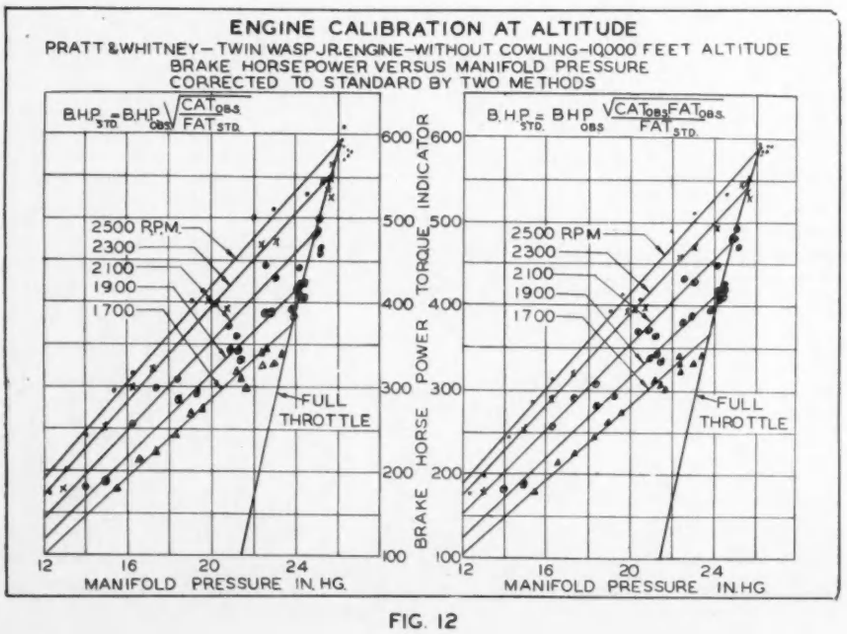


FIG. 12

changed between comparative flights. The method also presupposes the use of a constant-speed propeller or that the propeller is adjusted to maintain constant engine speed.

With cold air to the carbureter almost identical results are given when the power is plotted at the equivalent density altitude of the carbureter inlet. However, when preheated air is led to the carbureter, the equivalent density altitude of the carbureter is not a suitable method for correction, as no allowance is made for changes in manifold pressure and back pressure at the density altitude at which the power is plotted.

The general tendency of the engine at low altitudes to develop less power than predicted may be accounted for partially by the fact that the predicted power is based upon standard atmospheric conditions of absolutely dry air. In flight it is

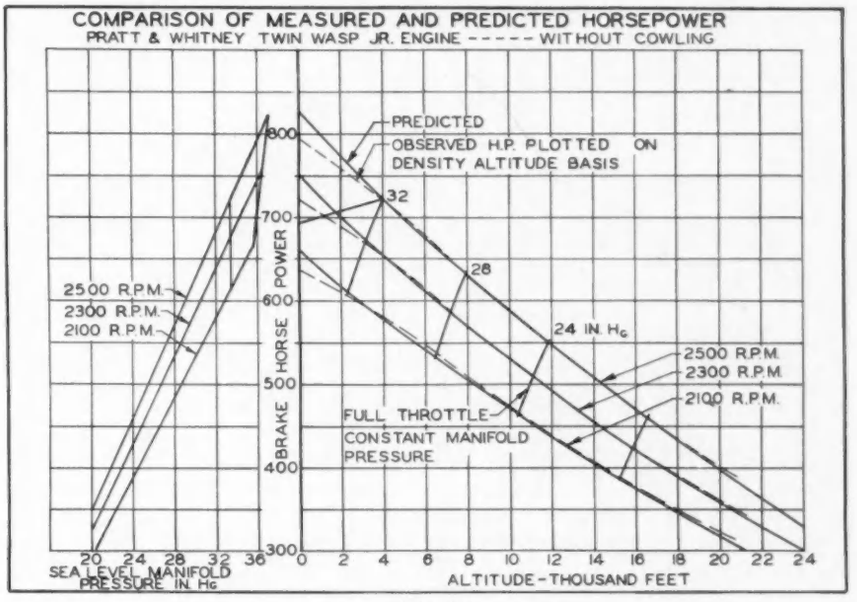


FIG. 15

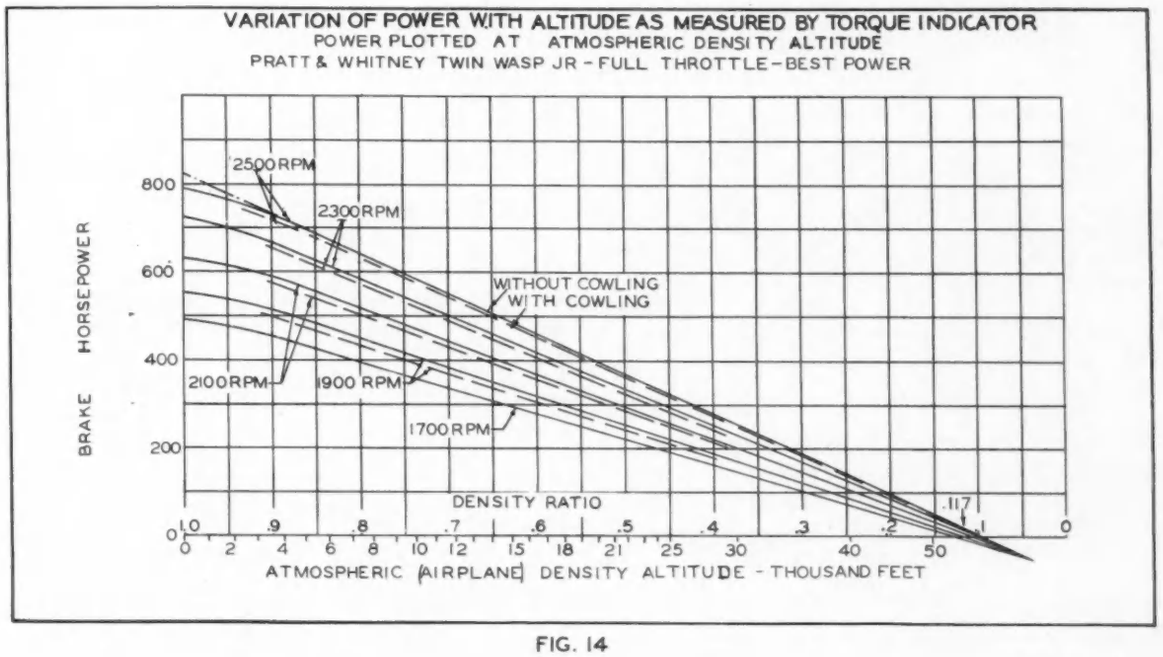


FIG. 14

impossible to obtain such a condition, and observed power at temperatures well above freezing must necessarily be lower than predicted because of the moisture in the air. Suitable corrections for vapor-pressure content may bring the observed flight power into closer agreement. In addition, the engine used during these tests is normally rated for full power at 4000 ft. and was operating above capacity when run full throttle at lower altitudes. The timing of the ignition and valves, carbureter capacities and other such items, were for normal operating conditions. The engine exhibited tendencies toward detonation at the very low altitudes when running full-throttle best-power, and it was necessary to have the mixture on the rich side of "best power" in order to maintain the highest power without excessive cylinder temperatures, in other words, these powers may more properly be called "best compromise power."

In Fig. 14 the straight portions of the lines of Fig. 13 have been extended to their intersections with the zero brake horsepower line and on to what is apparently their common intersection. It is most interesting to note that the zero brake horsepower for the various engine speeds comes between atmospheric density ratios of 0.092 and 0.125, at an average value of 0.108, a value that checks remarkably well with the one of 0.117 directly calculated from a formula in the paper by Gagg and Farrar.[1]

The common intersection of the lines is at a density ratio approaching zero and a negative value of power. Possibly this indicates that the friction horsepower in practically a vacuum would be of the magnitude indicated. However, it must be pointed out that the portion of the diagram above 21,000 ft. has been extrapolated.

The extensions of the straight portions to sea level coincide very well with the full-throttle sea-level power for standard atmosphere (dry) as shown by the conventional sea-level calibration chart. The comparison in Fig. 15 of corrected power as shown by this method and the predicted power is

[1] See S.A.E. TRANSACTIONS, June, 1934, pp. 217-225: "Altitude Performance of Aircraft Engines Equipped with Gear-Driven Superchargers," by R. F. Gagg and E. V. Farrar.

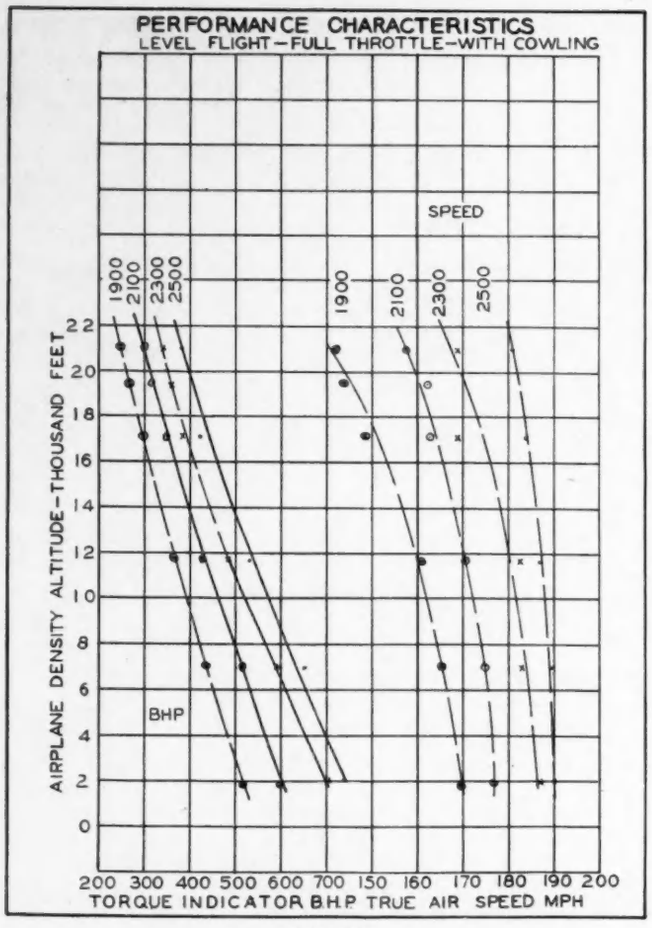


FIG. 16

in excellent agreement above 4000 or 5000 ft., and this agreement without any allowance for humidity.

Studies of Performance Factors

In addition to measuring engine power the torque indicator has proved itself to be very useful in studying other factors influencing airplane and engine performance. A discussion of some of the various performances obtained is given in the following paragraphs.

Altitude with and without Cowling

Fig. 16 shows the level-flight, full-throttle, best-power engine powers and airplane speeds during the tests with cowling, and Fig. 17 shows the performances without cowling. The airplane speed data show the scattering usually encountered if runs are made over a period of time and point out the fallacy of depending upon an isolated run as representative of the performance. During these tests there was a 12 per cent maximum variation in gross weight of the airplane due to consumption of fuel. Comparison of airplane speeds at 10,000 ft. shows an increase in speed of 3 or 4 m.p.h. (1½ per cent) with cowling over the speeds without cowling, accompanied by and in spite of, a drop in engine output of 10 or 12 hp. (1½ per cent). The gain in speed is less than might be expected for a carefully designed cowl and, no doubt, the speed was influenced by the large external oil cooler under the fuselage while the engine was cowled.

Effect of Cowl Flap Setting

The effects in level flight of cowl flap setting are illuminating. Fig. 18 shows the increase in brake horsepower resulting from the cooler running powerplant as the flaps were opened. Changing the flap setting influenced carbureter-air-temperature as well as the cylinder temperatures, and both factors contributed to the increased power. The increase in power at 10 deg. flap-angle was 3½ per cent of the power at 1 deg. flap-angle. The increase in drag of the airplane was between 2 per cent and 3 per cent which was more than offset by the gain in power with the result that the airplane speed increased slightly. However, as the flaps were opened farther, the increase in power was more than offset by the increased drag and the airplane lost speed. Specific fuel consumptions were maintained consistently at "best power," but there was a loss in miles per gallon with increased flap opening and the consequent increase in drag.

Fuel Consumption with Altitude

The general trends of fuel consumption with altitude at full-throttle, best-power are shown in Fig. 19. The total flow decreases approximately in proportion to the decrease in power output of the engine with altitude. The specific consumption gradually increases with altitude and is, no doubt, dependent upon the ratio of brake horsepower to friction horsepower, with friction horsepower falling off less rapidly than does the brake horsepower. Due to decreasing power, the range at full throttle in miles per gallon increases at an impressive rate as the altitude is raised, becoming some 30 per cent greater at 10,000 ft. than at sea-level, and about 70 per cent better at 20,000 ft., an improvement of about 3 per cent per thousand feet. The decrease in air speed at 20,000 ft. was about 10 per cent.

Effect of Mixture Strengths

Fuel consumptions are of vital interest since they affect range. The use of the torque indicator in combination with

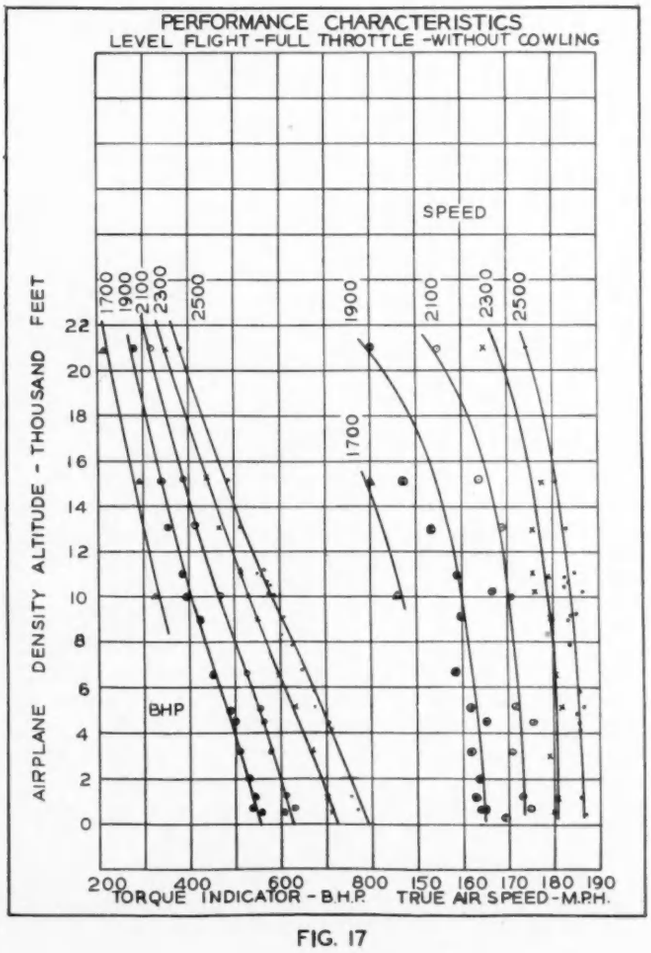


FIG. 17

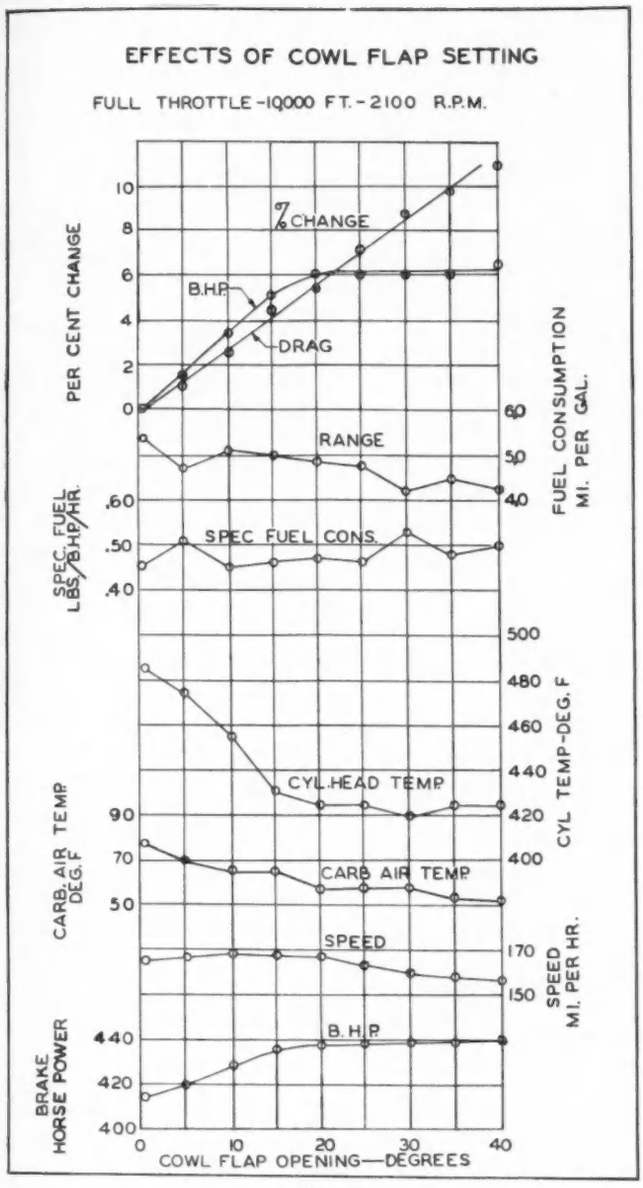


FIG. 18

a fuel flowmeter has enabled specific fuel consumption to be read with accuracy.

The following results were obtained on an experimental engine, and it must be understood that many of the operations performed were unusual and that continued operation under these conditions might be extremely detrimental to the engine.

Figs. 20 and 21 have been drawn to illustrate graphically the effects produced by movement of the mixture control. Small notches divided the quadrant in the cockpit into divisions of 0.10 in., the total number being 25. During this test the mixture control was started at full-rich position and moved progressively one notch at a time until the very lean positions were reached, where it was necessary to move in fifth-notch increments. Leaner mixtures than approximately best-economy were accompanied by a surging of the power that became more pronounced as the mixture control was moved to the successive positions until the tests were stopped by very large fluctuations and occasional complete interruptions of the flow of power. This condition is indicated more or less to scale by the forked portion of the brake-horsepower curve in Fig. 20. The results of a check run over the lean portion of the tests are shown by the broken lines.

The brake-horsepower curve shows a gradual increase in power as the mixture is leaned out until best power is reached which results in an increase of 4½ per cent in power with the particular carbureter used. Referring to the lowest portion of the specific-fuel-consumption curve, commonly spoken of as "best economy," it is found that the corresponding point on the power curve is approximately 90 per cent of "best power" at the highest portion of the power curve. It is also quite evident that practically maximum power is attained over a range of three or four notches in this case. The specific fuel consumptions show a variation from "rich best power" of 0.56 lb. per b. hp-hr. to "lean best power" of 0.46 lb. per b. hp-hr., a change of around 20 per cent in fuel consumption with practically no change in power. Airplane speeds reflect the changes in engine power. One reason for the lower speeds on the check flight may be the 10 per cent increase in weight of the airplane after refueling and its effect on induced drag.

A glance at the cylinder temperatures in Fig. 21 shows the effects of fuel consumptions and helps to clarify the engine manufacturers' instructions concerning use of the mixture control at full power. From "rich best power" to "lean best power" there is a rise in cylinder-head temperature of about 40 deg. fahr., and from "full rich" the rise is about 100 deg. fahr. It must be pointed out that the carbureter used for these tests was fitted arbitrarily with richer settings than normal, and that this test was flown in level flight at 10,000 ft., where the lowered air density enriches the full-rich mixture strength as compared with sea-level value. In full-throttle climbs, due to lower air speed, it was always necessary to keep the mixture near "rich best power" in order to keep the cylinder temperatures under 500 deg. fahr. to 520 deg. fahr. If climbs were made less steep and at higher air speeds than for best rate of climb, it was possible to operate with some-

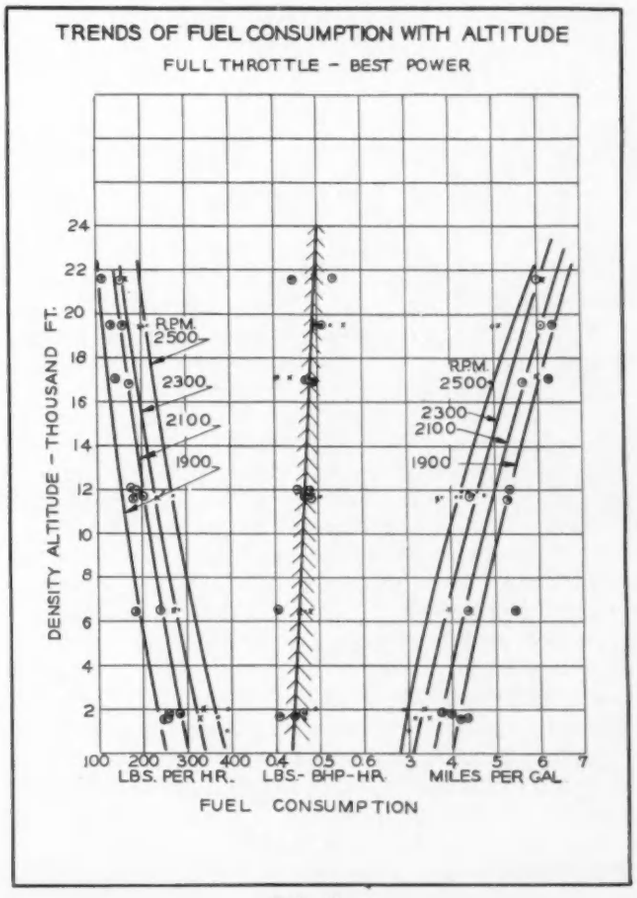


FIG. 19

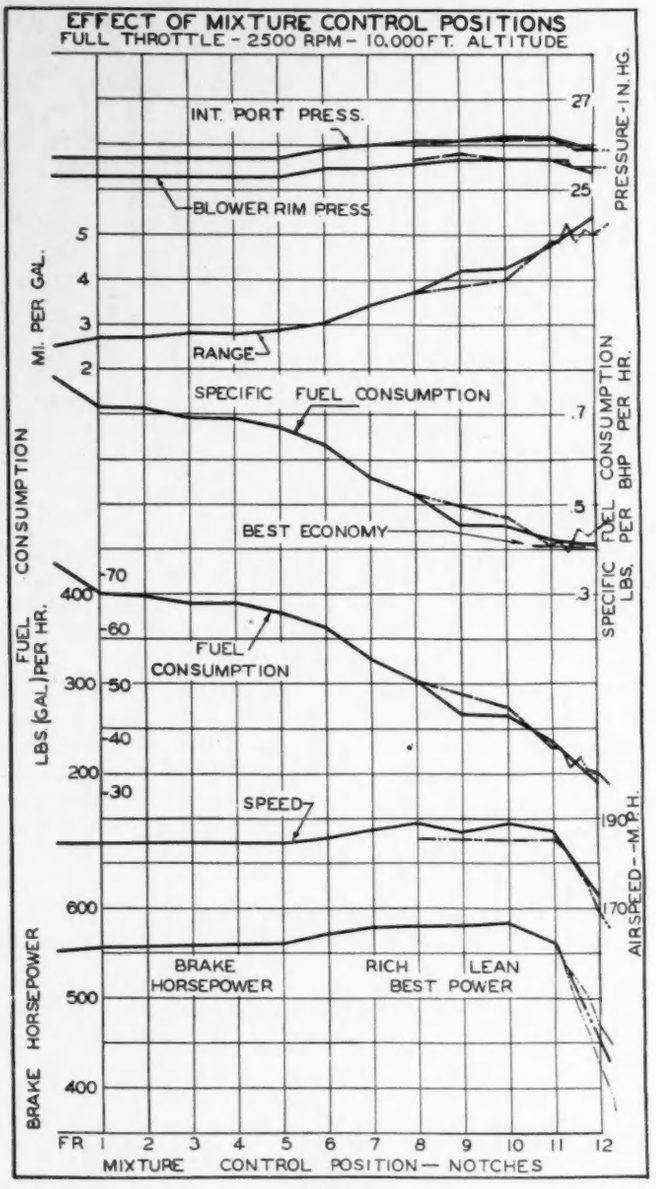


FIG. 20

what leaner mixtures and still keep the cylinder temperatures under control.

"Range," or fuel consumption in miles per gallon, improved from 3.5 miles per gal. at rich best power to 4.2 miles per gal. at lean best power as shown in Fig. 20, a change of 20 per cent. From best power to best economy, the improvement in range is somewhat misleading because of the reduction in engine power and a consequent improvement in range due to flying at slower speed. This phase was investigated and is discussed elsewhere in this paper. Accepting the sacrifice in speed, the range was improved about 20 per cent by the compound effect of lower power and of leaning out to best economy from best power.

It is interesting to note that the general trend of the induction-system temperatures beyond the blower is to rise with leaner mixtures as shown in Fig. 21, no doubt due to the increased temperatures of the cylinder-head being reflected by a general rise in temperatures throughout the engine and also due to less fuel to be evaporated and to absorb heat. Manifold pressures, either at the intake port or at the blower rim, show a rise of about 2 per cent as the mixture is leaned out.

The characteristic performances of an engine as fuel flow is modified are often shown by plotting the items against the fuel consumed per hour. Representative curves of this type are shown in Fig. 22 that give the characteristics of the engine at full throttle at several engine speeds at 10,000 ft. These curves exhibit a remarkable consistency in the data, more so than was expected at the time the tests were made. The high engine speed was run through a greater range of fuel flow than the other speeds and its curve shows the characteristic shapes that may be expected. At each speed the maximum power shown by the brake-horsepower curve as it changes from positive to negative slope is termed "best power." Best economy is indicated at the lowest point of the specific consumption curve.

These curves show "best power" consumptions between 0.45 and 0.52 lb. per b. hp-hr., the lower values going with the lower engine speeds. Best economy is around 0.40 lb. per b. hp-hr., with practically no difference between engine speeds.

In these tests it is seen that the cylinder temperatures, heads and bases alike, show a drop in temperature between best power and best economy. This tendency quite evidently follows the power curves. This result does not mean, however, that it is desirable to run engines under these conditions because lean mixtures result in an excess of oxygen in the burning gases; and points of high temperature in the spark-plugs, combustion-chamber, and exhaust port are eagerly sought out by the free oxygen with destructive results. Therefore, lean mixtures require adequate cooling of all parts of the combustion-chamber, pistons and rings, and other parts exposed to the hot gases. Every engine manufacturer makes thorough tests of his products to determine safe fuel-air ratios or mixture strengths for various power outputs, and it is well for the operator to follow the manufacturers' instructions. The possibility of saving a few gallons of fuel is not worth the risk of a damaged powerplant.

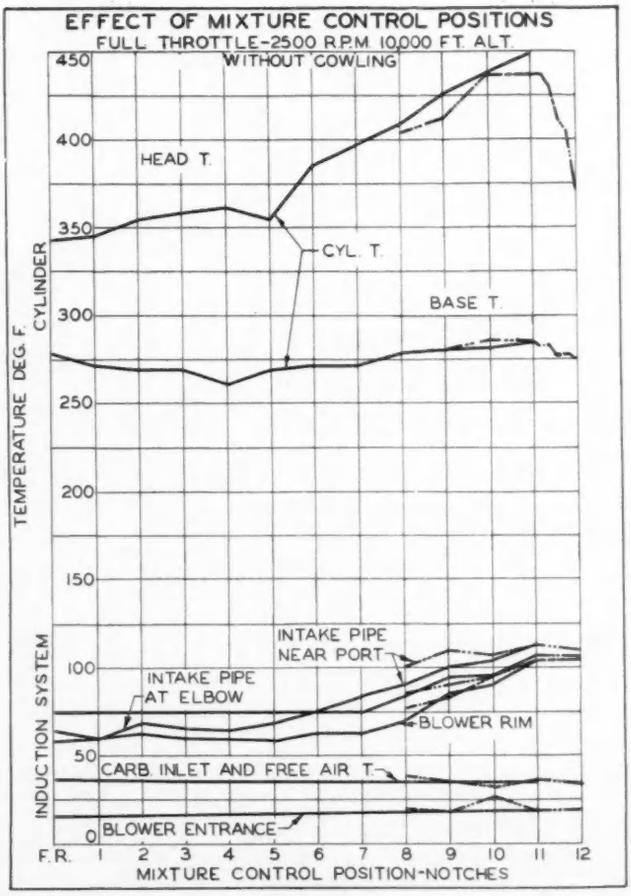


FIG. 21

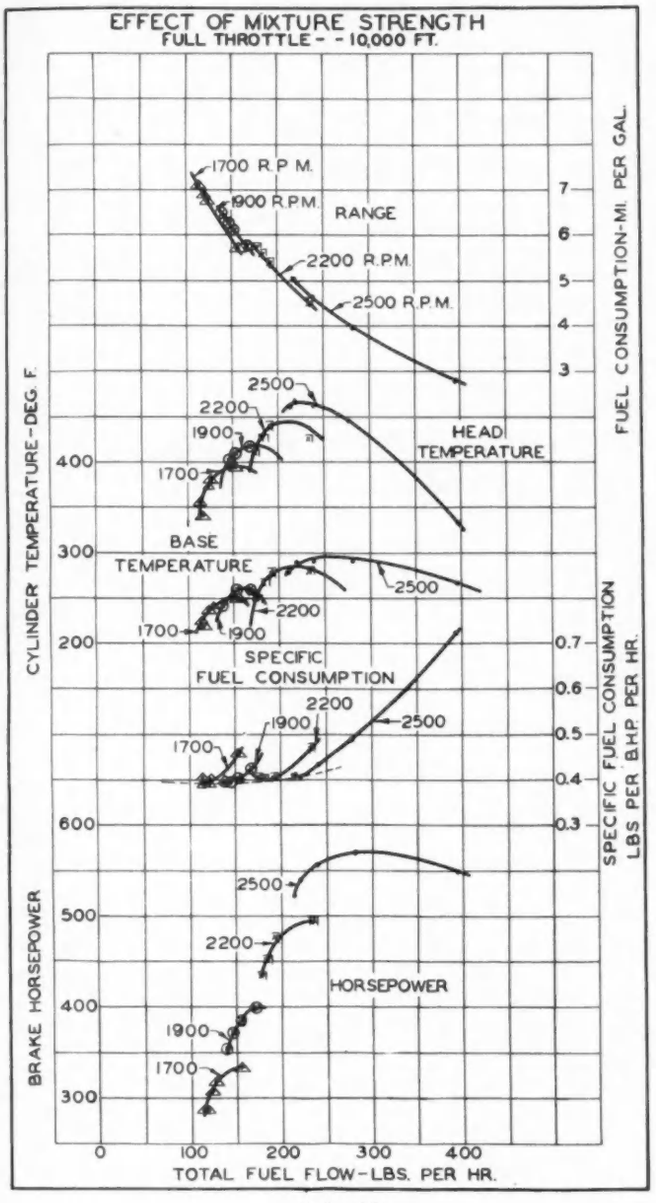


FIG. 22

Range in miles-per-gallon increases as the flow of fuel decreases, a considerable part of the apparent improvement being due to lowered power and consequent lower speed as the mixture is leaned out beyond best power. In Fig. 23 true air speed is used as a basis instead of fuel consumption. The brake-horsepower curves now show their similarity to the conventional thrust-horsepower-required curves. The short sections of the constant-engine-speed curves, when extrapolated, indicate a tendency toward higher brake-horsepower requirements at low engine speeds than at high engine speeds for a given air speed. Propeller efficiencies for these runs have not been investigated, and the curves do not cover the same range of air speeds. It is, therefore, fallacious to make any assumptions from these data as to relative power requirements at the same air speed.

The gain in power from over-rich mixture to best power (best speed) also shows up in Fig. 23. The specific consumption in pounds per brake horsepower per hour shows practically no change once the minimum value is reached and would indicate that nothing is gained by leaning out excessively but, rather, that it would be better to reduce the power and airplane speed by a reduction in engine speed and still maintain a reasonable fuel-air ratio. Envelopes of the curves of fuel consumption give ready means of estimating the performances with intermediate speeds.

Cylinder temperatures in Fig. 23 show the trends of the effects mixture strength and engine power have upon operating temperatures. It would appear from extrapolations of these curves to be possible to operate with an excessively lean mixture in order to lower the power and have lower cylinder temperatures than could prevail with the normal manner of operation.

Constant-Power Runs

To answer the question raised in the previous paragraph, a series of tests was made at constant power as shown by the torque indicator at three engine speeds. At each change in mixture strength and engine output, the throttle was adjusted sufficiently to regain the desired power. All the runs were at part throttle, and a power was selected low enough to allow considerable increase in throttle opening and manifold pressure to offset the loss in power with extremely lean mixtures. The test was flown at a low altitude in order to have sufficient manifold pressure available to cover a fairly wide range.

The results of the constant-power runs are shown in Fig. 24. The fuel-consumption and cylinder-temperature curves

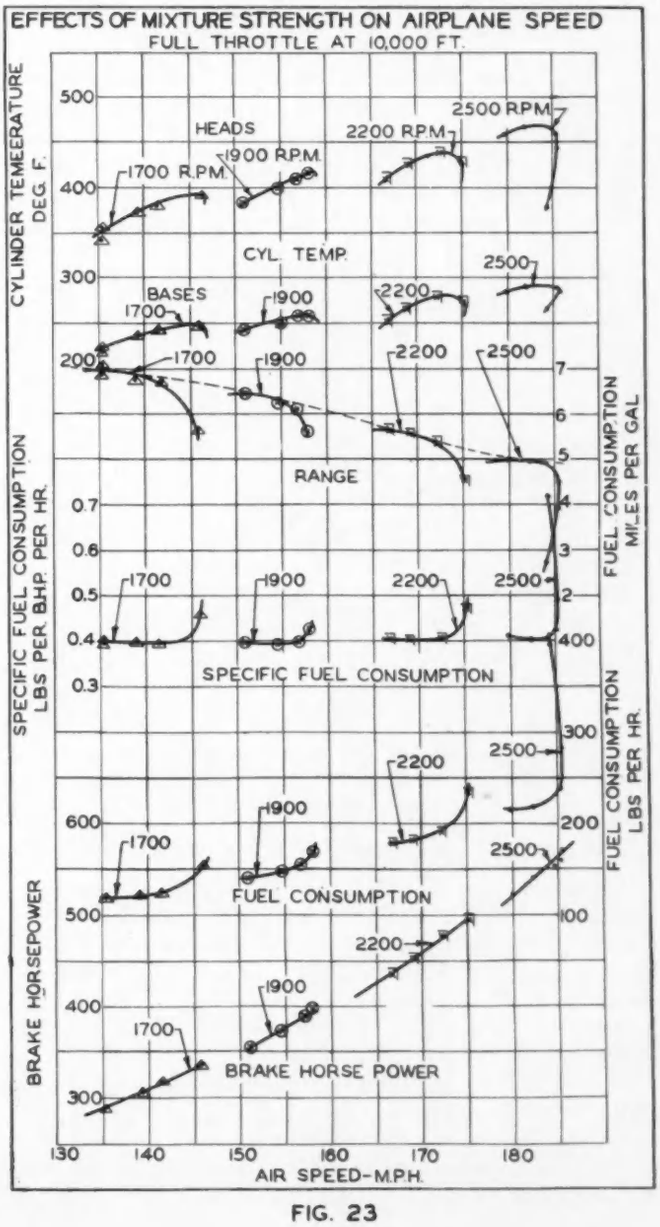


FIG. 23

indicate the ranges of manifold pressures required to maintain the desired power at the various engine speeds. The specific fuel consumptions and range show that no greater economy is gained by excessively lean mixtures than is attained by moderate mixture strength. Investigations of ignition timing, valve operations, turbulence, and other items possibly will lead to some improvements in the future. The question of engine reliability and maintenance expense always must be in the forefront, and many hours of testing are required at each step along the way to answer these questions. Peculiarly enough, cylinder temperatures show a consistent tendency to be lowered with extremely lean mixtures but, as mentioned in another paragraph, it is not known what effect prolonged operation with these lean mixtures would have on the cylinders, pistons, and valves.

Characteristics for Maximum Range

An example is given in Fig. 25 of the use of the torque indicator in determining operating characteristics for maximum range of an airplane for long-distance flying. In the present case the propeller pitch indexing was not high enough to absorb the power at lower engine speeds and higher brake mean effective pressures than shown. The trends, however, are indicated. The engine was operated at fixed throttle for each engine speed, and fuel flow was measured at various mixture settings allowing the power and air speed to vary.

It was found that the highest brake horsepower was required at the lowest engine speed. The power at 100 m.p.h. is some 3 per cent less at high engine speeds than at low. The fuel consumptions tell another story, and the low engine speeds show much better fuel economy than do the high.

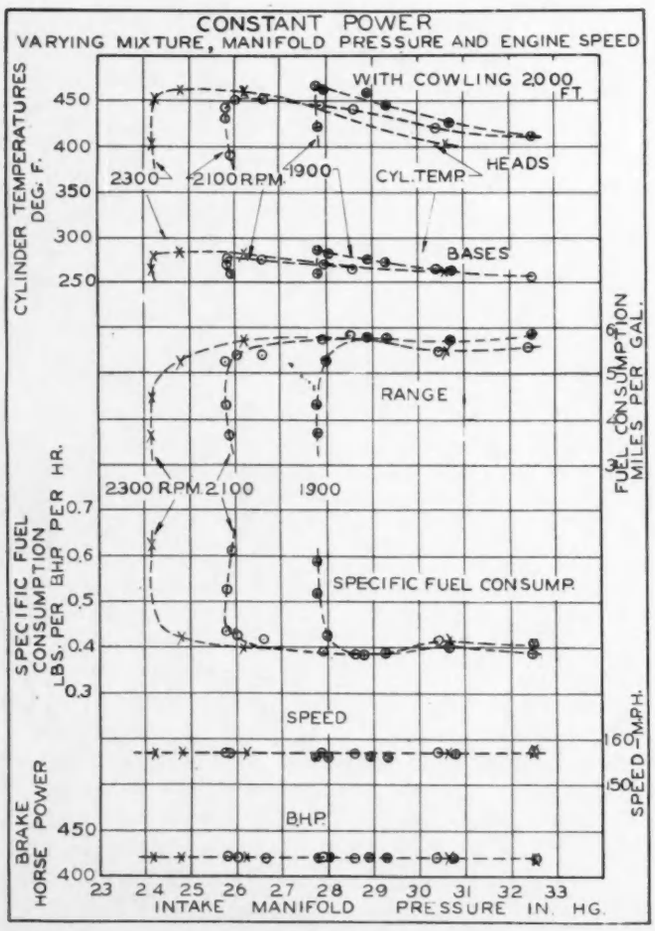


FIG. 24

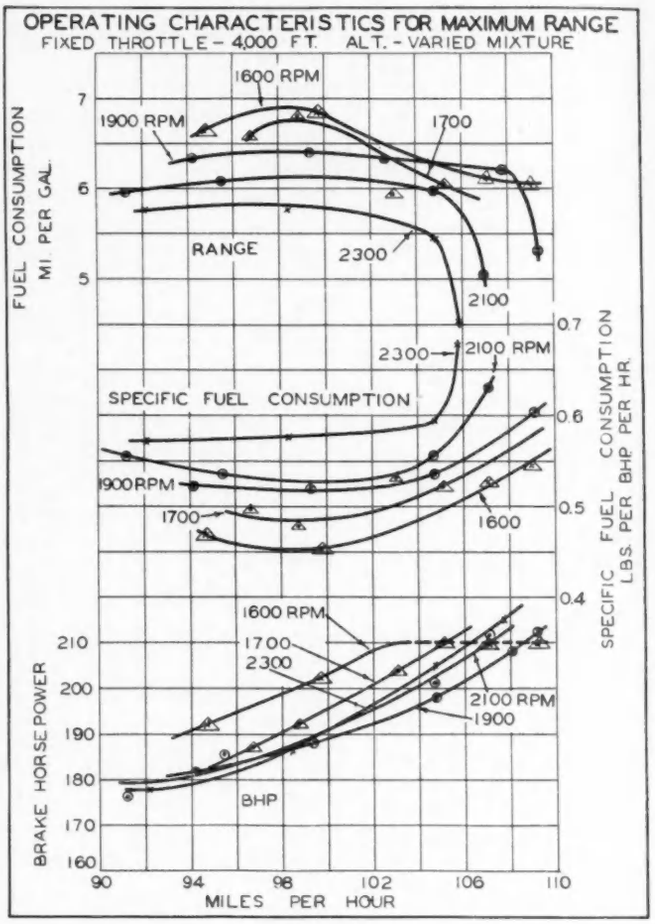


FIG. 25

This result indicates the necessity of finding the point at which engine and propeller characteristics give the best compromise. For a long-range airplane it would be necessary to develop a series of diagrams for various speed and weight combinations.

Effects of Carbureter-Air Temperature

Through moderate ranges the effects of carbureter inlet-air temperature are fairly well known from test-stand data. A short test was made at 10,000-ft. altitude at full throttle and rated engine speed to obtain information as to the effects of varying the preheat from cold air to the highest temperature available from the air heater surrounding the exhaust collector. The pressure at the carbureter inlet was regulated manually to the flight altitude, and the carbureter air was varied from 60 deg. fahr. to 245 deg. fahr. with results that were not anticipated.

Fig. 26 shows the various engine data plotted against carbureter inlet-air temperature. The measured horsepower showed an expected decrease up to about 120 deg. fahr. inlet-air temperature and then changed to a curve of lesser slope. The actual shape of the horsepower curve was difficult to visualize until the plotting of the induction-system temperatures definitely developed the outlines. The proportional increases in temperature at the blower entrance and intake pipes as the inlet-air temperature was increased from 60 deg. fahr. to 120 deg. fahr. and the definite leveling off above 120 deg. fahr. suggested the possible effects of vaporization and fractional boiling points of the fuel. The initial end point of the fuel distillation curve falls between 115 deg. fahr. and 120 deg. fahr., and the list of specifications for the fuel calls for

not more than 90 per cent evaporation at 240 deg. fahr. This information may explain the induction-system-temperature diagrams. The heat in the inlet-air is absorbed by the latent heat of evaporation of the fuel in proportion to the distillation curve.

The brake horsepower as measured by the torque indicator while the carbureter-air temperatures were below 120 deg. fahr. is interrelated in conformance with the conventional correction by the square root of the temperature ratio. The higher carbureter-air temperatures showed no appreciable effect upon cylinder temperatures. The manifold pressure was lowered approximately 2 per cent for 100 deg. fahr. rise in carbureter inlet temperature.

Static Thrust Characteristics

Fig. 7 shows the airplane without cowling set up for "static thrust" tests. A cable from the airplane to scales some distance back allowed the actual thrust of the propeller to be measured while, at the same time, engine power and other data were taken in the airplane. Fig. 27 shows the static thrust plotted against brake horsepower at fixed settings of the propeller, commonly spoken of as "propeller load curves," also at seven engine speeds with the propeller governed by the constant-speed control. These curves demonstrate some interesting

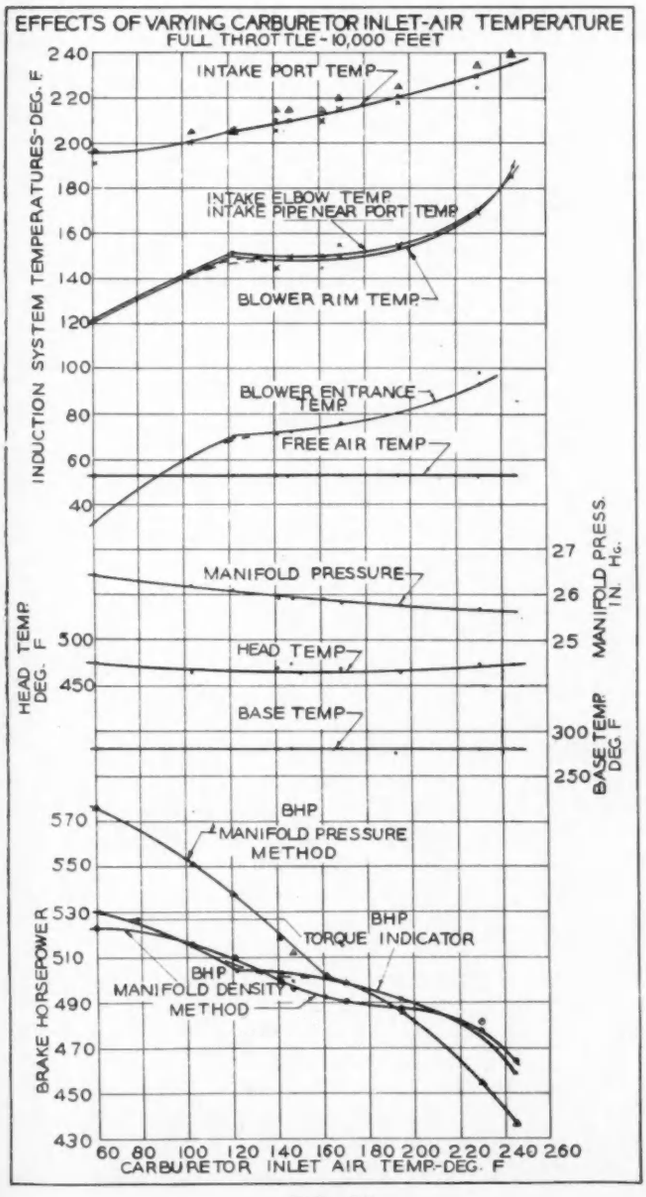


FIG. 26

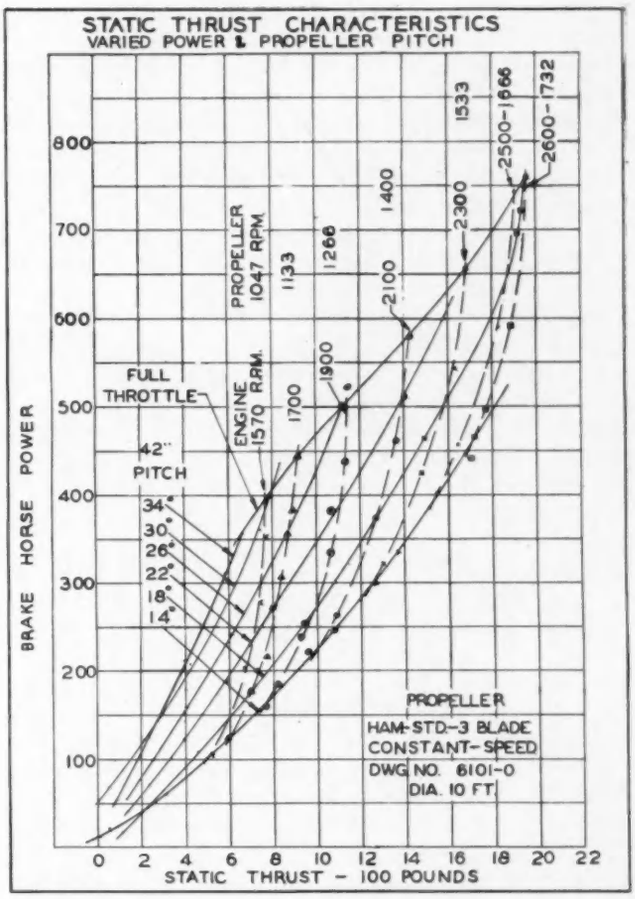


FIG. 27

features, particularly the great gain in static thrust at low pitch settings and high engine speed as compared with high pitch settings and low engine speed, at a constant-power output. At 350 b. hp., for instance, there is an increase of 117 per cent in static thrust due to a change in propeller pitch of 20 deg. and an increase in engine speed of about 60 per cent. The greatest gain, of course, came with full-throttle operations where the power increased about 100 per cent and the static thrust showed a rise of 190 per cent. The curves of constant-engine (propeller) speed show that substantially maximum thrust is attained with 50 to 100 hp. less than full throttle, and that overloading the propeller does not result in a material gain in thrust for take-off. These curves point out that the amount of power absorbed by a propeller is not necessarily a measure of the thrust.

Oil Temperature

During the tests it was noted that oil temperatures followed engine speeds quite closely and were affected to only a small extent by the actual power developed.

Ignition

The torque indicator showed itself to be of value in checking ignition troubles while a constant-speed propeller maintains engine speed. Under normal conditions there is a small lowering in torque reading on either switch as compared with both. With a spark-plug misfiring there is a pronounced difference in torque between the two switches. The pointer of the indicator gage showed marked torque variation when a spark-plug was firing intermittently.

Conclusions

The Pratt & Whitney Aircraft torque indicator has proved to be a simple, practical, accurate, and dependable device for

the measurement of engine power in flight. The instrument is automatic in its action and gives continuous indications of the engine torque under all conditions of operation in which the engine is developing power.

The altitude power of the engine investigated was found to be in close agreement with the predicted power except near sea level, where differences as high as 4 per cent occurred under certain flight conditions.

These tests indicate that the square root of the carbureter-air-temperature ratio is deficient in its correction of observed power to standard atmospheric conditions at any pressure altitude. The addition of a similar correction for cooling-air-temperature brings the corrected power more closely into line. A simple yet reasonably accurate method of correction is to plot the observed power at the density altitude of flight, a method that works hand-in-hand with the corrections for airplane and propeller performances.

The torque indicator as a piece of test equipment will be invaluable to:

(1) The engine manufacturer for correlating test-stand operation with flight performance.

(2) The propeller manufacturer in analyzing propeller performance.

(3) The airplane manufacturer in removing all doubt regarding engine performance during take-off and flight tests with new airplanes, and with airplanes in which modifications have been made in drag conditions.

(4) To the purchaser of an airplane in his acceptance of the performance.

As standard engine equipment the torque indicator will justify the slight additional weight by making possible efficient control of engine speed, throttle opening, mixture setting, carbureter-air temperature, cowl-flap setting, and other items that affect the daily commercial or military operation of an engine.

Engine power in flight is now an easily determined quantity.

Trend in Truck-Engine Design

GASOLINE engines will continue to be furnished in sizes ranging from 225 cu. in. up to 530 cu. in. in reasonably large quantities, and the variation in these engine sizes seems to be in the neighborhood of 60 to 70 cu. in. Continuous development of combustion-chamber design will permit further increases in the brake mean effective pressures of gasoline engines using existing fuels. The specific fuel consumption, judging from the excellent results thus far obtained, will be lower and fuel injection into the intake manifold probably will be used on trucks before passenger cars.

The thin-wall bearings for mains and connecting-rod big ends seem to have proved satisfactory. Because of their flexibility, they conform to the accurately bored bearing caps and housings.

Aluminum, cam-ground pistons with an "Alumilite" finish which converts the surface of the piston from aluminum to an aluminum oxide, which is hard and porous and facilitates lubrication and reduces wear, continue to replace cast-iron pistons.

The Franklin six-cylinder 110-hp. heavy-duty air-cooled engine is a recent development and is designed particularly for equipment which is stored in unheated garages, or operates under other severe conditions. Due to the use of automatically adjustable valve tappets, the engine operates without the usual valve clatter of air-cooled engines.

Diesel Engines

In connection with Diesel engines, it appears that there will be a slow but steady increase in usage. Wherever gasoline is relatively inexpensive, comparatively little is heard about Diesels for highway transport, but in Europe the topic is one of vital interest and, in the past few years, the development of highly effective units, both in Europe and America, has quickened noticeably. Writing in a recent issue of *The Engineer,* of London, Harry R. Ricardo, the well-known engine designer, traces that progress and explains how it came about.

First among the contributing factors he lists the breaking down of the belief that the heavy-oil engine cannot run fast. There never was the slightest foundation for such a belief, Mr. Ricardo states; but, nevertheless, it was a deep-seated conviction. The second influence was the interest taken in the Diesel by engineers of long experience in designing and building gasoline engines. They are now making Diesels with the same care and thorough methods that they have long applied to the manufacture of gasoline units. Third on the list is the development, by this same group of technicians, of a small precision pump that accurately will both meter and time the injection of minute quantities of liquid fuel over a very wide range of speed.

In designing current high-speed Diesels, many time-honored principles were abandoned. In fact, the provisions for injecting fuel oil into the cylinders have reverted to those that were used during the first ten years of Diesel development. This is but one manifestation of a revolution, which Mr. Ricardo describes as follows:

"A new school of engineers, with a new mechanical technique and no respect for inherited tradition, took possession of the heavy oil engine and determined to make it dance to an altogether livelier tune, and even to race the gasoline engine on the public highways. Its mechanical design they consigned straightway to the waste-paper basket, and started afresh with an entirely different conception, in which stiffness of structure and light moving parts formed the central ideal, while economy of material and accuracy of machining, in order both to eliminate hand-fitting and insure interchangeability, formed the economic theme."

Limitations of time and space will not permit a description of the various compression-ignition engines now available for trucks but, in passing, it seems fit to mention the Cummins fuel distributor as an interesting development of a Diesel engine accessory. Instead of the usual single fuel metering pump to supply each cylinder, Cummins employs a single pump and a distributor disc which is designed to meter and deliver a uniform charge to each cylinder. In broad principles it resembles the usual ignition distributor used on gasoline engines.

Excerpt from the paper: "Development and Trend in Truck Design," by Joseph A. Anglada, consulting engineer, presented at the Metropolitan Section-Regional Transportation and Maintenance Meeting of the Society, Newark, N. J., Nov. 9, 1937.

Effect of Test Conditions on Fuel Rating

Report from Cooperative Fuel Research Committee

Presented by A. E. Becker

THIS paper is a progress report of the Cooperative Fuel Research Committee, dealing with the laboratory section of the study of the knock-rating correlation problem. It is proposed that these results and those obtained on the road be the basis for further study aimed at the development of better correlation between road and laboratory ratings.

AT a meeting of the Detonation Subcommittee (now the Motor Fuels Section) of the Cooperative Fuel Research Committee in November, 1936, the Laboratory Procedure Group[1] was organized to study the effect of C. F. R. engine variables on the knock ratings of fuels. This work had as objectives:

(1) To develop several sets of test conditions to give ratings between those of the motor and research methods.

(2) To develop, if possible, a set of test conditions that would be more severe than the motor method.

(3) To develop a set of test conditions that would be as mild as or, if possible, less severe than the research method.

Scope

Variables. – The work performed in carrying out this assignment covered a study of the following:

(1) Standardization of knock intensity for the various conditions of test. To facilitate comparison of results, it was decided to make all tests at a knock-intensity level comparable to that of the motor method.

(2) The relationship between spark advance and power, and between spark advance and knock, throughout the compression-ratio range used in making octane ratings. This study was made at several speeds and two mixture temperatures.

(3) The effect of the following variables on the ratings of a selected list of the W fuels, as defined below:

(a) Speed – 600, 900, and 1200 r.p.m.

(b) Mixture temperature – 300, 200, and 150 deg. fahr.

(c) Constant spark advance – 26, 17, and 10 deg.

Participation. – This work was open to all members of the C. F. R. Group, as well as to other organizations that expressed a desire to participate. The participants were:

The Atlantic Refining Co.
Chrysler Corp.
Continental Oil Co.
Ethyl Gasoline Corp.
General Motors Corp.
Gulf Research and Development Co.
National Bureau of Standards.
Phillips Petroleum Co.
The Pure Oil Co.
Shell Petroleum Corp.
Sinclair Refining Co.
Socony-Vacuum Oil Co., Inc.
Standard Oil Co. of Calif.
Standard Oil Co. (Ind.)
Standard Oil Development Co.
The Texas Co.
Tide Water Associated Oil Co.
Tulsa Group.
Universal Oil Products Co.
Waukesha Motor Co.

Fuels. – Reference fuels A-5, C-10, and F-1 were used throughout the work. For convenience in comparing results, all data – except those by the L-3 method – were converted to the octane scale by using the motor-method calibration curves, Fig. 13.

The test fuels were selected from the winter (W) series of fuels used in the road-test program. In the later and more extensive portion of the laboratory program, they consisted of WA-1, WB-3, WB-5, WF-2, WG-1, and WH-1. To extend the work well above the 80-octane level, tests were also made using WH-1 plus 2 cc. tetraethyl lead per gal.

Knock-Intensity Standardization

To investigate the effect of an engine variable or test condition on the rating of a fuel, it is desirable that all other conditions be maintained constant. Difficult as this condition is to accomplish, all members of the procedure group recognized that knock intensity must be maintained as nearly constant as possible while investigating other variables. It was decided to use the knock-intensity level prescribed for the motor method. For the sake of uniformity, this "standard knock intensity" was defined as equivalent to that obtained with a blend of 65 per cent iso-octane and 35 per cent normal heptane at a compression ratio of 5.3:1, corrected for barometric pressure, after standardizing the engine with the benzene depreciation standard.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1938.]

[1] Personnel of the Laboratory Procedure Group: A. E. Becker, chairman; C. B. Veal, secretary; W. G. Ainsley; W. H. Hubner; J. R. MacGregor; A. W. Pope; Leonard Raymond; and J. R. Sabina.

Table 1 – Standard Knock-Intensity Data (Averages)

Method	Revolutions Per Minute	Mixture Temperature (Deg. F.)	Spark Advance at 5: 1 Compression Ratio	Observed Compression Ratio	Corrected Compression Ratio	Per Cent C-10
Motor method	900	300	26	5.34	5.31	63.5
Research method	600	Room	22.5	5.19	5.21	64.8
1	900	300	16	5.44	5.42	63.5
2	900	200	26	5.72	5.70	62.9
3	900	200	16	5.81	5.79	62.4
4	900	150	26	5.93	5.91	61.7
5	900	150	16	6.09	6.07	61.4
	600	300	26	4.97	4.92	61.6
	600	300	16	4.87	4.83	60.9
	600	200	26	5.24	5.19	60.3
	600	200	16	5.29	5.26	59.0
	600	150	26	5.34	5.29	57.8
	600	150	16	5.50	5.45	55.8
	800	300	26	5.17	5.14	63.2
	800	300	16	5.10	5.07	63.0
	800	200	26	5.46	5.43	61.9
	800	200	16	5.49	5.49	62.1
	800	150	26	5.63	5.63	60.4
	800	150	16	5.81	5.79	61.5
	1,200	300	26	6.41	6.34	64.5

Changing the mixture temperature, the speed, or the spark advance from the motor-method settings changes the compression ratio required for "standard knock intensity." It was, therefore, necessary to conduct preliminary experiments to determine the compression ratio and the blend of reference fuels that should be used for each set of conditions that it was proposed to investigate.

For all methods except the research, this investigation was done as follows: "Standard knock intensity," as just defined, was obtained in the usual manner under motor-method conditions, with the bouncing pin adjusted to give a knockmeter reading of 60. The speed, mixture temperature, and spark advance were then altered to the desired settings, and the compression ratio adjusted to give the same knockmeter reading as before with the 65 octane-heptane blend.

The blend of secondary reference fuels which matched the

Table 2 – Spark Advance for Maximum Power and Maximum Knock at 900 R.P.M. (Averages)

		300 Deg. F. Mixture Temperature				150 Deg. F. Mixture Temperature			
Reference Fuel Octane No.	Knock-Intensity Compression Ratio	Spark Advance for Maximum Knock	Spark Advance for Maximum Horsepower	Maximum Horse-power-	Compres-sion Ratio	Spark Advance for Maxi-mum Knock	Spark Advance for Maximum Horse-power	Maximum Horsepower	Com-pression Ratio
50	MM+0.5		11.4	2.30	5.37		15.5	3.01	5.92
	MM	22.8	19.7	2.17	4.89	25.2	18.6	2.86	5.42
	MM—0.5		24.2	2.12	4.37		23.2	2.78	4.94
	MM—1.0		26.0	1.95	4.01		28.5	2.64	4.44
70	MM+0.5		12.8	2.35	6.04		14.7	3.06	6.68
	MM	26.0	18.8	2.30	5.53	24.3	17.2	2.98	6.13
	MM—0.5		21.8	2.19	5.03		20.0	2.90	5.67
	MM—1.0		23.8	2.12	4.53		22.0	2.76	5.17
78.8	MM+0.5		13.2	2.39	6.54		13.0	3.14	7.11
	MM	27.2	17.8	2.38	5.97	23.0	16.5	3.04	6.57
	MM—0.5		20.6	2.41	5.50		20.5	3.01	6.10
	MM—1.0		22.2	2.30	5.01		21.0	2.90	5.61
90.0	MM+0.5		14.4	2.64	7.73		12.3	3.25	8.49
	MM	25+	16.0	2.53	7.25	25.7	15.2	3.22	7.97
	MM—0.5		17.8	2.52	6.73		17.3	3.26	7.49
	MM—1.0		19.6	2.44	6.23		19.0	3.20	6.99
99.5	MM+0.5		14.0	2.62	8.89		12.3	3.45	9.78
	MM	23+	14.3	2.73	8.52	20+	14.0	3.40	9.28
	MM—0.5		15.8	2.74	8.06		16.0	3.39	8.78
	MM—1.0		16.6	2.76	7.56		15.5	3.32	8.17

Table 3 – Description of Winter and Summer Fuels

Two series of test gasolines were employed: *a*, a series having winter volatility characteristics and designated by the prefix letter *W*; *b*, a corresponding series of fuels of summer volatility and designated by the prefix letter *S*.

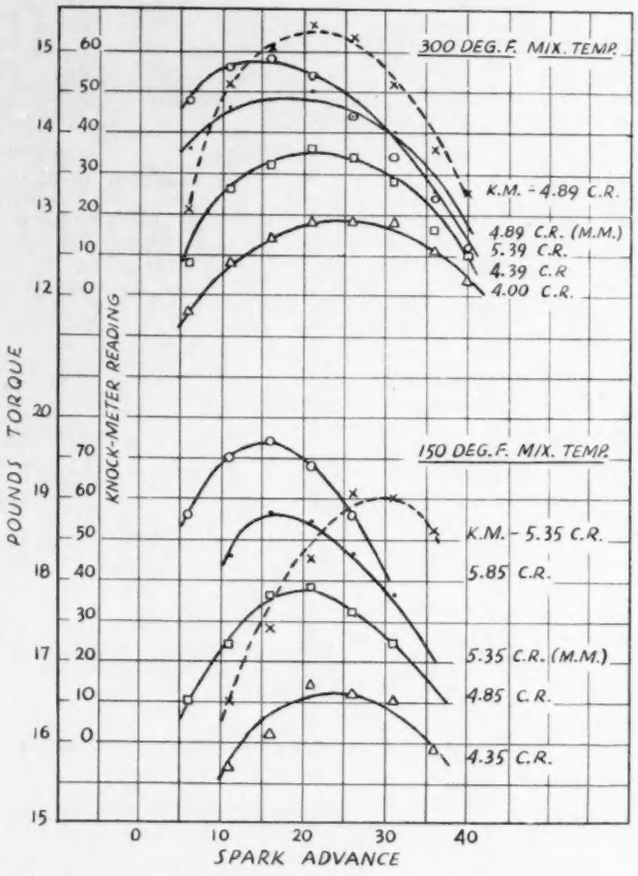
Fuel Designation			Added Material	
Winter	Summer	Type	Winter	Summer
WA-1*	SA-1	Pennsylvania straight-run	$PbEt_4$ (2 cc.)	$PbEt_4$ (2 cc.)
WB-1	SB-1	Mid Continent (50 per cent straight-run, 50 per cent cracked)	None	None
WB-2	SB-2	Mid Continent (30 per cent straight-run, 70 per cent cracked)	$PbEt_4$ (0.2cc.)	$PbEt_4$ (0.6cc.)
WB-3*	SB-3	Mid Continent (70 per cent straight-run, 30 per cent cracked)	$PbEt_4$ (2.2cc.)	$PbEt_4$ (2.8cc.)
WB-4	SB-4	Mid Continent (50 per cent straight-run, 50 per cent cracked)	Alcohol (10 per cent)	Alcohol (10 per cent)
WB-5*	SB-5	Mid Continent (50 per cent straight-run, 50 per cent cracked)	Benzol (18 per cent)	Benzol (18 per cent)
WC-1	SC-1	California straight-run (100 per cent)	None	None
WC-2	SC-2	California straight-run (100 per cent—not same as WC-1)	$PbEt_4$ (0.5cc.)	$PbEt_4$ (0.62cc.)
WD-1		Pennsylvania cracked (100 per cent)	None	
WD-2		Pennsylvania cracked (100 per cent)	$PbEt_4$ (0.4cc.)	
	SD-3	Pennsylvania cracked (100 per cent)		$PbEt_4$ (0.92cc.)
WE-1	SE-1	Mid Continent reformed (100 per cent)	None	None
WE-2	SE-2	Mid Continent reformed (100 per cent)	$PbEt_4$ (0.3cc.)	$PbEt_4$ (0.3cc.)
WF-1		California (40 per cent straight-run, 60 per cent cracked)	None	
	SF-1	California (34 per cent straight-run, 66 per cent cracked)		None
WF-2*	SF-2	Blend of California cracked and straight-run	$PbEt_4$ (0.2cc.)	$PbEt_4$ (0.5cc.)
WG-1*		Mid Continent cracked (100 per cent)	None	
WH-1*	SH-1	Coastal cracked (100 per cent)	None	None
WH-1*†		Coastal cracked (100 per cent)	$PbEt_4$ (2 cc.)	
WH-2	SH-2	Coastal cracked (100 per cent)	$PbEt_4$ (0.48cc.)	$PbEt_4$ (1.6cc.)
WI-1	SI-1	East Texas cracked (100 per cent)	$PbEt_4$ (0.65cc.)	$PbEt_4$ (0.74cc.)
WI-2	SI-2	East Texas straight-run (100 per cent)	$PbEt_4$ (0.73cc.)	$PbEt_4$ (0.83cc.)

* Preferred.
† Plus Lead.

Table 4 – Description of Test Methods

Method No.	Speed (R.P.M.)	Intake-Mixture Temperature (Deg. F.)	Spark Advance (Deg. B.T.C.)	Standard Knock Intensity	
				Compression Ratio	Per Cent C-10 in A-5
Motor	900	300	26	5.3	64.0
Research	600	Room	22.5	5.3	64.5
1	900	300	16	5.4	63.5
2	900	200	26	5.7	63.0
3	900	200	16	5.8	62.5
4	900	150	26	5.9	62.0
5	900	150	16	6.1	61.5
6	900	260	21	5.5	
7	900	250	21	5.6	63.5
L-3	900	260	16	5.5	64.0
8	900	300	26		
9	900	300	17		
10	900	300	10		
11	900	150	26		
12	900	150	17		
13	900	150	10		
14	1200	300	26		
15	1200	300	17		
16	1200	300	10		
17	1200	200	26		
18	1200	200	17		
19	1200	200	10		
20	1200	150	26		
21	1200	150	17		
22	1200	150	10		
23	600	300	26		
24	600	300	17		
25	600	300	10		
26	600	200	26		
27	600	200	17		
28	600	200	10		
29	600	150	26		
30	600	150	17		
31	600	150	10		

Automatic spark-control linkage used with following methods:
Motor, research, L-3, 1, 2, 3, 4, 5, 6, 7.
Spark advance fixed (spark-control linkage disconnected for methods 8 through 31).



Solid lines refer to the torque scale, and dash lines to the knockmeter readings

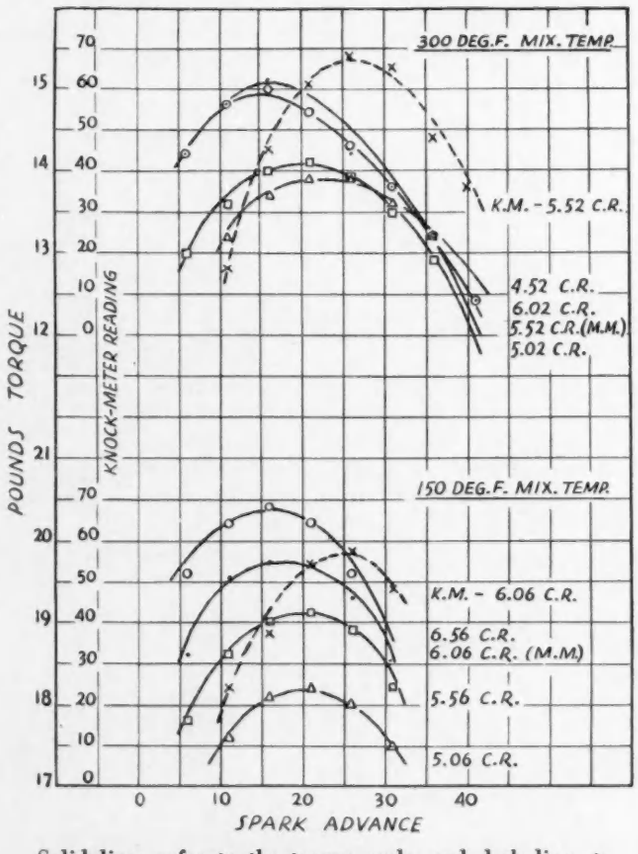
blend of primary reference fuels at this compression ratio was then determined by the bracketing method. Average values of the compression ratios and blends of secondary reference fuels required for "standard knock intensity" under various sets of conditions are given in Table 1. These values were obtained from the data submitted by the laboratories that participated in this phase of the work, and were used in defining conditions to be used for the various test methods outlined in Table 4.

For the research method a different procedure was used, namely, that of determining the compression ratio for incipient knock for the 65 octane-heptane blend and then raising the compression ratio one unit. The bouncing pin was then adjusted to give a meter reading of 60. The corresponding blend of secondary reference fuels was then determined.

Spark Advance for Maximum Power and Knock

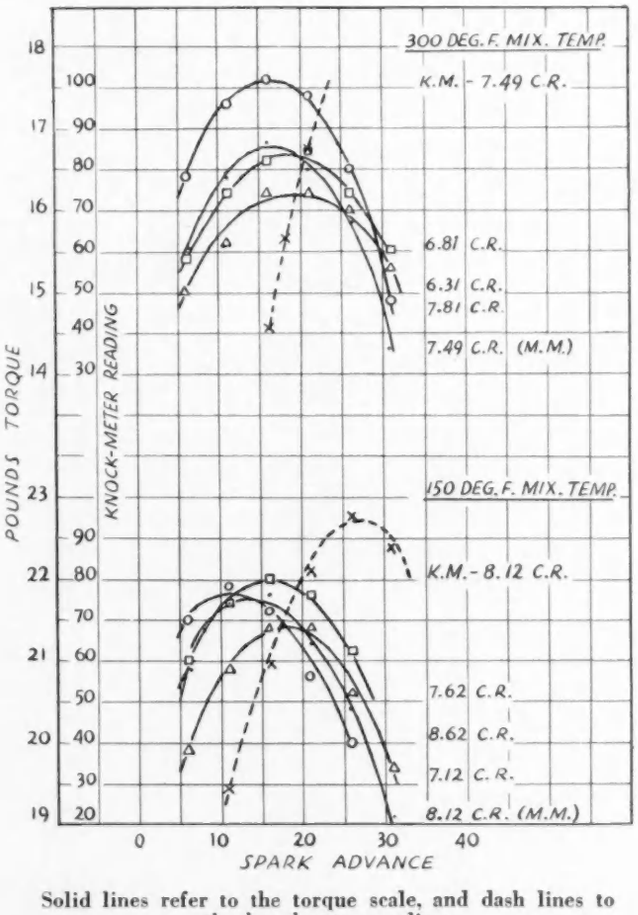
The earlier work of the group was directed toward rating a selected list of the W fuels by methods involving changes of speed and mixture temperature at various spark settings, using

Fig. 1 (*left*) – Spark Advance for Maximum Power and Maximum Knock – Typical Curves for 50-Octane Reference Fuel at 900 R.P.M.



Solid lines refer to the torque scale, and dash lines to the knockmeter readings

Fig. 2 – Spark Advance for Maximum Power and Maximum Knock – Typical Curves for 70-Octane Reference Fuel at 900 R.P.M.



Solid lines refer to the torque scale, and dash lines to the knockmeter readings

Fig. 3 – Spark Advance for Maximum Power and Maximum Knock – Typical Curves for 90-Octane Reference Fuel at 900 R.P.M.

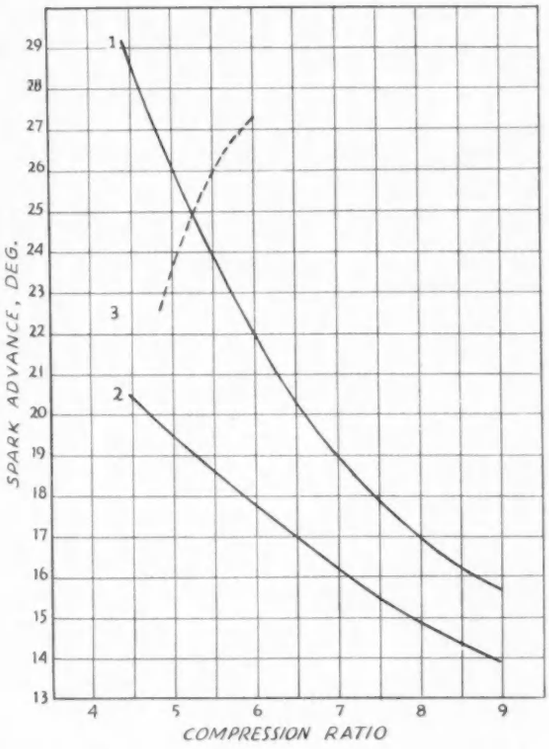


Fig. 4 – Relation Between Compression Ratio and Spark Advance:

(1) With automatic spark-advance control linkage

(2) For maximum power at compression ratios required for motor-method knock intensity

(3) For maximum knock at compression ratios required for motor-method knock intensity

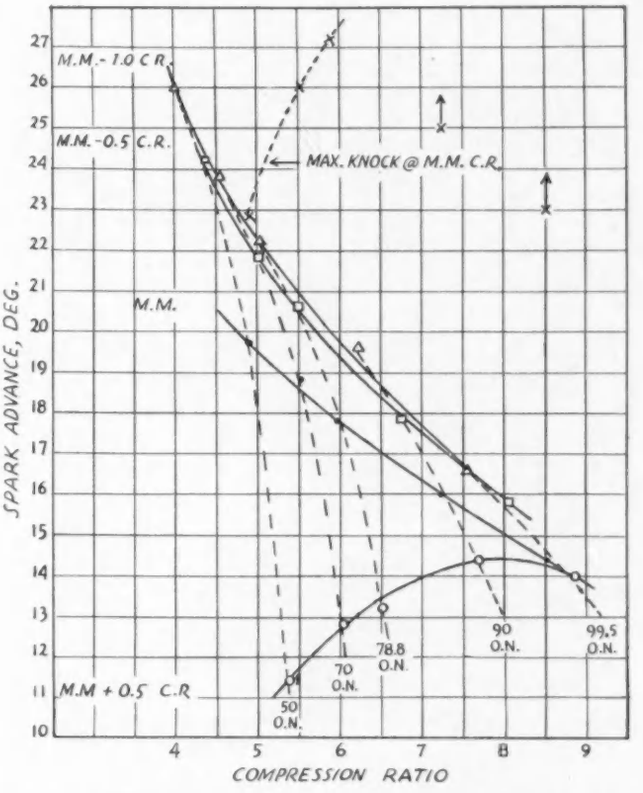


Fig. 5 – Relation Between Compression Ratio and Spark Advance:

(1) For maximum power for four different initial knock-intensity levels

(2) For maximum knock for motor-method initial knock intensity

900 R.P.M., 300-Deg.-Fahr. Mixture Temperature, and Blends of Reference Fuels A-5, C-10, and F-1

the automatic spark-control linkage (see Table 4). It was found that some of the fuels, more particularly those of high octane number, could be rated only with difficulty, or not at all, by those methods for which a less advanced spark setting than that of the motor method was specified. One difficulty appeared to be that, for such a setting of the automatic control, the spark was excessively retarded when testing fuels at high compression ratios.

Although it was known that the linkage of this control is such that the spark advance is ahead of that required for maximum power, there was no definite information available as to the relationship between spark setting, power, and knock intensity throughout the compression-ratio range in use.

To obtain this information, a program was carried out by a number of laboratories to determine the spark advances for

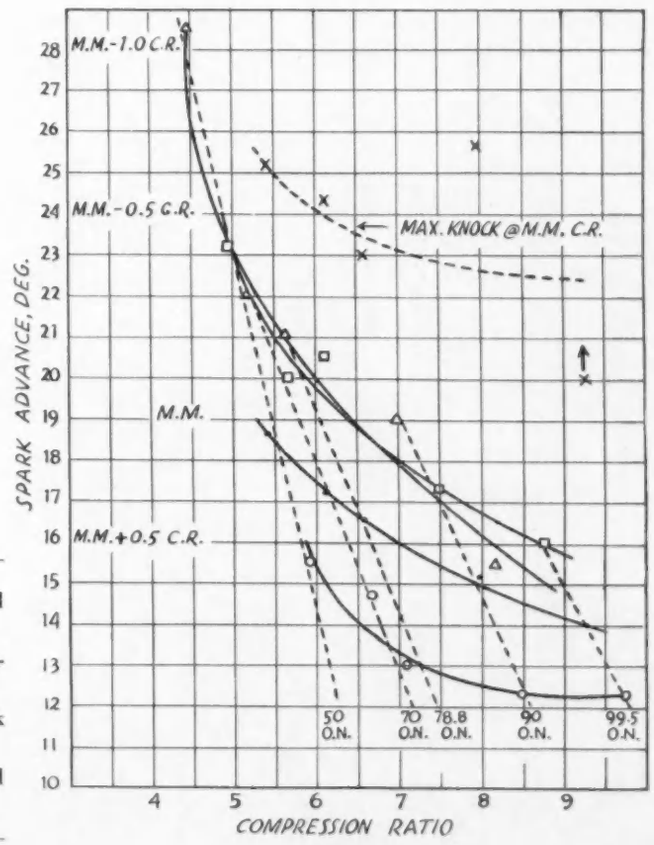
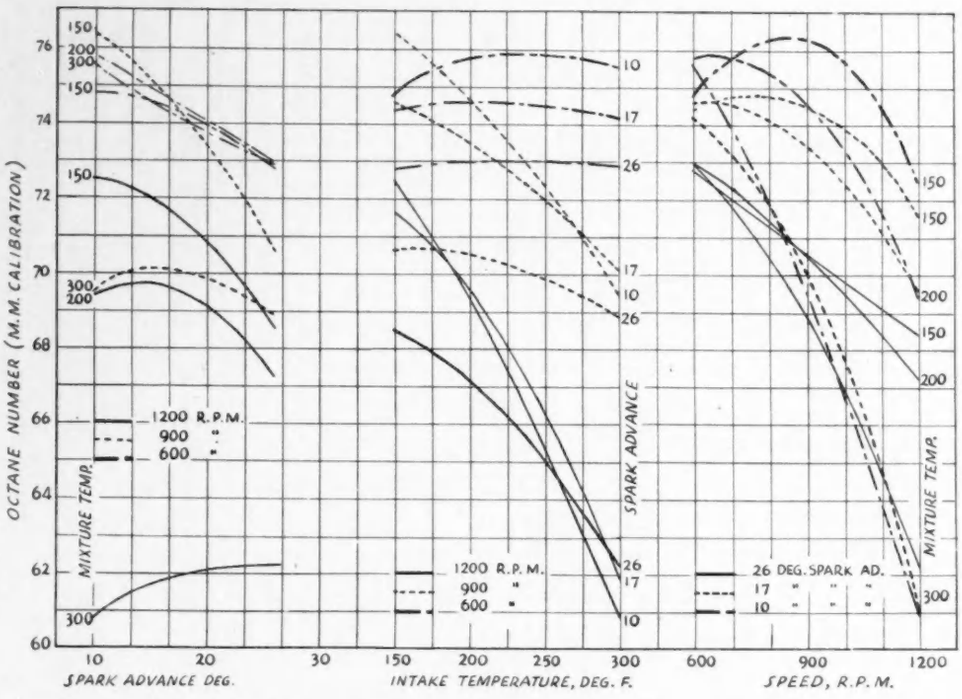


Fig. 6 (*right*) – Relation Between Compression Ratio and Spark Advance:

(1) For maximum power for four different initial knock-intensity levels

(2) For maximum knock for motor-method initial knock intensity

900 R.P.M., 150-Deg.-Fahr. Mixture Temperature, and Blends of Reference Fuels A-5, C-10, and F-1



The effect of spark advance, mixture temperature, and speed on octane number ratings with motor-method knock intensity

Fig. 7 – Fuel WG-1 – Ratings at Constant Spark

maximum power and maximum knock. Secondary reference fuels of 50, 70, 78.8, 90, and 99.5 octane number were used. Most of the work was done at 900 r.p.m., although some data also were obtained at 600, 800, and 1200 r.p.m. These tests were made at both 300 and 150 deg. fahr. mixture temperatures, and in each case at four compression ratios differing progressively from each other by 0.5.

Procedure. – The procedure used was as follows: With each

Table 5 – Average Ratings of W Fuels (Motor-Method Calibration Except for L-3 Ratings)

Method No.	Fuels																			
	WA-1	WB-3	WB-5	WF-2	WG-1	WH-1	WH-1+ 2.0 cc. Lead	WC-1	WD-1	WE-1	WD-2	WE-2	WI-2	API	WB-1	WB-2	WH-2	WB-4	WC-2	WF-1	WI-1
Motor	69.1	77.8	70.8	69.5	70.0	75.6	...	70.8	66.7	67.8	69.5	70.8	70.6	69.3	66.4	69.4	79.8	72.9	71.0	71.0	70.7
Research	68.6	80.2	74.6	71.2	78.7	81.8	...	71.7	69.6	70.6	71.6	73.4	69.6	77.5	68.1	70.8	85.2	77.0	69.3	78.7	73.7
1	69.4	77.2	70.6	69.9	70.9	75.7	...	71.1	66.6	68.4	69.4	71.1	70.4	71.1	66.1	68.7	79.2	71.4	71.7	71.8	...
2	69.7	79.5	72.7	70.8	72.3	77.6	...	71.6	68.5	69.9	71.1	72.9	71.3	72.1	67.7	70.8	82.3	74.5	71.1	74.1	...
3	69.6	79.7	73.2	72.0	75.4	79.6	...	71.7	69.3	70.4	72.1	73.1	70.7	74.9	67.9	71.0	83.0	76.5	71.7	77.3	...
4	69.6	79.8	73.9	71.0	73.4	78.6	...	72.1	68.6	71.3	71.8	74.0	71.4	73.4	67.9	70.7	83.4	76.1	70.3	74.5	...
5	69.0	80.1	74.6	71.7	77.6	80.9	...	72.2	70.1	72.0	73.6	74.3	71.0	77.1	68.9	71.7	78.8	77.3	69.9	78.5	...
6	69.2	78.8	71.8	70.1	72.7	77.3	...	71.4	67.4	68.9	70.7	71.0	70.7	72.1	66.5	69.8	80.6	73.7	70.7	74.1	...
7	69.3	78.8	72.3	71.1	73.5	77.8	...	71.7	67.9	69.4	71.2	71.5	70.6	72.4	66.5	70.3	81.2	74.9	71.1	74.5	...
Motor	69.8	78.2	70.7	69.1	68.9	75.7	...	70.8	67.0	68.1	69.4	70.8	70.9	...	66.6	69.1	80.1	72.3	72.2	71.4	71.3
Research	68.8	79.7	74.2	70.9	76.6	81.8	...	71.9	68.9	70.7	70.9	73.2	69.8	...	67.9	70.9	85.9	75.3	70.4	77.2	73.0
L-3	70.6	79.4	72.0	70.7	71.7	77.8	...	71.8	68.4	69.5	70.7	72.5	71.4	...	66.9	70.1	82.0	73.8	72.5	74.1	73.3
8	69.2	78.0	70.2	69.3	68.9	75.3	83.2														
9	69.6	78.0	70.2	69.8	70.1	75.4	83.3														
10	69.7	77.5	69.9	69.4	69.5	75.5	83.2														
11	69.2	79.0	72.5	70.2	70.6	77.4	86.6														
12	69.3	79.7	73.8	71.4	74.5	79.8	87.8														
13	69.3	79.7	74.1	71.7	76.4	80.6	88.4														
14	69.3	76.6	66.1	66.4	62.2	70.3	81.6														
15	68.8	75.9	66.1	66.5	61.9	70.3	81.0														
16	68.4	75.2	65.5	65.9	60.8	70.1	80.9														
17	69.6	78.0	68.8	68.4	67.2	74.2	84.2														
18	69.7	78.2	70.1	69.6	69.6	75.9	84.3														
19	70.2	77.8	69.8	69.1	69.4	75.8	84.2														
20	69.3	78.5	70.7	69.6	68.5	75.4	85.2														
21	69.3	78.7	71.5	70.4	71.6	76.9	86.0														
22	69.5	78.5	71.4	70.2	72.5	77.6	86.5														
23	68.5	79.4	73.4	71.0	72.9	79.5	86.9														
24	68.3	79.2	74.0	72.7	74.2	81.4	87.9														
25	68.0	79.1	74.4	72.4	75.6	81.6	87.6														
26	68.6	79.2	72.5	71.2	73.0	80.1	87.7														
27	68.4	79.3	74.2	72.2	74.6	81.5	88.8														
28	68.4	79.4	75.2	73.0	75.8	82.2	89.4														
29	66.6	78.7	73.5	70.3	72.8	80.5	87.1														
30	66.8	79.1	74.8	70.9	74.4	81.2	89.8														
31	66.7	79.2	74.8	72.2	74.8	81.8	90.0														
	Approximate Number of Laboratories Reporting																				
1- 7	10	2	11	3	4	11	...	9	9	9	5	5	5	4	3	3	2	1	1	1	...
8-13	10	10	10	10	10	9	9														
14-22	7	7	7	7	7	7	6														
23-31	4	4	4	4	4	3	4														

Table 6 – C.F.R. Engine Conditions for Perfect Correlation of Each Fuel With Its Average Road Value

Fuel WA-1 (Road Rating 67.8)			Fuel WB-3 (Road Rating 77.6)			Fuel WB-5 (Road Rating 74.4)		
Revolutions Per Minute	Mixture Temperature (Deg. F.)	Spark Advance	Revolutions Per Minute	Mixture Temperature (Deg. F.)	Spark Advance	Revolutions Per Minute	Mixture Temperature (Deg. F.)	Spark Advance
600	173	26	835	300	10	600	300	10
600	172	17	900	300	11	600	200	15.3
600	170	10	900	290	10	600	190	17
635	150	10	960	300	17	600	150	21.5
640	150	26	990	300	26	695	200	10
645	150	17	1,200	232	26	760	150	17
			1,200	278	17	835	150	10
			1,200	208	10			
Fuel WF-2 (Road Rating 70.4)			**Fuel WG-1 (Road Rating 73.9)**			**Fuel WH-1 (Road Rating 79.0)**		
600	153	26	600	300	19	640	300	26
600	150	23.5	600	200	20.8	685	200	26
735	300	26	600	150	20.5	720	300	17
805	300	10	635	300	17	730	300	10
845	300	17	700	300	10	735	150	26
845	200	26	835	200	17	900	243	10
900	272	17	900	217	10	900	227	17
900	263	10	900	182	17	900	150	20.5
900	150	25	900	150	18.6	960	200	17
1,050	200	10	960	200	10	995	200	10
1,075	200	17	1,005	150	17	995	150	17
1,165	150	10	1,130	150	10	1,080	150	10
1,200	150	13						
1,200	150	17						

of the reference-fuel blends and motor-method conditions throughout, the compression ratio was set to give a knockmeter reading of 60. The spark-control linkage was then disconnected so that spark advance could be varied independently of the compression ratio. Torque, as determined by a dynamometer, and knockmeter readings were observed for several spark advances. Using the same carbureter setting, similar torque observations were made at compression ratios 0.5 higher, 0.5 lower, and 1.0 lower than motor-method compression ratio for the fuel under test. This procedure was repeated at 150 deg. fahr. mixture temperature.

Typical curves, as obtained by one of the laboratories, are shown in Figs. 1, 2, and 3 for the 50-, 70- and 90-octane fuels respectively. The spark advances for maximum power and maximum knock were read from such curves, averaged for each compression-ratio level as given in Table 2, and used to prepare the curves of Figs. 4, 5, and 6.

Power Vs. Spark Advance

A study of the curves (Figs. 1 to 3) shows that, for a given fuel, as the compression ratio is lowered, the spark setting for maximum power increases. It will also be noted that the spark advance for maximum knock does not coincide with that for maximum power. The divergence increases as higher-octane-number fuels are used (Fig. 4). Stability of knockmeter readings should be greatest at the spark setting for maximum knock, due to the flatness of the curve in this range. The curves for spark advance with the automatic control linkage now in use and for maximum knock cross at 5.3:1 compression ratio, indicating that best reproducibility should be obtained in this range. This deduction is borne out by the general knock-rating experience that greatest reproducibility is obtained in the 65- to 70-octane-number range.

The curves of Figs. 5 and 6 show the relation between compression ratio and spark advance for maximum power at motor-method compression ratio, 0.5 and 1.0 below, and 0.5 above. Interpolation would indicate that an increase of 0.2 to 0.3 over that of the motor method would permit the use of

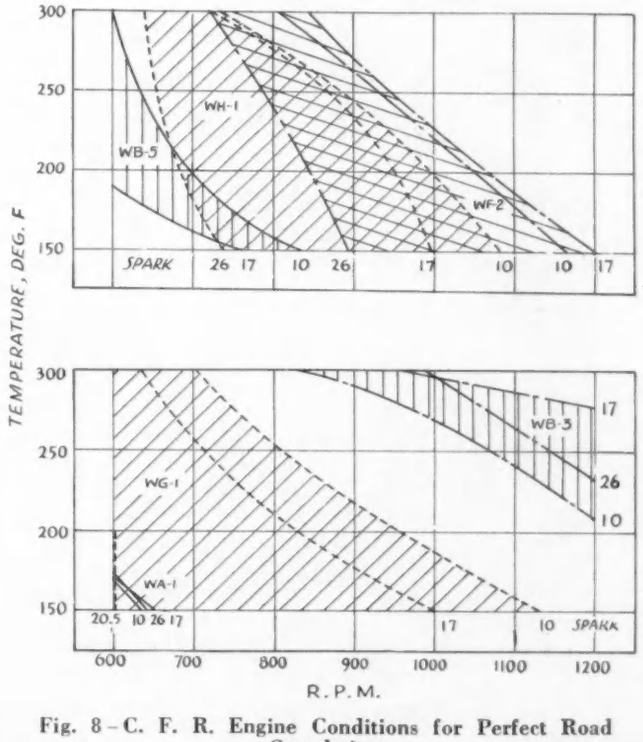
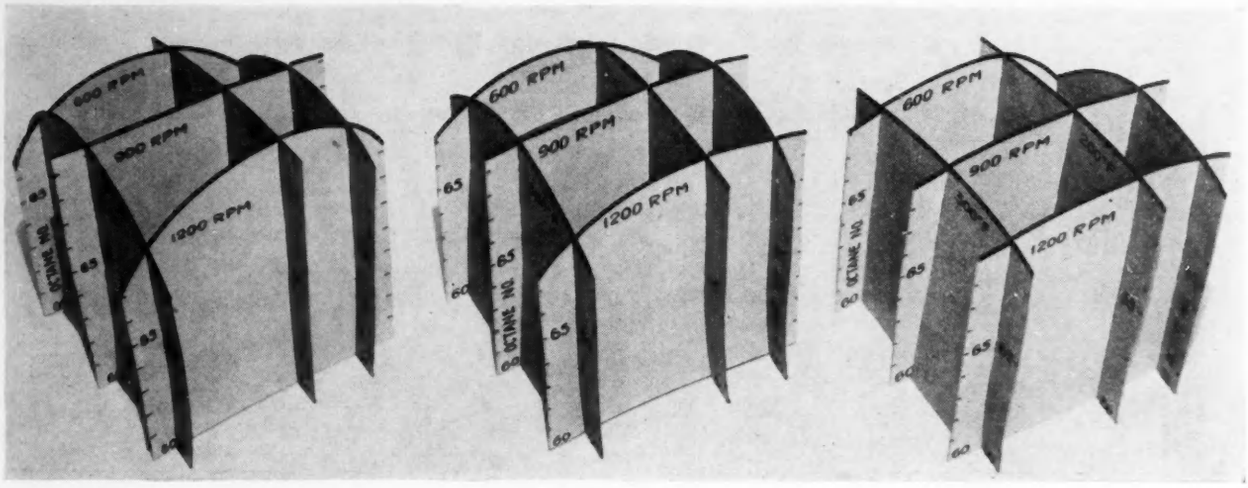


Fig. 8 – C. F. R. Engine Conditions for Perfect Road Correlation



Spark advance, 10 deg. Spark advance, 17 deg. Spark advance, 26 deg.

Fig. 9 – Fuel WA-1

a constant spark advance for maximum power throughout the compression-ratio range investigated.

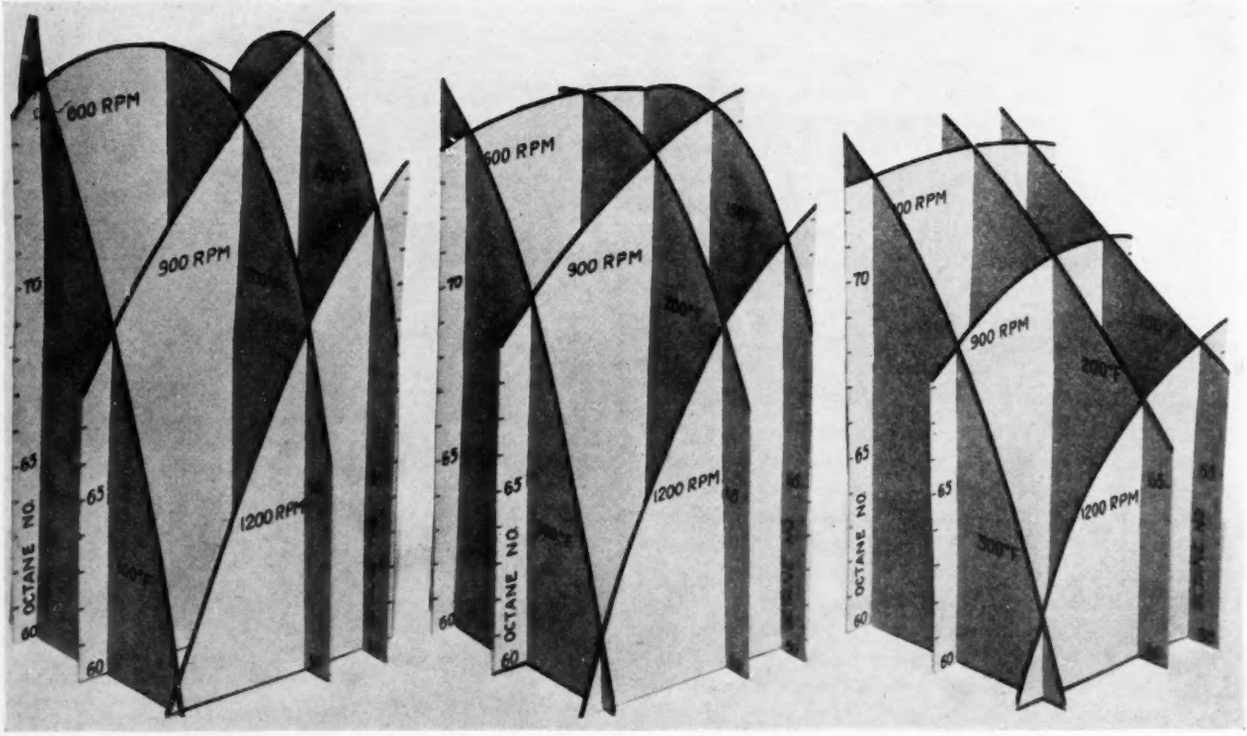
Effect of Engine Variables

This, the major portion of the investigation, was carried out with fuels selected from the W series of road-test fuels. Variables studied were limited to speed, mixture temperature, and spark advance. The various combinations used are shown in Table 4.

Test Conditions. – To obtain comparable results among laboratories, compression ratios and reference-fuel blends to give standard knock intensity under the various test conditions were standardized as shown in Tables 1 and 4. Test conditions other than the ones covered in the two tables were those of the motor method, except for the research method for which they were as follows:

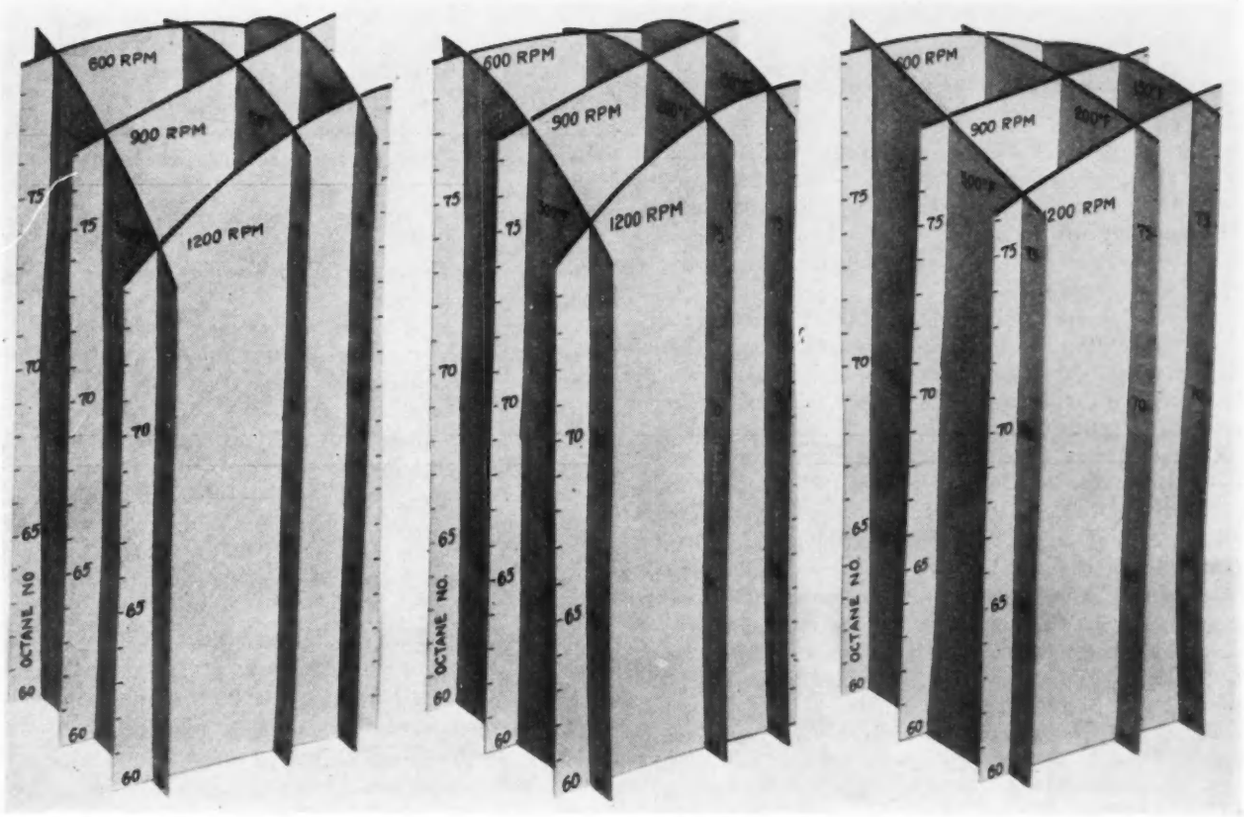
(1) Plain intake valve.
(2) No mixture heater.
(3) Three-bowl carbureter, with adjustable fuel level, with throttle plate and gaskets.
(4) No water pump.
(5) Speed of 600 r.p.m.
(6) Spark advance of 22.5 deg. at 5:1 compression ratio.
(7) Jacket temperature of 212 deg. fahr.
(8) Carbureter air at room temperature.
(9) Air-fuel ratio for maximum knock.
(10) Compression ratio of 5.3:1 (corrected for barometric pressure) for standard knock intensity, using the 65 octane-heptane blend (64.5 per cent C-10 in A-5).

Methods 1 to 7, L-3 (Table 4), and research were run with the automatic spark-advance control linkage. In methods



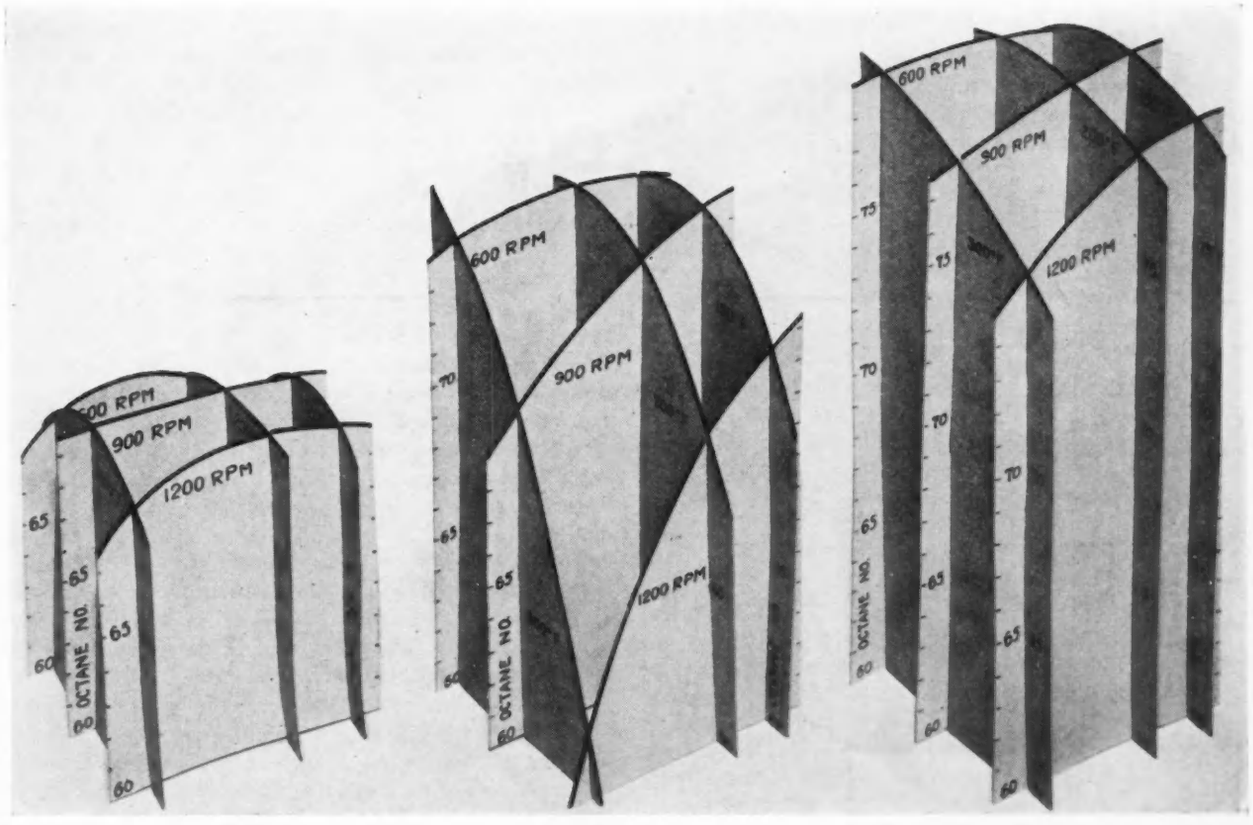
Spark advance, 10 deg. Spark advance, 17 deg. Spark advance, 26 deg.

Fig. 10 – Fuel WG-1



Spark advance, 10 deg. Spark advance, 17 deg. Spark advance, 26 deg.

Fig. 11 – Fuel WB-3



Fuel WA-1 Fuel WG-1 Fuel WB-3

Fig. 12 – Spark Advance, 17 Deg.

8 to 31 the linkage was disconnected, and the spark advance kept constant as shown in Table 4.

Results for Engine Variables

Average values for the ratings reported by the various laboratories are given in Table 5. All ratings were made using secondary reference fuels A-5, C-10, and F-1. For comparative purposes, the ratings have been converted to octane numbers by the motor-method calibration curves – except for the L-3 method, for which calibration curves against the primary standards were available.

The effect of spark advance, mixture temperature, and speed on the rating of one of the seven fuels used is shown by the curves of Fig. 7. Although these results are presented in groups so that the effect of each of the three variables may be studied independently, it is difficult to obtain a clear picture of the combined effect of these variables. Three-dimensional models were, therefore, constructed. Photographs for three of the fuels are shown in Figs. 9 to 12. In each case the surface formed by the boundaries of the speed and mixture-temperature planes covers the range, and the intersections the specific points investigated for the given spark advance and fuel. In Figs. 9 to 11 these surfaces are shown for each of the three fuels at the three spark advances used. In Fig. 12 the relationship is shown for three individual fuels at a 17-deg. spark advance.

To simplify the presentation of the effect of engine variables on fuel ratings, three fuels (WA-1, WB-3, and WG-1) were chosen as typical of low-, intermediate- and high-sensitivity fuels, as expressed by the difference between research- and motor-method ratings. A study of Figs. 9 to 12 leads to the following deductions:

For the non-sensitive fuel, WA-1, it is apparent that the combined effects of speed and mixture temperature are to lower the ratings at low speed and low temperature as compared to the motor method; and also, at 10-deg. spark advance, to a lesser degree at high speed and high temperature. As the spark is advanced, the effect of high speed and high temperature is reduced; whereas spark advance appears to have no effect at low speeds and temperatures.

For the sensitive fuel, WG-1, the combined effect of high speed and high temperature is to reduce materially the rating.

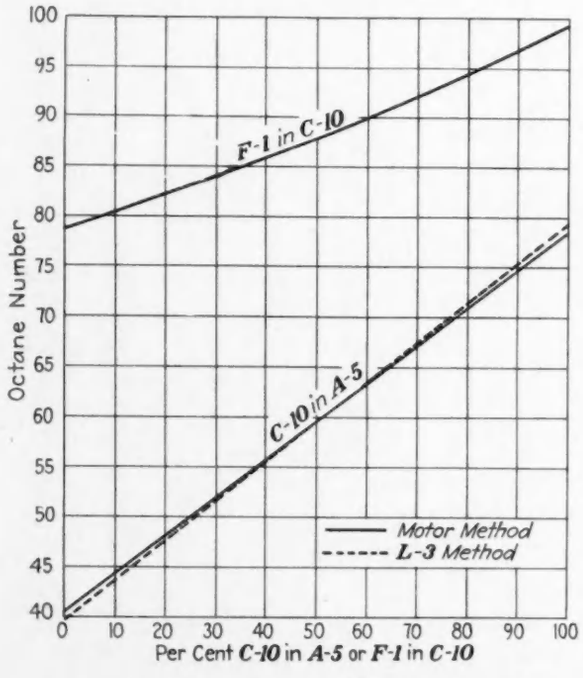


Fig. 13 – Reference-Fuel Calibration Curves

The effect of advancing the spark setting is to increase slightly the rating at high speed and high temperature, but to reduce it at low speed and low temperature.

For the fuel of intermediate sensitivity, WB-3, the combined effect of high speed and high temperature is to reduce the rating. In general, spark advance has little effect.

In the case of a non-sensitive fuel, it will be observed that there are a large number of combinations of speed, temperature, and spark advance that result in approximately the same rating. Thus, the rating of such a fuel can only be changed through a narrow range. As fuel sensitivity increases, ratings are more critical to the variables under discussion. Although there are fewer combinations that can be used to produce a given rating, the range of ratings obtainable by changing engine variables is much greater.

The reactions of fuels of different types to engine variables have a direct bearing on the possibility of obtaining good correlation between road and laboratory ratings. To obtain such correlation with a single-engine method, combinations of engine variables to produce ratings equivalent to road values for each fuel must have at least one common combination.

To determine such conditions for a fuel as WG-1, a line representing the road rating was drawn across the curves shown in Fig. 7. Each point where this line intersects the various curves represents a set of conditions which will reproduce the road rating. For example, a 17-deg. spark advance, at 185 deg. fahr. mixture temperature and 900 r.p.m., is one such set of conditions for 73.9 octane number, the average road rating for this fuel as obtained by the road-test group. Various conditions producing this rating have been tabulated in Table 6 and plotted in the lower part of Fig. 8. This figure also shows the various conditions for the other five fuels for which road ratings are available. The grouping of these fuels has no particular significance, but has been used for illustrative purposes. Overlapping of the areas bounded by the curves pertaining to each fuel indicates common engine conditions to obtain their respective road ratings. Wherever there is no overlapping, as in the case of WB-3 and WG-1, or WF-2 and WB-5, there are no common engine conditions to produce their respective road ratings. Within the scope of this investigation it appears that a compromise is necessary if a single engine method is to be used.

An alternate compromise is the development of a procedure based on two laboratory ratings, as outlined at the inception of this work. Along these lines, methods less severe than the research and more severe than the motor method have been investigated. As described earlier in the report, the lack of uniformity of fuel reaction to engine conditions makes it difficult to obtain a method that will be less severe than the research for all fuels and likewise one that will be more severe than the motor method for all fuels.

To illustrate this point: if 1200 r.p.m., 300 deg. fahr. mixture temperature, and 26 deg. spark advance are chosen for one method, fuels WA-1 and WG-1 will have ratings of 69.3 and 62.2 (Table 5). If 600 r.p.m., 150 deg. fahr. mixture temperature, and 10 deg. spark advance are chosen for the other the respective ratings will be 66.7 and 74.8 octane number. While the rating of one fuel decreases with this change in conditions, the other increases.

The conclusions to be drawn from this work are that correlation between road and laboratory values is not a simple problem. Although it is recognized that there are other important variables than those studied that influence the rating of a fuel, it is believed that the work reported is adequate to permit the defining of a laboratory procedure that will be a material improvement over the present method of testing.

Purchasing, Planning, and Scheduling for Multiple-Model Automobiles

By D. A. Wallace
Chrysler Corp.

TO the casual observer the manufacture of automobiles seems extremely complex and extremely simple. Complex, because of the hundreds of intricate operations which must be so coordinated that the automobile rolling off the end of the assembly line is a complete car in all respects. Simple, because once the formula is worked out and proved, one automobile is very much like the next and there is nothing to interfere with smooth, even production.

That is a fine thought but, unfortunately, it is not exactly correct. Today's automobile production is not confined in every instance to the building of one model. Many factories build a variety of models with a price differential of many dollars.

For instance, our 1938 production consists of three distinct models – The Chrysler Royal, the Chrysler 8 Imperial, and the Chrysler 8 Imperial Custom. With each of these models the usual variety of body types are furnished. Add to this line a total of 72 standard engine combinations, 14 standard paint combinations, 3 standard trim combinations, plus the special-order situation, and you have quite a problem.

To manufacture efficiently multiple models of this type requires a highly concentrated effort not alone between planning and production, but also among engineering, purchasing, sales, planning, and production, and commitment for materials is a matter of vital importance. We have a definite program for meeting this problem:

(1) The Sales Department throughout the building year estimates the number of cars that it will require and how many cars are to be built daily, 90 days in advance.

(2) These estimates are forwarded to the General Production Manager who, after consulting with the management, issues a Material Releasing Authorization to the Planning Department, setting up a manufacturing schedule that will insure maintenance of the estimated sales volume daily without making commitments for material which might later prove excessive.

(3) This detail schedule forms the basis on which all materials are released for manufacture within the plant or for delivery by vendors.

(4) Against this authorization, the Sales Department releases to the Planning Department a building schedule for the first month's production and a tentative schedule for the second month.

(5) The Sales Department follows by furnishing daily in detail the particular colors and trim required for its individual customers and carload combinations seven days in advance of the shipping date.

Naturally, since the brunt of the load is carried in the manufacturing end of the business, the planning division must follow engineering, purchasing, and sales to insure that their activities are so coordinated that it will be possible for the factory to build the cars when and how the Sales Department wants them.

The Planning Department, to protect building dates at the beginning of a new model program, must:

(1) Follow the Engineering Department and make sure that long-time items on the automobile are designed first.

(2) See that blueprints on these long-time items are released to the Purchasing Department first, and that remaining releases follow without unnecessary delay so that the business can be placed by the Purchasing Department.

(3) Follow the Purchasing Department so that, immediately when the business is placed, a copy of the purchase order is furnished the Planning Department.

(4) Contact the vendor and obtain the date for the completion of tools, dies, and so on, and the date that the sample as well as production parts will be available.

(5) Maintain an accurate record of component parts of the automobile showing full pertinent data that tell the complete story of progress on each model.

(6) Notify the management of the dates when various materials will be available, so that man power can be provided for the various operations.

As the period approaches for the introduction of the new automobile, taking into consideration the various time elements involved, a starting month for manufacturing operations is set, and decision is made as to the approximate number of automobiles to be built the first month.

In the second month production probably will be tripled and, in the third month, capacity will be attained.

On this basis, the Sales Department releases a building schedule by models for the first month's building activities, a tentative schedule for the second month, and approximate figures for the third month.

The first month's schedule becomes known as the Master Building Schedule and is used as a basis by the Planning Department, with the second and third months' figures, to notify vendors of the commitment, both for fabrication and raw materials, which they can make to start moving materials into our plants. These materials are released on the month's requirements, in most cases on a five-day bank basis; small parts, ten days; standard parts, in bulk lots.

The Planning Department is now in possession of a Material Releasing Authorization and a building schedule for the first month's production, and the actual job of releasing to the vendor is as follows:

Material requirements against the building schedule, fabrication for the second month's tentative schedule, and raw material releases are calculated for the Planning Department by a division known as Inventory and Order Control Department.

[This paper was presented at the National Production Meeting of the Society, Flint, Mich., Dec. 9, 1937.]

These calculations are based on General Material Authorizations on hand, a breakdown by months established, predetermined banks to be carried established, and the material is released by the Planning Department accordingly.

Calculations are made by machine and, in the releasing of a complete authorization of 10,000 component parts, a total of 150,000 calculations are made.

Addressograph equipment is used to place repetitive information of the vendor's release, the Follow-Up Man inserting shipping dates and quantity to be shipped on specific dates, and the release is mailed to the vendor.

Material Follow-Up

Shortage of even the smallest component part of a car may cause a shutdown of operations affecting hundreds of men and cars. It is quite obvious that proper releasing and follow-up by the Follow-Up Man is of prime importance.

Here is his problem for a single day. The Schedule Division has ordered production of 600 cars. They consist of four- and two-door sedans, rumble seat and business coupes, convertible coupes and sedans, traveler and seven-passenger sedans, to be painted either in any of ten standard colors or in special colors, and to be trimmed in broadcloth, mohair, or leather. There are 72 different engine combinations. Rear axles, emergency brakes, and propeller-shafts may be for overdrive or standard transmissions. Radios, heaters, defrosters, and other special equipment may be required in this combination of 600 cars. The Follow-Up Man must have this material available for that schedule.

His job is made more complex by the fact that there is limited space for the storage of the thousands of parts used. He is limited to certain banks ahead of actual assembly for protection against delays due to traffic or troubles at vendors' plants. Approximately 90 per cent of vendors will ship according to the releasing schedule and cause the Follow-Up Man no trouble, but delays in shipment by the other 10 per cent may make necessary the chartering of an airplane to bring in material so that assembly operations may not be interrupted. Again, it may be necessary for him to request parts to be hand-made until a vendor's trouble is cleared up and, as a last resort, he asks the Purchasing Department to establish a more reliable vendor.

The Manufacturing Division has now been authorized to commit for a certain definite amount of material, schedules have been received from the Sales Department, releases have been forwarded to vendors, material has been shipped from vendors and received in the plant, and we are in position to commence manufacturing operations.

Scheduling

A centralized Scheduling Department, a unit of the Planning Division, controls the entire operation of the plant, starting and stopping each producing division.

The Scheduling Department works with the shop and management in arranging a schedule mutually agreed upon and endeavors to coordinate the different units to the ultimate consummation of the schedule, tying in and regulating the hours to evenly pro-rate the month's business on a daily basis.

On the basis of plant layout of the number of units that can be produced, this division will:

(1) Accept orders from the Sales Department for the proper number of units to insure the predetermined maximum production but not to exceed the quantities authorized by the General Production Manager.

(2) Issue a Daily Master Building Schedule showing in detail the color and trim of body, type of motor, and so on, seven days in advance of the requested shipping date. Copies are sent to key points for preparation of various sub-assemblies.

(3) Prepare metal code plates from the Master Building Schedule and attach them to the bodies. The information recorded on these plates serves to:

(a) Notify the Paint Department what color the body is to be painted.

(b) Notify the Trim Department how the body is to be trimmed.

(c) Reference to schedule and item number directs attention to the Master Building Schedule for all subsequent car assembly operations for the application of all special or optional equipment.

(4) After the code plate is attached and the body enters processing, a teletypewriter installation, the most elaborate one in so far as scheduling is concerned in the industry consisting of 7 sending stations and 18 receiving stations, furnishes all pertinent information to the various departments as follows:

(a) Cut and Sew Departments for preparation of trim to meet the body as it reaches the Trim Department.

(b) Motor Division for assembly of proper type motor.

(c) Cushion and Back Department for these items to be assembled.

(d) Sheet Metal Paint for proper color of fenders, radiator shell, and other attaching sheet-metal items.

As the body is completed and delivered ready for application to chassis, the model, body type, schedule, and item number are telautographed to the Schedule Group. The Car Assembly Line-Up Clerk teletypes all pertinent assembly information to the following points which starts the assembly of the car, together with loading of conveyors with materials from finished stock and sub-assemblies, so timed that they will meet the assembly at the proper moment:

(1) To start the body along the body-assembly line for application of the instrument panels, coils, and so on.

(2) To Chassis Assembly where the frame is placed on the conveyor, motor set, and so on.

(3) To Motor Assembly where the proper motor is placed on the conveyor.

(4) To Sheet Metal Paint where painted attaching parts, wheel, and tire assembly are placed on the conveyor.

(5) Rear-Axle Department
Car-Wiring Department
Steering-Gear Department
Wheel and Tire Department
} For Conveyor Loading.

(6) Final Car Assembly for sub-assembly of radiators and shells, running boards, and special equipment assemblies.

(7) Cushion and Back Department for loading the proper parts on conveyors.

All the preceding conveyors are so timed that materials reach the assembly line at the proper time.

All other materials required are stocked along the assembly line at the point of application.

Car Shipping

Although cars to be built are scheduled in a manner to insure carload combinations being assembled in close sequence, the method of assembling these cars for shipment is as follows:

Cars are delivered to a receiving station of the Car Storage Building, the Sales Department having delivered to this point, 48 hr. previous, Sales Orders covering the complete day's production. When the car arrives at the receiving station it is matched to the Sales Order, after which it is assigned to a specific location in the Storage Building to await the arrival of the remainder of the cars on the order. This job is handled by pulling a four-way shipping ticket from the car and marking on the back of this ticket the location to which the car has been assigned, and then attaching the ticket to the Sales Order.

When the final car on the Sales Order has been received, the orders are turned over to the Domestic Loading Dock if they are to be forwarded by rail; to the Drive-Away Department if they are to be forwarded by truck, by boat, or delivered to the customer at the plant; if the cars are to be exported, they are delivered to the Export Shipping Department where they are assigned to a Foreign Order and prepared for loading.

In recent years a great many of the railroad companies have equipped a number of freight cars with an automatic steel loading device for automobiles and, upon receipt of a completed order when this special equipment is used, the loading division must be careful to assign the shipment to a freight car owned by a railroad that serves that particular destination. This type of equipment is used practically 100 per cent.

It is important that the Shipping Division be able to furnish information rapidly to the Sales Department concerning the status of any order to permit this information being given to the field upon receipt of long-distance telephone calls, telegrams, and so on. Immediately when a shipment goes forward, this information is teletyped to the Accounting Division which enables it to invoice cars on the same day that they are forwarded.

It is also the responsibility of the Shipping Division to hold shipments upon requests of the Sales Division, returning them to production when changes in equipment have been requested by the customer, or transferring the cars from one order to another as may be required to meet the conditions which face the Sales Department.

At the end of each day, complete inventories are furnished the Production, Planning, Accounting and Sales Divisions, indicating the number of cars on hand, and their status in relation to the Sales Orders. In this manner, each department concerned is kept advised currently as to the status of our finished product.

Machining

There are numerous Machining-Department activities which do not tie in with the Daily Master Building Schedule, except for the total units required for the day's program.

The principal departments are the Motor, Axle, and Transmission Machining Divisions which are set up to provide each day sufficient cylinder-blocks, crankshafts, camshafts, axle, and transmission parts to meet the total cars to be built. It is only in the assembling of these parts into complete motors that the 72 combinations are accomplished.

There are other divisions such as Car Wiring Harness, Steering Gears, and Oil Pumps, lined up in the same manner, and their operations are not made as complex by variety as are body-building and car-assembling.

Painting Operations

In the procedure outlined in preceding paragraphs, we have dwelt on the importance of an efficient method of follow-up, material handling, and shop scheduling. Painting this variety of models and body types the colors specified, with an assortment of types traveling down the same production line grouped in carload-loading combinations, is just one phase of the many operations involved in our manufacturing program, but it is one worthy of a broader explanation.

Bodies are scheduled on our Master Building Schedule from individual sales orders, grouped by body types without regard to color, on a definite hourly basis. The Paint Department totals the individual paint combinations appearing on the Master Building Schedule and arranges for sufficient paint of the various colors to be mixed and available to paint the bodies as shown on the Master Building Schedule.

Body code plates are made and grouped in the same order as shown on the Master Building Schedule and, as the code operator applies the plates, he spreads all special painting jobs over the entire production for each shift so that the same number of special paint jobs flow into the Paint Shop each hour.

Metal-clean is the first operation in the Paint Shop. Dirt, oil, and all foreign substances are removed with a cleaner containing phosphoric acid. Water and other chemicals are also properties of this cleaner to keep it from evaporating rapidly so to give it sufficient time to remove oxidation products and oil and to remove any other foreign substance which might be harmful to the painted surface.

The body is washed automatically with a mixture of hot water and cleaner that is sprayed over the job through high-pressure pumps and then rinsed with clean hot water at two separate stations to insure a perfectly clean surface. The body continues into the Bonderite spray where a heavy coating of Bonderite is sprayed automatically on the surface of the body and rinsed automatically with cold water to harden the coating of Bonderite. As an extra precaution to insure that the body is perfectly clean, it passes through another warm-water rinse to which a small percentage of chromic acid has been added to destroy and remove alkali that may be present in the cold-water rinse. The body is dried by steam and induction heat before the prime coat is applied.

A coat of red-oxide primer is next sprayed over the surface both inside and outside, then the body is inspected for file marks, solder pits, strainer marks, or other metal defects, and the job is glazed where necessary. As the body moves down the conveyor line, two separate coats of oxide surfacer are sprayed on the outside, with a 5-min. period for drying between coats. This material fills the small file marks and any metal defects that have not been glazed.

Body-sealing and silencing operations follow. A heavy black sealer that will not harden to the point of losing its elasticity is forced with a 200 lb. per sq. in. pressure between metal joints, making the body dust-proof and free from slush and drafts. A thinner mixture of the same material is sprayed on the under side of the steel top and cowls. Silencer pads are applied by rolling the pad with a heavy roller, imbedding it in the silencing material. Silencer pads are then placed on the floor of the body, and the job proceeds into an oven where it is dried for 75 min. at a temperature of 260 deg. fahr., thus vulcanizing the silencing pads to the metal floor.

The body proceeds to the sanding line; is cooled with an automatic spray of cold water; surface-sanded with 280-C sandpaper and water-finished with 360-C sandpaper to remove any scratches in the surfacer. The body is then blown out with compressed air and passed through an oven heated by steam and induction heat to insure that it is perfectly dry. The surface is cleaned and sprayed with black lacquer sealer which makes small defects such as slight file marks visible. It is then inspected and spot-puttied with pyroxylin putty and sanded down to an even surface. To make for a more uniform job, all spots that have been sanded are spot-sprayed with black lacquer sealer.

From this point the body goes first to lacquer spray, and this is the first operation affected by the paint code.

Each lacquer spray gun is stamped with a paint code so that all the operator has to do is to select the gun with the same number as that which appears on the body code plate.

As an extra precaution to insure painting to code, a duplicate of the paint code number on the code plate is placed in the windshield opening so the spray man can readily identify the proper color that the job is to be painted.

The body passes through a battery of three spray booths, each booth operated with an air circulating system, having a down draft around the outer edge and a suction fan in the stack above, removing fumes and spray dust from the booth to the open air and not permitting spray dust from one color to settle on any other job.

Upon completion of lacquer spraying operations the body passes through an oven and is dried 1 hr. at a temperature of 160 deg. fahr. It is cooled by automatic cold-water spray, insuring thorough hardening of lacquer for sanding and polishing operations.

The outside surface of the body is sanded with water and 320 wet or dry sandpaper, removing spray pits, orange peel, and other minor defects. The water is then blown off by compressed air, and polishing operations are begun. This operation is done by applying a polishing paste over the surface, using a portable motor running at 1800 r.p.m. driving a metal disc to which is attached a sheep's wool polishing disc to produce a high luster.

It might be of interest to you who are not familiar with the situation that a process for precipitating and recovering all the solids from excess paint-spray operations has been developed by the Chrysler Corp. This process has been licensed to many large lacquer and synthetic-enamel users who are recovering an average of one-third of all paint sprayed at a cost sufficiently low to result in a handsome saving to the users. The material recovered is of exactly the same quality as that originally purchased. However, where various types of colors are used, there is naturally a mixture of these colors in the recovered material, and ways have been found to replace the original material in locations where color is relatively unimportant, such as ground coats, under fenders, inside wheels, crankcase sealers, and so on.

This process has made the handling of refuse from the Paint Department a much simpler, as well as much safer, proposition, and keeps the Paint Department in a much neater and more orderly manner for handling materials.

Time for Complete Building

From the beginning of processing operations in the Body Framing Department, the time required to finish a body completely totals 14½ hr., for finishing and painting sheet metal, 12 hr.; the assembly of an engine, 2½ hr.; and, from the start of the frame and chassis assembly operations to the finished automobile turned over to the Traffic Department for shipment, 1¾ hr.

The Manufacturing Division accepts orders from the Sales Department for carload combinations, and so on, and promises to meet shipping dates on a basis of a seven-day schedule delivery. Deliveries of the finished automobile after receipt of order and commencement of framing operations is usually handled on a three-day basis, but seven days is required for the necessary manufacturing bank.

Material Handling

The activity of the Material Department must be coordinated closely with the Follow-Up and Scheduling Divisions of the Planning Department, particularly when production involves multiple models and when storage space for inventory is limited to a three- and five-day bank. Best results in this respect have been accomplished by including the Material Department as a branch of the Planning Division, working under the supervision and direction of the Supervisor of Planning and his Assistant.

With a minimum available inventory, the requirements of the Material Division are speed and precision – speed in reporting material receipts to the Follow-Up Man and the dispatching of material through the various stations necessary to point of usage and precision in the identification of material and its routing to the correct plant location.

Immediately when material is received, receiving reports are teletyped to the Planning Department in triplicate, serving Planning, Accounting and Traffic Departments. Through this teletype set-up it is possible for these departments to know in a very few minutes that material has been received.

Material is routed from the Receiving Department to Inspection, then to a routing station where a routing specification card is used for reference by the dispatcher; then it goes to the processing department or department depot. In many cases inspection is performed at the unloading station, in which case the material is routed direct to the processing or department depot and not through the Receiving Inspection Department.

Stockchasers are assigned to a given department or section. They supervise the handling and storing of material in their respective depots, both when it is transported into the depot and when transported from the depot to the processing operations.

Stockchasers maintain a three-day shortage list, also what is known as a "hot-shot" shortage list. This "hot-shot" shortage list is for items whose shortage shows possibility of becoming serious. Stockchasers contact with the Planning Follow-Up Department through a Shortage Contact Clerk in the Material Supervisor's office. Setting up this Shortage Contact Clerk permits of a central contact for the entire Material Handling Department.

Engineering changes are checked and effective points accomplished by stockchasers through an engineering change record set-up in the Material Supervisor's office.

Inventory Control

A difficult problem in most automobile manufacturing plants is the proper control of inventory to insure the correct material usage, maintaining predetermined banks, immediate removal of obsolete material, and to avoid obsolescence, particularly at the end of the model's manufacture.

The Inventory and Order Control Department, a division entirely separate from the Planning Department, has the responsibility of constantly checking usage and inventories to insure that they are kept within the original quantities authorized for a given period and in direct relation with the Daily Building Schedule. A complete monthly check is made to see that quantities released are strictly in accordance with the authorized calculated quantities, thereby assuring a balanced inventory to the lowest possible point.

This explanation briefly covers the methods used in the Chrysler Division of the Chrysler Corp. The present system represents years of development. We have incorporated the latest and best known methods in planning, shop scheduling, and building operations.

All manufacturers have a system to fit their individual requirements. This system is necessary to produce the volume and variety of product required by its customers.

Regardless of the system and methods followed, coordination among divisions to permit the building of the schedule to the individual order to the specifications furnished by the customer, and to meet promised shipping dates is imperative.

To accomplish this coordination, the manufacturing organization must be versatile. The plant, its equipment, and its layout must be arranged to permit the free flow and identification of the variety of materials used in the building of multiple models.

All these things must be considered but, in the final analysis and the most important of all, is the necessity that a quality product be produced. Quality cannot be sacrificed for volume. Volume and quality are essentially one and the same thing and must be considered as such.

From the moment material arrives at our receiving dock until the automobile is loaded securely in the freight car or delivered to the customer at our gates, quality is the watchword.

Semi-Trailers Versus Six-Wheelers

By J. G. Moxey
Transportation Engineer, Sun Oil Co.

ALTHOUGH dealing with a subject of unlimited breadth, this paper touches on the primary guides for an operating engineer in the purchase of automotive equipment from a practical standpoint, treating only lightly the operating phase of the subject. Torques, engine displacements, ability factors, gross vehicle weights, and recommended practices, together with their relative relationship one to another, are discussed in detail, and also are shown in chart form as a ready guide for a prospective purchaser.

The economical fields of operation of four-wheel trucks, six-wheel trucks, and tractor semi-trailers, together with basic operating costs, acting as a further purchasing as well as operating guide, are brought out, the general conclusion being that each type of transportation under discussion has a definite economic field of operation and should be held definitely in its respective field as indicated advisable by operating surveys.

IT is indeed with considerable misgivings that I accepted the invitation to speak on the subject assigned, which was further defined to be a treatise "giving a complete exposition of exact consideration which should govern the operator in the *purchase* and *usage* of each type of transportation". Needless to say, the latitude of coverage permissible under the assignment is unlimited and allows anyone handling such a subject considerable poetic license in handling the matter.

At the start I wish to thank the many contributors to the subject matter contained herein, who have given the necessary time and service in preparing much of the basic information to be presented.

It is recognized readily that foresight based on fundamental accepted principles of truck transportation and its application to the road, cannot help but draw from any specific problem the reasonably correct solution, and we shall endeavor to unfold, in the short space allotted and in not too technical detail, the necessary method for handling the subject matter in definitely determining the proper *purchase* and its application to the road of the proper and necessarily, the most economic type, of transportation.

The subject matter carries with it considerable controversial phases and features of each individual problem as a whole. There are many lines of thought, and actually as many schools, offering either one or another line of procedure toward the ultimate solution of the problem.

We must take for granted that there is at the command of a prospective individual, a complete first-hand analytical study of the particular hauling problem, as well as all ordinary available literature offered by vendors of various types of automotive transportation, to say nothing of those far-seeing concerns selling transportation merchandise who maintain at their factories and many of their branches, and put at the disposal of their customers, at no obligation, the services of transportation engineers, in which category many of us care to throw ourselves in the conduct of our respective activities.

Anyone attempting the handling of such a paper, to meet satisfactorily all requirements and answer all questions arising in the minds of interested parties, both sellers and users, has an impossible task. It is my desire to bring such facts and figures before you on which to build the structure, be it ever so large or small, for a commercial fleet selection and operation. With a concise presentation of facts we hope to be able at least to bring before those in the activities of trucks, such matters in intimate detail that, to the best of our knowledge, have never been attempted before and, if the time and energy put forth herein will do nothing more than promote within the field of manufacturers and users matters for intimate further discussion, our time has been well spent and the cause justly served.

Much of the subject matter contained herein consists of abstracts from information kindly furnished during the preparation of this paper.

Introduction

The subject of semi-trailers versus six-wheelers ultimately carries with it the general inference that the types of transportation considered deal with heavy-medium- and certain heavy-duty units, and in no way touches the so-called light-duty field. However, it is impossible to confine ourselves to two types of machines in the subject matter without some reference to the four-wheel vehicles.

In handling the subject matter we will take it for granted that there are many diametrically opposed lines of thought in many points of question. A glance through the literature and sales presentations, I believe, gives no definite and clear line of thought to guide one in the purchase and usage of either of the two types of machines in question, and its correct application to a specific load.

We shall be bold enough to make the statement that there does not seem to be any "versus" side to the problem. Semi-trailers versus six-wheelers does not altogether seem to fit the order of the day, but rather substitute the relative suitability, maneuverability, and transferability, of one type of equipment versus the other, to say nothing of avoiding as far as possible the customary pitfalls which are the lot of most purchasers of equipment today, only to find that an error in judgment has been made *after* the expenditure and the vehicle actually

[This paper was presented at the Metropolitan Section-Regional Transportation and Maintenance Meeting of the Society, Newark, N. J., Nov. 10, 1937.]

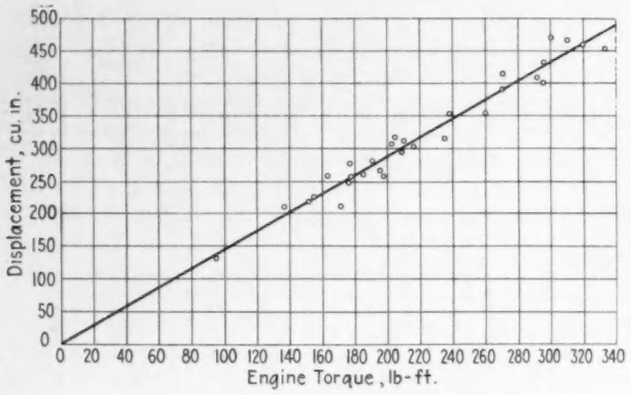


Fig. 1 – Engine Torque Vs. Displacement

entered service. To guard against such calamity the three principal pitfalls to which all of us have been subjected are named:

(1) The ideas and opinions of our personal well-wishers in our own particular business activity.

(2) Statements of manufacturers, possibly in the form of advertising fronts, giving satisfactory performances of satisfied users or, possibly, over-zealousness on the part of sales representatives, whose sole and primary object in life is to make a living and possibly sell something, based on not so much an intimate knowledge of your own business, but rather to discharge their duties as a sales representative.

(3) Attempting to make a direct comparison, even within the same industry, of one successful operator and taking for granted that that same type of equipment will, of necessity, successfully solve your own problem and bring about the much desired economic condition within your own fleet.

We hope to show you that, in the selection of two types of equipment, there are certainly underlying principles which make them two entirely different types of motor vehicles, to be employed in entirely different classes of transportation, and we trust to be able, in this presentation, to take advantage of the opportunity to do motor transportation at least a small amount of good by reducing this semi-trailer versus six-wheeler controversy to a matter of sane common sense, with an engineering and practical background to solve the problem leaving, as far as possible, any guess work out of the solution.

The principles, although under discussion for years, are still unchanging as time goes on.

Under present State laws we do not consider that two types of equipment are necessarily competitive. If we have a hauling problem and were contemplating the use of semi-trailers or six-wheelers, we would have to be governed by conditions at hand, some of which are the nature of haul, length of haul, States where hauls are to be made, typography of States, and many other phases which of necessity must be known to answer the problem properly.

It has been the experience of many manufacturers that individual truck operators are not concerned with statistics showing advantages of one unit over the other but, when an individual buyer is in the market, he usually buys what he personally believes to be the best unit for his needs.

We will now take up the important and real subject matter of this discussion on how a prospective purchaser is to be guided in the acquisition and application of transportation to a hauling problem.

Premise

The basis on which we have attempted this presentation is that it will, of necessity, be non-mechanical. No effort will be made to prove facts or opinions expressed relative to merits of, for example, single-drive against dual-drive rear axle, single- against double-reduction rear drives, relative merit of variable load application of idling axles or, in the case of semi-trailers, location of fifth wheels, height of fifth wheels, relative wheelbases, and so on, any one of which would be sufficient subject matter for presentation within itself.

No attempt will be made to analyze any one specific hauling problem as, without most intimate first-hand knowledge of a proposed operation, a satisfactory answer cannot be drawn. All data, however, will have to be premised on something tangible within the type of transportation under discussion, and the principal points of tangibility will be that all comparative operations must be kept within the normal machinery capacity of units, and that performance must, of necessity, be equivalent in different types of transportation as applied to any specific loading or hauling problem. In other words, the performances as measured in speed and ability factor will be considered as uniform in units under discussion, and there will be used as a basis the much discussed, and somewhat accepted, minimum limit of 4 per cent net ability factor at 20 m.p.h. It is to be hoped that the trend of motor-vehicle legislation of the future will result in a more uniform performance ability established for all types of machines, whether four-wheel, six-wheel, or tractor semi-trailers.

We will take for granted in all operations that, the larger the unit, the less will be the unit operating cost, provided the vehicle is kept busy within the limits of its economic performance.

Torque Versus Displacement

To start, of course, we all know that the engine in our truck is the source of power ultimately to take the load over the highway. These engines have certain characteristics, both as to ability and performance, and fortunate is the operator who can recognize the difference between torque and horsepower for different size engines as, in the field of endeavor which we are attempting to cover, torque seems to be the important factor of successful operation. Referring to Fig. 1 we show the torques of various engines as obtained from the listing of specifications appearing in a recent issue of the *Commercial Car Journal* and, in passing, it is interesting to note that the same size engine listed by different manufacturers carries with it a fairly wide difference of torque rating as assembled in the different trucks.

You will note from the chart that the performance of practically all of the engines seems to cluster around the line indicating the average performance, which has a slope of approximately 7/10 to 1, and certainly has proved in actual service to be, to all intents and purposes, the approximate torque obtained from engines in practical operation, certainly within the limitations of accuracy of the ordinary observing instruments. I refer to the practice of road gradient surveys, particularly in the heavier territories, establishing the percentage of the total distance to be covered of up-grades collected according to their individual grade, for accurately determining the selection of the proper size engine.

In passing we wish to remark that some authors advocate use of a slope of 0.65 instead of 0.70, but this value does not seem to be quite as much as the machines have due them when actually put into service under normal operating conditions.

We state the foregoing, bearing fully in mind the much discussed characteristic of increased horsepower with decreased torque, as well as their relative relationship, as observed in the gasoline engines of today.

Transmissions – Four- and Five-Speed

Let us digress for a moment and take up the matter of transmissions as applied to the subject, transport activity.

They are made mostly in either four or five speeds, whether the latter be underdrive or overdrive is not within the scope of this discussion, but we will take for granted that they have been designed properly to cope with the requirements of the ordinary user. The five-speed type is used invariably in long-distance heavy-duty service, first speed being practically what is frequently called the pull-out speed being used, of course, to get massive pieces of equipment in motion, when it is shifted immediately into second gear and, from there on, it is practically a four-speed transmission. In other words, it is a four-speed transmission with a pull-out feature lower down in the range. At this point reference is made to the discussion on the subject of transmissions presented by B. B. Bachman at the recent Semi-Annual Meeting of the S.A.E., and published recently by *Automotive Industries*.[1]

If we will take the normal low ranges of transmissions for starting with the premise that these speeds are simply for getting the vehicle under way, with third, fourth, and fifth speeds in a five-speed transmission, as against second, third, and fourth speeds in a four-speed transmission, for running, we can draw the following information by mere mathematical calculation: If we assume that the 4 per cent net at 20 m.p.h. is to be obtained in third gear in a heavy-duty machine, we obtain the following:

Ability Factor, per cent		
Net	Gross	M.P.H.
4.00	5.25	20
2.25	3.50	30
1.38	2.63	40

The basis of the preceding table is the survey made by the *Commercial Car Journal*, and seems to fix rather definitely the desirability of this type of performance being embodied in machines of the future. To those seriously interested in further study of this most interesting phase of transportation, we can refer the October, 1937, issue of the *Commercial Car Journal*.

G.V.W. Versus Engine Displacement

Fig. 2 has been prepared, based on mere mathematical calculations, to show the relation of the cubic inch displacement of the engine necessary to move certain gross vehicle weights over the highway at an equivalent speed of 20 m.p.h. in second or third gear, depending on whether a four- or five-speed transmission is used; it acts as a ready guide for the specifying of a minimum size engine or approximate ability factor, the latter being mostly used, when putting out inquiries for automotive equipment. On to this chart we have also placed the 4 per cent net ability factor referred to, after adding 1.25 per cent for a so-called 25-lb. road, representing, of course, the normally accepted road resistance factor as encountered under good road operating conditions.

[1] See *Automotive Industries*, July 24, 1937, pp. 117-118, "Vehicle Performance," by B. B. Bachman.

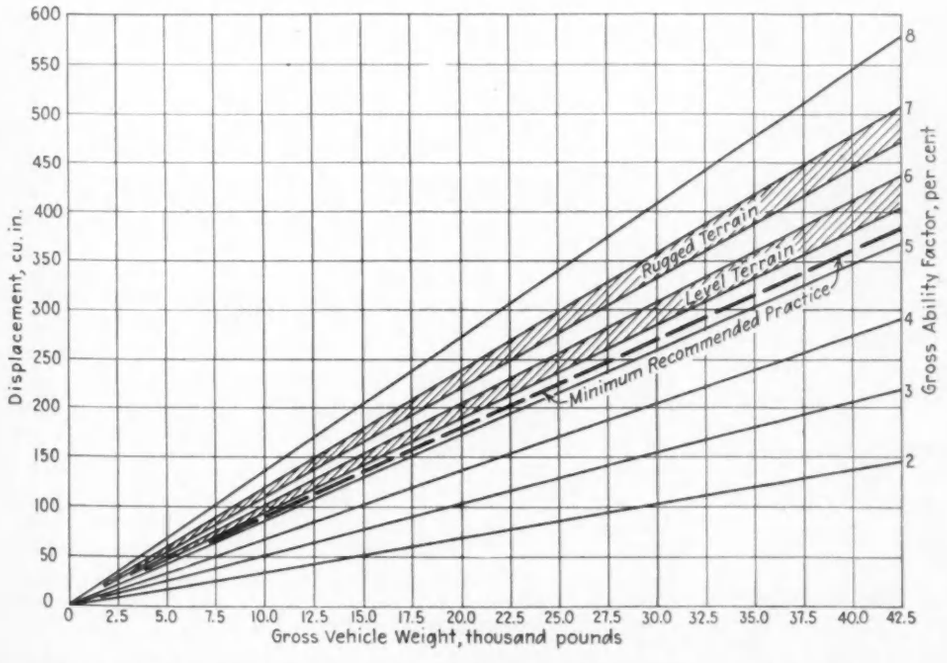


Fig. 2 – Gross Vehicle Weight Vs. Displacement Showing Gross Ability Factor

Selection of Proper Size Engines

Assuming the operator is desirous of having his equipment up to the limit specified by referring to Fig. 2, we have taken a portion of the curves indicated in the previous chart and have shaded in the approximate ability factor normally desired by experienced operators for their equipment in the two types of territory usually encountered, namely, level and the more rugged type of territories, adding for long-distance work at least ½-deg. ability factor at 30 m.p.h. where the body and load are carried directly on the unit. For any specific gross vehicle weight under consideration, it is apparent that the smaller-engined transportation usually is favored for covering the less rugged territory but, as to which engine size is to be selected, we leave entirely to the results obtained by the road gradient survey of the particular routes over which the specific machine is to operate. By making calculations of ability factors of machines with various size engines in various transmission speeds, and operating within reasonable range of engine speed, and then calculate for any run the amount of time to be spent in each gear with each proposed size of engine, the correct size can be arrived at and definite calculations made as to the return on additional investment necessary to supply, in more rugged territories, machines of the proper ability factor. To make better time on the highway is frequently paramount, and some of the results to be obtained from such a study will be surprising.

The successful operator of today, particularly in the heavy-duty field, by placing equipment on the highway within the preceding ranges, will save himself considerable embarrassing moments and much criticism from other users of the highway, by being able with such equipment to "keep up with the procession," to say nothing of forestalling the many thoughts and moves in the past among the legislatures of our various States in endeavoring to keep so-called undesirable vehicles off the highway.

Let us realize and take for granted the absolute necessity of living up to the reputation of satisfactory performance on the highway, which reputation I feel is desired by the great majority of operators.

Engine Characteristics

The engine as it plods along the highway has certain characteristics as shown in Fig. 3, showing results of laboratory tests of a typical motor, giving horsepower, torque, and fuel-economy curves. On this chart I have placed the manufacturer's recommended governed speed of engine, 2150 r.p.m.

when running light, and 1900 r.p.m. when running loaded, which we will assume will be taken in fourth gear at approximately 30 m.p.h., and have also shown how this same speed can be affected by simply shifting into fifth where operating conditions permit this maneuver. The surprising thing noted is that the ability remains about the same for this newly selected transmission range, while the machine immediately moves over into a much higher torque performance, a much higher, in fact practically the highest, fuel-economy range, and with ample additional horsepower where an additional spurt of speed is required while operating. We wish to emphasize the importance of recommending this most vital point of economical operation, not only from the fuel-economy, but also from the mechanical-maintenance standpoint of motor vehicles, to say nothing of the increased speed available with a few more revolutions per minute of the engine if such is desired, completely avoiding any operation at the so-called hilt of the governor, as any operation designed and carried out at or near the top speed of the governor cannot help but produce disastrous and expensive mechanical results.

State Regulations

Before going into the actual problem of equipment, let us glance for a moment at the restrictions placed upon normal operators by various States. We are all familiar with the work done by General Motors Corp., and now being carried on by the National Highway Users' Conference, in preparing charts to show these State variations, as applied not only to gross vehicle weights but also to axle loads, to say nothing of widths, which fortunately are somewhat standardized, also total overall lengths, and as to whether or not additional units in the nature of four-wheel trailers are permitted behind each of the types. We are interested, however, mostly in that question of gross vehicle weight and axle loadings but, in checking over these laws as to which vehicles have the advantage as far as gross weight allowance is concerned, six-wheel trucks do not seem to have the advantage in any State. In

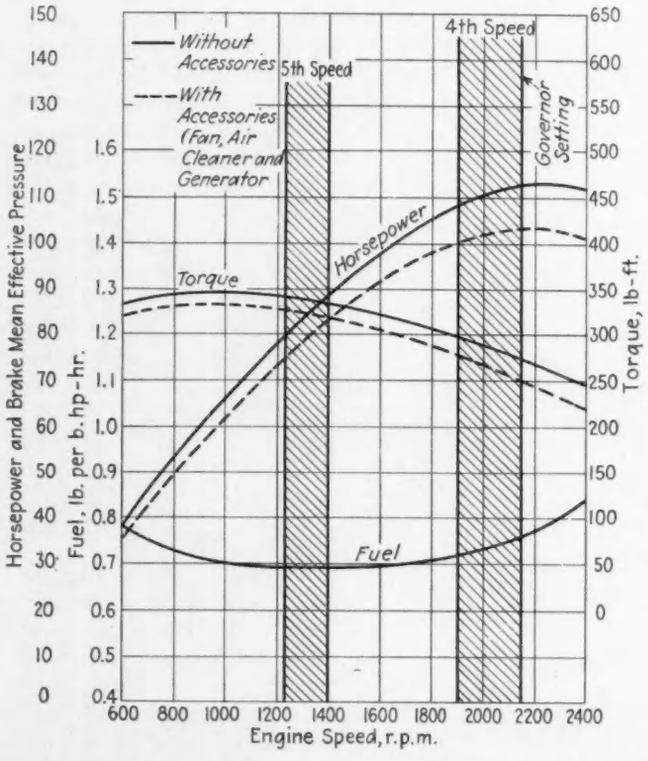


Fig. 3 – Torque, Horsepower and Fuel-Consumption Curves of Typical Engine

17 States it is allowed the same legal limit as the semi-trailer but, in 30 States, the semi-trailer is allowed a legal advantage. These laws cover 46 States and the District of Columbia. In other words, engineers and road experts most generally can see that axle spacing does have something to do with legal limit allowed and, therefore, they create an advantage favorable to the semi-trailer. In practically all States it is possible to get more bulk load capacity on the semi-trailer than on the six-wheel unit from mere set-up of legal limitations. In 5 States the gross load is cut down to such an extent that neither tractor semi-trailer nor six-wheel truck has any advantage over the four-wheel truck. Therefore, from a load-carrying standpoint and from a legal standpoint, the tractor semi-trailer seems to have a decided advantage.

The foregoing can prompt but one thought and that is the absolute necessity of some recognized uniform code to assist the manufacturer in stabilizing, if nothing else, the market of automotive equipment, and thus appreciably simplifying the problem of the operator, especially those in wide and diversified territories.

The well-known differences of opinions among legislatures, of the limitations to be placed on various types of equipment by the various States, need no introduction.

A map of the United States is shown in Fig. 4, giving the main marketing divisions of a theoretical corporation operating on a national basis, and the States sub-dividing each division or zone, presenting a practical problem of a possible economic set-up of operation.

By glancing at the tabulations of Figs. 5 and 6, there is brought forcibly before you the difficulty of interchangeability within various States, all within individual marketing divisions encountered by the average motor vehicle superintendent or operator.

Choice of Type of Transportation

From all the foregoing it is very evident that one must have clear in his own mind the accepted limitations of various types of trucks available on the market. For application to the delivery problem and for reasons to be touched on under each individual heading, we will assume for the problem of further presentation, the following approximate gross vehicle weights:

Four-Wheel Truck	24,000 to 26,000 lb.
Six-Wheel Truck	32,000 to 34,000 lb.
Tractor Semi-Trailer	40,000 lb.

The preceding figures, for machines, are well within the average normal limits insuring satisfactory operation on the highway of those machines as are produced today, each type, we feel, filling its own individual needs of constructive economic effort in its respective range.

The charts to follow will be based primarily on these same assumptions.

Although some States in various sections of the country permit of much larger types, this general average closely approximates the general average shown on Fig. 6.

In passing, we will take for granted that the prospective purchaser has looked carefully into the problem as to ability of his machines to negotiate the highway with the engine which he might select, especially where use of the bogie or trailing axle dividing the load is used, thus reducing the resistance to rim pull of the driving axle. Under any circumstances no matter what the type of transportation, the combined weight of the driving axles when translated to the road with a coefficient of friction of 0.60, which represents that of rubber to concrete, must be in excess by at least 10 to 15 per cent of the rim pull exerted by the machine when put in low gear. Otherwise there will be experienced a spinning effect in the low gears, and the machine will be of no or

little value in its low gears, and thus appreciably hinder the operating efficiency of the unit.

Any motor truck, as a self-contained transport unit, must combine complete loading and carrying facilities with power-plant, driving mechanism, and controls. It must be designed to carry a specified useful load at a certain speed over roads of various kinds, and up grades of various steepness. It must be provided with means whereby it can be guided and controlled accurately and conveniently, and be equipped with brakes to bring it to rest within a reasonable distance. All of this work it must perform with reliability and economy. This specification must all be gaged to give satisfactory performance, long life, and reliability without excessive weight, purchase price, or operating cost. If equipment is overloaded excessively, it will be unable to deliver normal performance. Over-stressed parts will fail and wear out prematurely, and the cost of operation will increase out of proportion to the increased tonnage which it is called upon to carry.

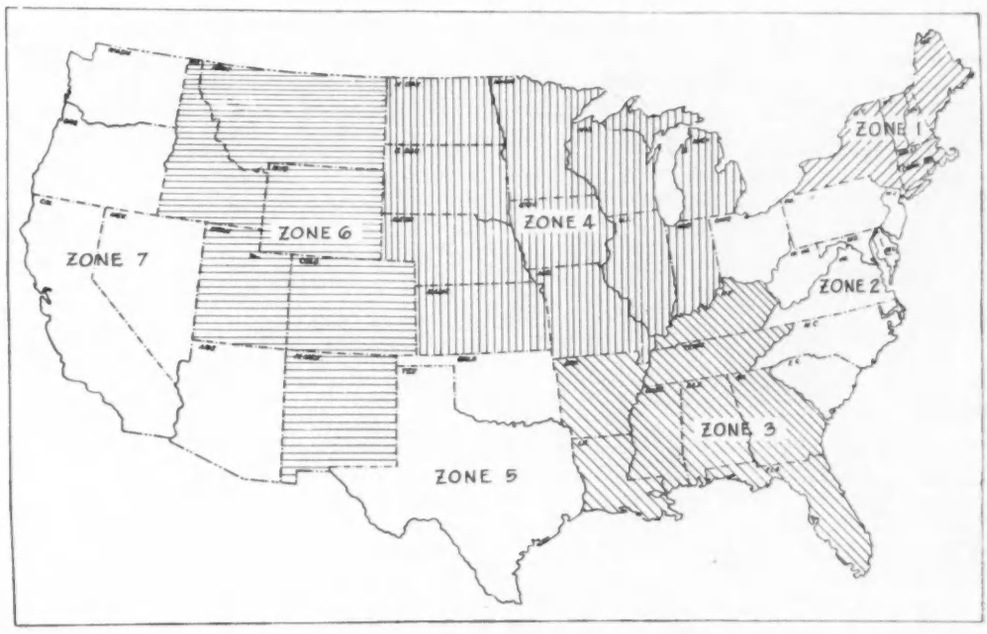


Fig. 4 – Theoretical U. S. Marketing Divisions

A given engine with a proper gear ratio will push or pull so much weight against so much grade resistance and, whether that gross weight is made up of a simple riveted chassis of four or six wheels, or a more complicated one jointed in the middle, will not enable the engine to do more work by any magic of possibly pulling the load instead of pushing it.

Four-Wheel Trucks

The four-wheel truck generally has been recognized as the conventional type of truck for handling all of the normal business over the highway, but there has been created, however, a need for a departure from this conventional machine brought about by the ever-increasing demand for more economy, which can be obtained only by increasing the size and weight of vehicles. It can, therefore, be said that the subsequent development of six-wheel trucks and tractor semi-trailers has been the outgrowth of the creative thinking of manufacturers and users toward accomplishing but one thing, and that is to move cargoes more economically from point to point by means of motor transport. It is obvious therefore that, with the increased demand for larger loads, there must of necessity be increased weights and equipment designed to carry these larger loads.

The four-wheel truck of today, having served its purpose well in light- and light-medium-duty service, and whose characteristics and faithful performance are known and recognized, unfortunately has been compelled to step aside due to its mere basic shortcomings when limited as it is to four wheels.

In the mere matter of tires alone, tire sizes must increase with this ever-increasing demand for service, and they soon reach the point where they are clumsy to handle and very costly; also, it is difficult to keep the machines so equipped within legal limits of widths of State restrictions.

We must also increase the engine size, and increased engine size means that increased strength must be built into the driving train, clutch, transmission, and rear, to say nothing of the increased loading on the front-axle assembly.

The country, it seems, in its search to accomplish these things, apparently first turned to tractor semi-trailers, and subsequently there has been brought into the picture the six-wheel truck, both of which seem to have found their respective economic places in the course of perfection of their mechanical designs.

Six-Wheel Trucks

The six-wheel truck, together with a complete detailed discussion of its construction and operation, has been amply taken care of in a paper presented at the April, 1936, Meeting of the Metropolitan Section of the Society, by Austin M. Wolf.[2] I refer you further to this presentation and make no attempt at further analysis of a most ably prepared and presented treatise.

We have seen in the State limitation charts presented, the apparent wide discrimination against six-wheel units practiced by various legislatures. Doubtless, this discrimination has been due to advantages which the tractor has gained through loose and unscientific wording of some of the laws under which an operator can put a fake six-wheel unit on the highway, such as is common on both the Atlantic and Pacific Coasts.

So flagrant have some of these fakes become that some six-wheel units have been operated with one dummy axle with a cable arrangement of some sort so that, when running light, this axle could be pulled clear of the highway. This procedure would be out of the question if the dummy axle were so constructed to carry a fair part of the load. It is exceedingly difficult to account for most legislation, but perhaps motor legislation is particularly hard to rationalize, which deficiency, however, seems to be correcting itself under our American Trucking Association. There is no question but that the favoritism shown for tractor semi-trailers, to some extent, has been due to extensive lobbying efforts and, in the preceding remarks, we can see readily how some operators have made six-wheel units unpopular with legislatures.

We will rather take up the basic assembly going into such units, whether they are: (1) single-drive, trailing axle; (2)

[2] See S.A.E. TRANSACTIONS, September, 1936, pp. 362-380: "The Construction and Operation of Six-Wheel Trucks," by Austin M. Wolf.

Fig. 5—Gross Weight and Axle Limits

States	GROSS WEIGHT 4-W	6-W	T.S.T.	AXLE LIMITS 4-Wheel F	R	6-Wheel F	I	R	Tractor Semi-T. F	I	R
Zone 1A											
Me.	27,000	36,000	36,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
N.H.	23,000	23,000	35,000	15,000	15,000	15,000	15,000	15,000	15,000	15,000	15,000
Vt.	20,333	22,000	23,666	15,000	15,000	15,000	15,000	15,000	15,000	15,000	15,000
Avg.	23,444	27,000	31,555	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Zone 1B											
Mass.	30,000	40,000	40,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
R.I.	32,000	40,000	40,000	22,400	22,400	22,400	22,400	22,400	22,400	22,400	22,400
Conn.	32,000	40,000	40,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
N.Y.	36,000	44,000	F	22,400	22,400	22,400	22,400	22,400	22,400	22,400	22,400
Avg.	32,500	41,000	40,000	22,400	22,400	22,400	22,400	22,400	22,400	22,400	22,400
Gen.Avg.	27,972	34,000	35,777	19,200	19,200	19,200	19,200	19,200	19,200	19,200	19,200
Zone 2											
Pa.	27,300	37,800	40,950	18,000	18,000	8,000	16,500	16,500	18,000	18,000	18,000
N.J.	30,000	40,000	60,000	Tires	Tires	Tires	Tires	Tires	Tires	Tires	Tires
Del.	25,000	36,000	40,000	18,000	18,000	18,000	16,500	16,500	18,000	18,000	18,000
Md.	26,000	40,000	40,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Ohio	24,000	24,000	42,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
D.C.	30,800	39,600	39,600	24,640	24,640	24,640	15,400	15,400	24,640	24,640	24,640
Va.	24,000	35,000	35,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
W.Va.	F	F	F	Table	Table	Table	Table	Table	Table	Table	Table
N.C.	20,000	40,000	40,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
S.C.	20,000	20,000	20,000	10,000	10,000	10,000	10,000	10,000	10,000	10,000	10,000
Avg.	25,089	34,511	39,728	17,520	17,520	16,091	15,986	15,986	17,520	17,520	17,520

F - Formula

N.R. - No Restriction

double-drive, trailing axle; or (3) double-drive, double-driving axle.

It seems to make little difference which type is selected as to the size of engine which will propel such a machine over the highway. At this point, however, we wish again to call attention to a remark made earlier in this paper, that experience has proved that, for vehicles carrying their load directly on their backs at a reasonably high speed over the highway, there must be provided a sufficient additional amount of power. This added power is, in all probability, necessary in order to overcome some of the increased friction of the drives, higher center of gravity on some designs brought about by limited loading space, fight of bogie in six-wheelers, more wind resistance, no articulated load, or other contributing causes, all seeming to contribute to the phenomenon necessitating an increase in power. We have estimated and found this to be an increase of practically ½ per cent of gross ability factor when attempting 30 m.p.h. road speed or over.

It can, therefore, be calculated that a six-wheel truck with an engine capable of hauling 32,000 to 34,000 lb. over the highway, as a tractor can certainly handle successfully a larger gross vehicle weight, approximating 40,000 lb. Further than this, when a six-wheel truck passes out of the 32,000- to 34,000-lb. class to a higher gross-vehicle-weight rating, it is frequently necessary not only to change to a larger engine and transmission, but also to a larger rear assembly adopting, in the case of a nationally popular double-drive axle, from one to a second series of axles for normal operation on the highway, thus automatically again increasing chassis weight at a very slight increase in payload and, of course, at a higher price, to such an extent that it is considered inadvisable by many operators to take the machines of this description beyond the limits discussed. If, however, this type of truck is to be operated in considerable excess of the weights legally fitting such an operation, the manufacturer must build further into these units considerable increased strength and durability of service, which carry with them again automatically increased chassis weight and price which, if they are to carry the increased vehicle weight with the same engines as are placed in lower-rated machines, it must be done at a sacrifice of proportionate speed.

In ordinary highway operation the six-wheeler can go over roads and into places where it would not be feasible nor possible to take any other type of machine, particularly when that machine is of double-drive type, which type incidentally can be obtained at very little difference in initial investment cost.

The six-wheel units usually are attached to service where heavy loads and not too large body capacities are required. Under this heading we would place dump trucks, coal trucks, heavy-machinery trucks, and the like. Where maximum performance is essential with high-gear or low-gear grade ability on virgin terrain and unimproved roads, the six-wheeler stands out far above any other type of transportation.

In an endeavor to meet further the demand for ever-increasing payloads over the highway, the manufacturers of automotive equipment have turned away from the heavy-duty, heavy-engine machines to the lighter-weight, smaller-engine machines, giving practically the same chassis performance under set-up and have completely withdrawn from the market many of the larger engines of a few years ago.

Let us then leave the six-wheel unit with the thought that, for the average hauling problem when properly and adequately designed, it seems to fit into its niche of transportation, approximating the intermediate step between the operation of the four-wheel truck and the tractor semi-trailer.

Tractor Semi-Trailers

There is, without doubt, a legitimate field for tractor semi-trailers, and it is probable that it has not yet been fully developed, but pressure of legislation caused so many of the competitive fields to be neglected and ignored, that tractor semi-trailer manufacturers have been able to push their sales to a remarkable degree.

Fig. 5 *(Continued)* – Gross Weight and Axle Limits

	GROSS WEIGHT			AXLE LIMITS							
	4-W	6-W	T.S.T.	4-Wheel		6-Wheel			Tractor Semi-T.		
				F	R	F	I	R	F	I	R
Zone 3											
Ga.	22,000	39,600	39,600	17,600	17,600	17,600	17,600	17,600	17,600	17,600	17,600
Fla.	18,000	18,000	34,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Ala.	20,000	20,000	20,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Miss.	22,000	22,000	22,000	12,000	12,000	12,000	12,000	12,000	12,000	12,000	12,000
Ky.	18,000	18,000	18,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Tenn.	18,000	18,000	18,000	18,000	18,000	18,000	9,000	9,000	18,000	18,000	18,000
Ark.	Table	Table	Table	F	F	F	F	F	F	F	F
La.	7000NL	7000NL	10000NL	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Avg.	19,666	22,600	25,266	15,866	15,866	15,866	12,866	12,866	15,866	15,866	15,866
Zone 4A											
Ind.	F	F	F	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Ill.	24,000	40,000	40,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Mich.	Table	Table	Table	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
Wis.	19,500	29,200	35,000	15,500	15,500	15,500	15,500	15,500	15,500	15,500	15,500
Avg.	21,750	34,600	37,500	16,725	16,725	16,725	16,725	16,725	16,725	16,725	16,725
Zone 4B											
N.Dak.	35,000	35,000	35,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
S.Dak.	20,000	24,000	30,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Minn.	N.R.	N.R.	N.R.	18,000	18,000	18,000	12,000	12,000	18,000	18,000	18,000
Iowa	F	F	F	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Mo.	26,000	26,000	40,000	19,200	19,200	19,200	19,200	19,200	19,200	19,200	19,200
Kan.	26,000	34,000	43,250	16,000	17,250	16,000	17,250	17,250	16,000	17,250	17,250
Neb.	32,000	32,000	32,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Avg.	27,800	30,200	36,050	16,742	16,907	16,742	16,064	16,064	16,742	16,907	16,907
Gen.Avg.	24,775	32,400	36,775	16,733	16,816	16,733	16,394	16,394	16,733	16,816	16,816
Zone 5											
Okla.	24,000	24,000	31,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Texas	7000NL	7000NL	7000NL	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Avg.	24,000	24,000	31,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Zone 6											
Mont.	24,000	34,000	40,800	16,800	16,800	13,000	13,000	13,000	16,800	16,800	16,800
Idaho	24,000	50,000	F	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
Wyo.	36,000	48,000	F	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
Utah	F	F	F	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
Colo.	24,000	34,000	F	17,000	17,000	17,000	17,000	17,000	17,000	17,000	17,000
N.Mex.	F	F	F	17,000	17,000	17,000	17,000	17,000	17,000	17,000	17,000
Avg.	28,000	38,666	40,800	17,133	17,133	16,500	16,500	16,500	17,133	17,133	17,133
Zone 7											
Wash.	24,000	34,000	42,500	18,500	18,500	13,000	13,000	13,000	18,500	18,500	18,500
Ore.	54,000	54,000	F	17,000	17,000	17,000	17,000	17,000	17,000	17,000	17,000
Nev.	25,000	38,000	38,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
Cal.	24,000	34,000	34,000	17,000	17,000	17,000	17,000	17,000	17,000	17,000	17,000
Ariz.	22,000	34,000	40,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000	18,000
Avg.	23,750	35,000	38,625	17,625	17,625	16,250	16,250	16,250	17,625	17,625	17,625

F - Formula
N.R. - No Restriction
N.L. - Net Load

There is no question that a very large number of tractor semi-trailer outfits have been purchased for work where a four-wheel truck might even have been used to better advantage and more economically, possibly because laws in many places permit economical hauling only on tractor semi-trailers.

We will admit that the tractor semi-trailer field seems to be occupied more definitely by manufacturers than is the six-wheel field and this condition, therefore, has brought about considerable publicity on this type of transportation.

Basically, however, the machines are fundamentally sound, even to an extent that claims have been made that semi-trailers represent the safest, most economical, and most fundamentally sound in design of all transport vehicles. We will admit of more favorable load distribution from a road-impact standpoint where commodities are hauled, requiring large body which, if mounted on a truck, would cause same to be very long, and which brings the desirability of semi-trailers into the picture. Provision for the articulated joint provides for greater flexibility, shorter wheelbase of power unit, and lower center of gravity, but one must guard definitely against artificial stimulation of the market for oversized semi-trailers to be drawn by undersized tractors, which might be mis-

interpreted by vendors of this type of equipment as being due to misleading advertising.

It might be interesting to have it recorded that it is not necessarily true that it is easier to pull a load than to push it. It is not true that a truck can pull more than it can carry, nor is it true that motor trucks are excessively over-powered and that this power can be used to haul a load for nothing. The operator pays only for power that the engine actually develops, and that engine cannot develop any more power than is absorbed in doing actual useful work.

Each type of transportation, of which the semi-trailer is no exception, must be studied carefully in its real engineering set-up of load application, as well as the real point of application of power to move that load effectively over the highway. This requirement, we feel, is a matter of sane engineering thought in execution of design and manufacturing detail. The thought of excessive power and its application to tractors is not well taken. The less repair-shop floor space necessary and the greater accessibility of the modern tractor are two favorable points, to say nothing of the advances which have been made in recent years in attempting to eliminate the differences in weights of the companion semi-trailers and bodies by building into the structure a combination semi- and body to give satisfactory performance.

One of the main reasons given against this type of equipment is its tendency to "jack knife." On this particular question, a survey recently conducted of approximately 125 fleet operators covering 4000 semi-trailer operations in the country, reported only 1½ per cent as having experienced any difficulty with "jack knifing," whereas 60 per cent expressed the opinion that they were entirely free of this complaint, the remainder having little or no complaint to offer on this score.

Inter-terminal or shuttle service is one point of operation where semi-trailers apparently stand unchallenged, that is, using a few tractors for many semi-trailers where it is possible to get far better performance cost where terminal time, that is, loading and unloading of units, is of considerable importance.

Another interesting phase of motor transportation is the so-called ferry or bulk-haul service with long trains hauling cargoes between terminal points performing interplant or inter-warehouse service.

These last two phases of the transportation problem, however, are confined strictly to a single kind of transportation, being merely mentioned in passing but not considered within the scope of this presentation.

Going back to one of our original statements, that the larger the unit the less the unit cost, it is interesting to note that a large machine actively engaged in the work normally performed by a machine of one-half the larger machine's capacity while operating to full capacity, will return the same unit cost per ton-mile although kept only within approximately 65 per cent of its own, that is, the larger machine's capacity, leaving the balance or 35 per cent capacity for picking up additional business at lower unit cost.

The general conclusion is, therefore, that this type of equipment has found its way definitely into transport problems of today, and can be reckoned with to continue to maintain its place as a machine much sought after and economical when placed in service.

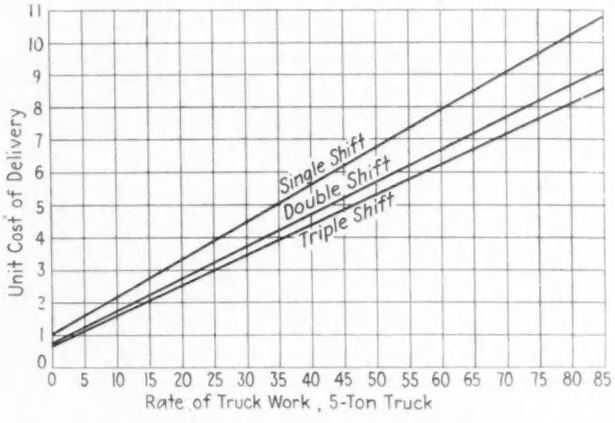


Fig. 7 – Chauffeur Shifts Vs. Relative Delivery Cost

Chauffeur Rotation

The much-discussed, and probably little understood, problem of rotation of chauffeurs is based on the fact that, although it is possible to double the output of a vehicle by simply operating it twice the number of hours, it also follows

Fig. 6 – Summary of Average Weights and Axle Laws

	WEIGHT LIMITS			AXLE LIMITS							
	4-W	6-W	T.S.T.	4-Wheel		6-Wheel			Tractor Semi-T.		
				F	R	F	I	R	F	I	R
Zone 1-A	23,444	27,000	31,555	16,000	16,000	16,000	16,000	16,000	16,000	16,000	16,000
" 1-B	32,500	41,000	40,000	22,400	22,400	22,400	22,400	22,400	22,400	22,400	22,400
" 1	27,972	34,000	35,777	19,200	19,200	19,200	19,200	19,200	19,200	19,200	19,200
" 2	25,089	34,511	39,728	17,520	17,520	16,091	15,986	15,986	17,520	17,520	17,520
" 3	19,666	22,600	25,266	15,866	15,866	15,866	12,866	12,866	15,866	15,866	15,866
" 4-A	21,750	34,600	37,500	16,725	16,725	16,725	16,725	16,725	16,725	16,725	16,725
" 4-B	27,800	30,200	36,050	16,742	16,907	16,742	16,064	16,064	16,742	16,907	16,907
" 4	24,775	32,400	36,775	16,733	16,816	16,733	16,394	16,394	16,733	16,816	16,816
" 5	24,000	24,000	31,000	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.
" 6	28,000	38,666	40,800	17,133	17,133	16,500	16,500	16,500	17,133	17,133	17,133
" 7	23,750	35,000	38,625	17,625	17,625	16,250	16,250	16,250	17,625	17,625	17,625
Gen.Avg.	24,750	31,597	35,424	17,346	17,360	16,773	16,199	16,199	17,346	17,360	17,360

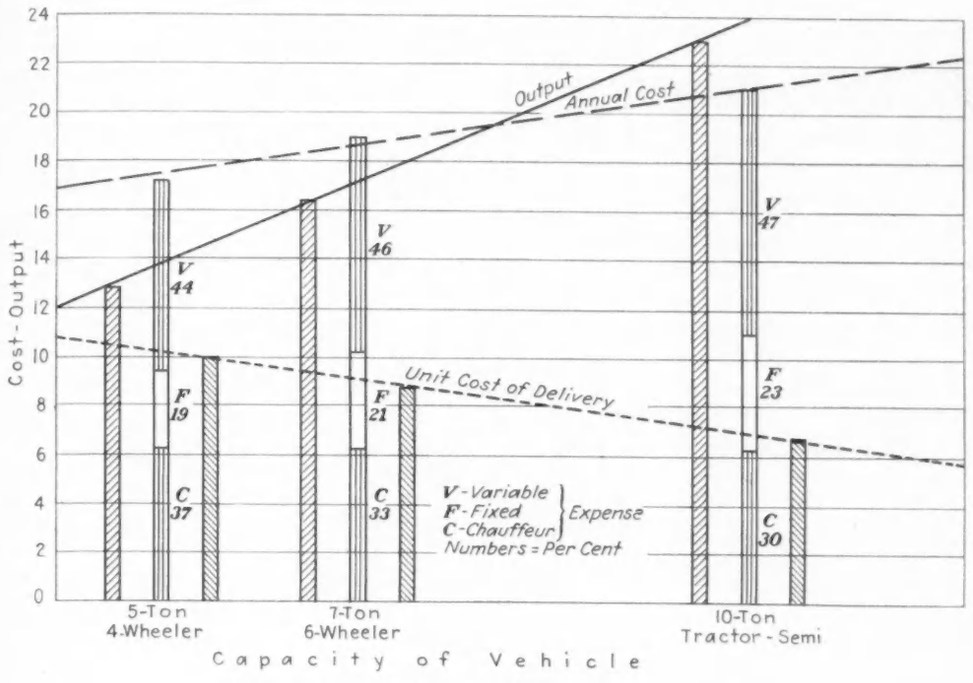


Fig. 8 – Chart of Motor Vehicle Cost-Performance

that cost of delivery will diminish appreciably. In order to bring this point clearly before you we have indicated on Fig. 7 the relative unit operating expense of a medium-size vehicle with single, double, and triple shifts of chauffeurs, at varying lengths of haul. This phase of specialized transportation needs no further comment.

Operating Costs

In order to present more clearly and in graph form the relative costs obtained from operating the various types of equipment under discussion, Fig. 8 shows the relative performance of these types. These figures are based on average operating costs as furnished by several manufacturers and operators and, although the vertical scale has been distorted purposely by multiplying by a constant, it will show any operator now using any one of the types of transportation equipment already covered, the relative output, annual cost of operating, and unit cost of commodity delivered, based on the application of each vehicle competing in an equal distribution of territorial assignment.

The layman operator can use these figures as a fair guide of relative merits of machines within the scope mentioned, and it is to be noted that there is relatively little difference in annual operating costs of the various vehicles; yet the amount of work performed is considerably in favor of the larger machines with a substantial saving in unit cost of product delivered.

Fig. 8 was obtained from the average questionnaire answers as to engine sizes and actual chassis weights of various types of transportation as follows:

Number of Wheels	Type	Gross Vehicle Weight, lb.	Recommended Engine Size, cu. in.	Total Chassis Weight, lb.
4	Truck	25,000	395	9,056
4	Truck	30,000	451	10,486
6	Truck – (Single-Drive)	32,000	451	12,010
6	Truck – (Double-Drive)	32,000	451	11,860
6	Tractor and Semi-Trailer	32,000	421	12,900
6	Truck – (Single-Drive)	40,000	495	12,950
6	Truck – (Double-Drive)	40,000	495	13,930
6	Tractor and Semi-Trailer	40,000	456	14,400

Note: Tractor and semi-trailer includes semi-trailer weights.

Conclusion

In conclusion, although little or nothing has been said relative to operation of motor transportation, it is to be recommended that any operator, before attempting to purchase for operation any additional equipment within a fleet, thoroughly study the requirements of that fleet and assign those present vehicles now in service where they will do the utmost good, placing the largest and speediest machines on the furthermost points of coverage, thereby placing the smaller, slower machines close to warehouse points.

Do not load up your fleet too much with small pieces of equipment where the load is heavy and, consequently, suffer from uneconomical operation. Place equipment within the fleet always according to a definite plan of operation of the fleet as a whole – not divided according to distribution within various regions nor minor subdivisions or districts. After purchase of new equipment, see to it that plans so carefully laid by surveys preceding the purchase, are carried out in detail, and there can be but one result, and that most gratifying to the management, allowing each machine, either in present operation or to be purchased, to find its own particular economical niche of service.

Finally, in presenting this paper we hope in a small way to have opened the minds of those interested to further interesting subjects, to guide them in bringing about a more thorough understanding of "engineered transportation."

Discussion

Discussion from the Trailer Builder's Viewpoint

– W. G. Retzlaff
Fruehauf Trailer Co.

WE, in the trailer field, must also be very neutral in our viewpoint for you truck engineers represent our best customers; for every year your branches and dealers sell millions of dollars worth of trailers for us, so we remember the old diplomatic slogan: "The customer is always right."

Again, our company manufactures and sells six-wheel attachments as well as trailers. Incidentally, we inherited this six-wheel attachment business on the West Coast about three years ago and it has been paying dividends ever since, so obviously we must be neutral.

Who brought up this question of six-wheelers *versus* semi-trailers anyway? We agree 100 per cent with Mr. Moxey when he says: "there does not seem to be any *versus side* to the problem." This *versus* may have been a burning question 5 or 10 years ago, but today enough facts and figures are available obtained through old man experience to guide us definitely in our future recommendations and, naturally, local conditions will be the determining factor.

We question the statement: "that the favoritism shown for tractor semi-trailers to some extent has been due to extensive lobbying efforts," and that: "pressure of legislation caused so many of the competitive fields to be neglected and ignored that tractor semi-trailer manufacturers have been able to push their sales to a remarkable degree." When we

consider the small trailer organizations compared with the large truck organizations, the last statement is really a compliment.

It is a case of about 200 truck salesmen to every trailer salesman. The Fruehauf Trailer Co. has confined its lobbying to joining and helping truck associations whose members, incidentally, buy both trucks and trailers and, if these operators see fit to help pass laws in favor of semi-trailers generally over the dead bodies of highway engineers, it means that some people are really sold on the trailer idea.

Just a word about this 4 per cent ability factor at 20 m.p.h. propaganda. I know that, if John Law steps in and forces me to buy a 150-hp. automobile able to climb a 5 per cent grade at 60 m.p.h., especially for use in a fairly level country, I am going back to using railroads. Old Man Economy will force me to do this. By the same token, if you try to legislate a 4 per cent ability factor at 20 m.p.h. for trucks, six-wheelers and tractor-trailers, you are going to force a lot of truck and trailer users out of business because they cannot all buy 400 cu. in. engines and compete against other low-cost types of hauling.

Many of our 14,000 customers will be affected by such a ruling and naturally, when they find out who sponsored such a law and when the inevitable reaction takes place, it will mean embarrassing moments for your sales department and the engineers who indorsed it.

Imagine trucking conditions in Kentucky with an 18,000-lb. gross maximum law – a larger truck or tractor means more weight, less payload and higher costs and, eventually, less trucking.

If an evil actually exists, let us first dig up the facts then, basing our recommendations on facts not opinions, let us make a complete study in an attempt to localize our final recommendations instead of attempting to saddle new rules on our industry on a National scale.

In other words, if a certain highway between New York City and Philadelphia is overcrowded or the *facts* show that accidents are caused by slow-moving but otherwise safe equipment, possibly the answer is a wider road or a new parallel road instead of trying to curb truck transportation.

Let us borrow a page out of the railroad engineer's book of over 50 years of experience. Does he try to move all freight at express speed? On the contrary, today we have express, fast-freight, and slow-freight service. Does the railroad engineer use mountain-type locomotives on one grand National scale? No, it is not necessary and it is too expensive. Does he use a single- or double-track system Nationally? No, he studies his freight movements and, in congested areas, we find wide road beds with many tracks.

Reference is made to the favorite slogan of the trailer manufacturers that *"a truck can pull more than it can carry."* Even as in the automobile and truck advertising we take a little poetic license in this phrase. We do not say that every truck can pull more than it can carry. We do not say that this condition is possible at the same speed. We do not claim that it will do so on a Pike's Peak climb. Carried out to seven decimal places the trailer manufacturer actually means that "a standard truck *generally* can pull more than it is *designed to carry*."

A standard truck generally *can* pull more than it is designed to carry because thousands of practical installations prove it. Almost 60 per cent of our trailer volume is repeat business which means that a practical truck operator first experimented on a small scale and, after checking figures, he believes it good business to buy more tractors and trailers.

Theoretically, a special truck cannot pull more than it is designed to carry because a truck engineer can so design a vehicle as to give the same performance ability as a tractor and trailer but, with the heavy axles, frame, and springs permitting it to carry all of the payload on its back – that is if he could ignore State legislation and if he could ignore the dollar and cents factor. However, from a practical standpoint, the law will not permit the same equivalent payload on one weight-carrying axle as on two axles and, during the spring season, capacities would be reduced further because two axles permit oversized tires designed for spring use.

Performance is important and, in case of a tractor-trailer combination with larger payloads, the operator sacrifices some speed and performance for more payload and increased income which means more profit.

Summing up, a truck engineer designs a standard 1½-ton rated truck knowing that most operators will load 3 or 4 tons on the truck's back which possibility is considered in the original design and protected against in his factor of safety. However, if 6 to 10 tons are loaded on the truck's back, generally failure results whereas, if the truck is used as a tractor with a weight distribution of only 45 per cent on the tractor's back it means that almost 9 tons can be hauled without exceeding the 4-ton limit on the truck. Therefore, as a truck 4 tons is the safe limit whereas, as a tractor, 9 tons is possible.

Finally, if there is any doubt in a truck engineer's mind as to the practical application of trailers, we will gladly loan you a trailer to make tests and should the S.A.E. desire to experiment or collect facts, we stand ready to help. Remember, we are but a small infant compared with the truck manufacturers. Figures show 611,644 trucks sold during 1936 compared with approximately 25,000 commercial trailers sold according to U. S. statistics, or about 4 per cent.

As a whole, trailers increase the economic operating range of trucks, handle special commodities, and actually provide more outlets for trucks. We try to help increase truck transportation. We are concentrating on the old obsolete trucks still in use, trying to convert them into safer more modern equipment, so we should be regarded as an ally with only a limited amount of sharp-shooting permitted.

Contends Six-Wheeler Penalized Too Much

– Austin M. Wolf
Consulting Engineer

MR. MOXEY'S statement that there seems to be no controversy regarding one type of equipment over the other but that each one fits into a distinct niche, I am pleased to note. However, I believe the discussion of his paper will prove that this ideal is not accepted as generally as many of us would like it to be.

The maximum torque rating of an engine should also mention the r.p.m. at which it is developed and which would be a further contribution if this figure were added to the *Commercial Car Journal* tables. It is furthermore important to know whether the torque curve is relatively flat or has a decided droop after reaching the maximum. No doubt the differences in ratings in the *Commercial Car Journal* is due to the truck manufacturer selecting a particular speed for the torque output which is not the same as that used by his competitor using the same engine. Furthermore, there is no standardization in giving the output figure as to whether accessories are included or not. If the figure is to be of value, it should include accessories exactly as installed in the particular truck, for what the bare engine does on the dynamometer is of no practical use. This same remark applies to horsepower output. The shape of the torque curve is of considerable importance as is shown in Mr. Moxey's Fig. 3 where better engine output utilization and better economy become possible with the use of a more flexible transmission. The personal equation comes in here too, for the success of an operation is often tied up with the driver's ability and performance.

With reference to the remark on the semi-trailer having a legal advantage in State regulations, Mr. Moxey lays this advantage to the axle spacing. I believe that the Bureau of Public Roads tests conclusively proved that spacing of the six-wheeler bogie unit axles is sufficiently great to prevent undue stresses in the roadbed and that any additional spacing between axles is of no help.

The weight classification given, while based on the summation of State laws, seems to indicate that the respective gross weights can be handled best by the three types of truck units. This point is confirmed further by the tabulation on the last page. I cannot subscribe to this conclusion based on the subject matter in Mr. Moxey's paper unless we assume many equipment and structural restrictions to which I believe no truck manufacturer would acquiesce.

I am heartily in favor of road gradient surveys and subscribe to the statement that traction is limited to the rim pull of the driving wheels, whatever their number. It is well and good to make such surveys and figure the coefficient of friction as 0.6 under ideal conditions but, from now on until the spring of the year, at least where snow and ice are likely to change the whole picture, these calculations go overboard. I venture to say that, if we have a winter such as we had two years ago and that we were discussing this problem after the experiences of such a season, our optimistic calculations would be quite changed. We cannot gamble on winter conditions if the trucks are going to pull through and that statement concerns everything starting with the tires and the efficient delivery of power to enough wheels to propel the vehicle effectively.

Mr. Moxey mentions that "jack-knifing" is of little consequence in view of a recently conducted survey of approximately 125 fleets, but again I would like to have this survey made in the spring of the year. From now on, count the number of units that you will see off the road on their side or back.

Mr. Moxey estimates the increase of practically ½ per cent gross ability factor at 30 m.p.h. or over for the truck carrying the load on its back. I would like to have some data on how this figure was obtained. The "additional friction of the drives," I believe, is a fallacy. We must remember that the loads on the bearings and gear teeth, for instance, are divided in a multiple drive and, if each drive and axle is doing its share, we are not adding friction because of the number of parts. I will admit that, in the laboratory, it would be possible to find a slight amount of increased friction but it would be of little consequence. Naturally there are different types of drive, and I am basing my premise on one that is designed properly.

The higher center of gravity attributed to a six-wheeler can happen only in a case where the wheelbase is not of the proper length and the loading volume is obtained by reaching skyward. This arrangement is not proper engineering. Furthermore, I see no reason why the body flooring should be any higher than that of a semi-trailer with its fifth wheel and drop frame as presently used. The "fight" of the bogie unit depends on design; the use of an inter-axle controlled differential will obviate this kind of trouble. Wind resistance should be no greater with the six-wheeler unless bodies are unduly high, which design, as mentioned before, has no reason for existence when wheelbases are selected properly. If the six-wheeler manufacturer wanted to streamline his cab and body with wind resistance in mind, there is no doubt that he could do a better job than is done on the detached cab and body on the tractor-semi-trailer. The articulated load in a tractor-semi-trailer is, in my opinion, an energy-absorber because you cannot agitate the load without contributing energy for this purpose.

Fleet History - A Maintenance Guide

By W. J. Cumming
Surface Transportation Corp.

BEFORE describing our maintenance scheme, it seems necessary to call attention to the fact that the procedures involved have been worked out over a period of years, and have been selected because they have proved satisfactory for our specific operating conditions. The time elements involved or the processes described may, or may not, apply to operations other than our own, for maintenance requirements can be designed only after operating experiences have been analyzed intelligently.

We have a special method for the assembly of these operating experiences which really constitutes a daily log of each bus, and these mileposts of operation guide us from year to year in the proper selection of inspection, unit replacement, and repair periods.

It is interesting to note the items that provide the maximum number of interruptions to our schedules. It should be understood at this point that our daily log charges any interruption to continuous scheduled operation against the bus.

Because of the location of our bus routes and the type of equipment operated, tires consistently have had high score. This item, over the past six years, has ranged from 40 per cent of the grand total, to as low as 10 per cent.

Fare-collection devices are the next greatest cause of delay, and this item ranges from 11.4 per cent down to less than 5 per cent. Gas lines are the next item, ranging from 5.1 per cent down to 2.7 per cent. Springs, transmissions, and air compressors represent about 3 per cent of the total. Clutches, brakes, batteries, and air lines contribute about 2.5 per cent. Differentials, fans and belts, carbureters, accelerators, and starters contribute approximately 2 per cent.

Doors, engine valves, connecting-rod bearings, distributor units, and ignition coils and condensers, add 1½ per cent.

Axle shafts, propeller shafts, and water pumps add approximately 1 per cent, and all other 79 items represent less than 1 per cent of the total.

I have elected first to describe our inspection procedure regarding engines for, in our fleet, the engine, because of its complex design, offers more failure possibilities than any other mechanical unit.

Inspection periods for engines, disregarding fueling, oiling, and external cleaning, occur at 900 to 1000 miles, at 2500 to 3000 miles, and 20,000- to 30,000-mile intervals. The 10,000-mile spread in the last item covers six different engines.

At the 900- to 1000-mile inspection period all lubrication instructions for each type of engine are followed minutely, including draining and refilling the crankcase when due and changing or cleaning oil-filtering elements when called for.

At each 2500- to 3000-mile inspection all external units or parts are checked and, if necessary, they are adjusted or replaced.

At our 20,000- to 30,000- mile inspection, which we call our "long" inspection, all interior parts are checked and cleaned, cylinder-bore dimensions are recorded, piston-rings are replaced, and valves are ground, touched up, or changed. Some slight variations in the internal inspection procedure occur at regular intervals, namely, when the cylinder-block wear limit is reached, at which time the block is replaced by a resleeved block, and when the main-bearing and crankshaft replacement period arrives.

Since we find more disagreement with our scheme over the internal or "long" inspection periods than with any other item, I will recount the reasons for them.

Mass transportation, such as we encounter with its frequent stops, reduces our average speed on many lines to as low as 7 m.p.h., but increases our accelerating periods proportionately and this operation, combined with the extremely hilly nature of the territory over which we operate, obliges us to maintain extremely high engine revolutions and extended periods of low and intermediate gear work. This condition results in abnormally high wear factors in cylinder bores, rings, wrist-pins, and bushings, and main and connecting-rod bearings.

In fact, the condition becomes so serious with some engines that we would much prefer to allot our engines for inspection on the basis of thousands of feet of piston travel, rather than on the basis of vehicle miles. Several attempts, however, to count engine revolutions have failed because of weak devices.

Excessive wear in all engine parts has led us into some very interesting research work regarding these items, particularly that of cylinder wear. Our studies of cylinder wear were related so closely to preventive maintenance that they resulted in very definitely prolonging our engine life. The selection of iron alloys, during the past few years, with increasingly superior wear characteristics has provided us with improved dry sleeves which, in one aggravated case, represented five times greater engine life.

Our method of recording cylinder-bore dimensions at every 20,000- to 30,000-mile interval is relatively simple, but the results obtained have repaid us enormously. On a sheet containing only enough information to identify the cylinder-block or blocks being measured are twelve circles representing the top and bottom of ring travel in each bore, and in these circles are recorded their accurate dimensions. These sheets are filed and, when occasion arises, charts are prepared for study, showing the development of wear in the blocks involved.

It was this type of chart, showing wear factors for various alloys of iron in a variety of engines, that prolonged engine life far beyond our expectations.

These same charts made it quite apparent that modern bus cooling systems designed for sustained engine speeds are of such dimensions and efficiency that slow-speed operations such as ours could not hope to maintain sufficiently high water temperatures to combat excessive condensation successfully, without resorting to mechanical means. Water jacket temperatures of 160 deg. fahr. or less without adequate crankcase ventilation and modern heat-control devices, invite emulsification of all types of oil. We were able, with the help of thermostatically controlled shutter fronts and cotton-bag filters, to reduce emulsification to a minimum.

[This paper was presented at the Metropolitan Section-Regional Transportation and Maintenance Meeting of the Society, Newark, N. J., Nov. 9, 1937.]

Engine valves have required much of our thought in the past but, in recent years, they have caused us exceedingly little trouble. We feel that this improvement is due to having arrived at the very definite conclusion that most valve trouble originates at the valve face. All valves eventually become hard-faced with Stellite on our property. We are particularly careful in fitting valves, using the most modern methods for valve-seat and face preparation. We prefer the old cast-iron seat to the hard-faced seat, although we have many of the latter type in operation. In fact, we run hard-faced valves on hard-faced seats in many of our engines.

We prefer cast-iron seats because we can keep them narrow, never exceeding 1/16 to 3/32 in. With hard-faced valves dimensioned to stand high on these narrow seats we are assured of unobstructed valve openings and cool valves. Many of our engines have operated under these conditions for years without a single case of "dropped valve seat."

We have experimented very successfully with Stellite wrist-pin bushings and nitrided-chrome-vanadium wrist-pins and, although they gave us exceptional wear, we have not adopted these very long-lived materials because of the additional expenditure necessary for the special materials, and the fact that our scheme called for opening up engines at comparatively short intervals. Anyone suffering from abnormal wrist-pin wear should look into these materials.

Piston-rings, a very important item, are changed every "long" inspection for we feel that ring wear very definitely is tied up with oil economy. We also prefer to operate some engines up to the 0.025 in. wear mark before reboring, and this practice requires more frequent ring changes.

We favor specially prepared connecting-rod bearings and main bearing caps which are poured and not spun, with the tinning carefully spread in V threads to give added adhesion.

Our method of prolonging crankshaft life, by spraying high-carbon steel on the bearing areas thereby continuously retaining standard bearing dimensions, is interesting.

The procedure in this scheme is simple and the results are effective, providing the shaft is prepared correctly. We first grind the worn shaft to dimensions that will give us a sprayed bushing of sufficient thickness to withstand the pressures involved, usually not less than 0.060 in., finished.

The ground area is then "shot-blasted" and as many gashes or nicks are cut as are necessary to anchor securely the high-carbon steel bushing to the shaft. The steel is sprayed on the shaft in a ductile state and flows evenly and smoothly into the prepared keyways.

Practically all of our older shafts have sprayed bearings, some of them in operation well over 100,000 miles.

Many of our water-pump housings that are badly corroded are sprayed with a coating of zinc.

Transmissions, although far less complicated than engines, are the weak link in many of our bus designs. This condition is brought about, generally, because of narrow gear faces and resultant high tooth pressures.

Rigid inspection and the careful selection of lubricants partially solve this problem, but it is so critical in some cases that we have been forced to experiment with carefully selected steels and heat-treatments before satisfactory mileages that even approximately fitted our schedules could be obtained.

We would suggest that one of our rules regarding transmission inspection is an important one that is often overlooked. It contemplates readjustment of the main shaft and countershaft bearings shortly after the units are first put into operation. It assures us that bearing cups or races that may not have been seated thoroughly in the assembly line, are "home," and that the play in the shafts has been removed. It has saved us many bearing replacements.

Because of the conditions stated, we have found it practically impossible on certain types of equipment to anticipate transmission failure. In spite of this fact we are certain at specified intervals that gear-tooth fracture or fatigue has not occurred.

Our front- and rear-axle inspections occur regularly at the 2500- to 3000-mile period. Lubrication, of course, takes place at the regular 1000-mile period. Special differential lubricants we have avoided, particularly because it is practically impossible in a large fleet to assign special lubricants to special types of buses without encountering endless mistakes and confusion. The front axle, which is so important from a safety angle, has required more of our time and thought than any other unit of the vehicle.

As a preventive measure against failure in highly stressed units, such as drag links, Pittman arms, steering knuckles, in fact, any parts of a ferrous nature, we use the magnaflux method of crack detection. These checks occur at our 2500- to 3000-mile inspections and at our "long" inspections, and have been in use since 1932. They are so important to a scheme such as ours that further explanation is necessary.

For years previous to 1932 a startling increase had been indicated in cases of fatigue failure throughout our fleet. In part, the increases occurred in many of our newer vehicles and were, no doubt, due to their increased power and speed over our older equipment. Failure occurred in many varying parts hitherto unaffected by this dangerous condition and it finally became, not only a serious economical problem, but it threatened our safety standards as well. A long list of mechanical units and parts were affected and it indicated, upon examination, that further increases might well be expected unless a preventive measure was quickly found that could identify fatigue in its initial stage and thereby make it possible to eliminate a recurrence of the failures, or to establish the necessity for absolute and immediate discard. This seemed a problem unlikely to be solved for quite some period of time, but finally a method was found for the periodic inspection of all parts of a ferrous nature that are likely to be affected.

This positive inspection, once under way, convinced us of the importance of large fillets and well-rounded corners, and the absolute necessity for more strict attention to good design on the part of the manufacturer, for the majority of our failures were design or surface-condition failures and not material failures.

The serious consideration of many of our failures led us to the establishment of standard salvage practices for hundreds of parts that formerly would have failed in service. By these salvaging practices such faults as sharp corners are eliminated from steering knuckles, manufacturers' part numbers are removed when found at critical points, more spacious fillets are ground in crankshafts, connecting-rod forgings are draw-filed to smooth out tool marks on critical surfaces, in fact any previous serious cause of failure overlooked by the manufacturer is rectified before application or re-application.

This inspection procedure is very simple and rapid. The part to be inspected is cleaned, magnetized, and dusted with magnaflux powder. When the piece is tapped, all the powder falls off unless there is a flaw, in which case the defect is outlined with a ridge of white powder that is plainly visible.

In conclusion, there is one item in all maintenance schemes that requires the vehicle manufacturers' help, and this help is becoming more and more necessary every year. I have in mind the necessity for special tools to service modern bus equipment.

We are quite sure that all automotive equipment operators will agree that these tools are much more essential to the successful maintenance of his equipment than are jacks, common tool packs, extra rims, and other materials usually included in specification lists.

Do We Understand the Grinding Process?

By Roland V. Hutchinson
Olds Motor Works

A PARTLY descriptive, partly analytical discussion of facts and opinions concerning the fundamentals of occurrences in the zone of metal removal in conventional surface- and cylindrical-grinding operations is set forth in this paper.

The paper considers in sequence the basic geometrical interrelationships involved; it discusses broadly the contributing causes of heat production in both work and wheel; it comments on effects of dressing practice and kinds and uses of coolants; and finally it illustrates what is attempted in the selection of a wheel for a particular job.

Cam-profile grinding in some of its phases is touched upon as one familiar example of a peculiarly repetitive process cycle occurring while the grinding wheel progressively is changing remarkably in cutting characteristics.

And it was all started by the innocent query: "Do We Understand the Grinding Process?"

SOME months ago W. H. McCoy put the title question to the writer, who answered: "Partly yes and partly no." At his urging, this presentation has been prepared, in the endeavor to record the results of a series of mental gymnastics indulged in while considering this familiar manufacturing process.

We have no quantitative values for many of the items needed in a rigid analysis of the general, or even of a particular, problem in grinding. Further, did they exist, their application to an inherently transient case, as is *any* grinding problem, obviously would involve reasonable mathematical complexity.

So this paper recites a few general premises and considers qualitatively what the writer, by observation and a little measurement, has come to believe *approximately* happens. It is not, and cannot be, meticulously accurate in all its phases.

[This paper was presented at the National Production Meeting of the Society, Flint, Mich., Dec. 9, 1937.]

The Grinding Wheel

The grinding wheel is an agglomeration of abrasive particles of controllable shapes, sizes, and physical properties, intimately mixed with some bonding material, the mix molded under more or less pressure, and the whole heat-treated.

After heat-treatment the resulting wheel is a composite of grains whose orientation is that consequent to their own interlocking tendencies in the presence of the bond under the forces to which they have been subjected. The grains are inter-supported by the bonding material, and the whole structure may be more or less porous. The bonding material may, depending upon its own properties in contact with the abrasive, wet the grains during heat-treatment and cover each one with a very thin film of itself.

By choice of material and processing, the abrasive grains may be hard and brittle or somewhat softer and tougher. The shapes of the abrasive may be elongated or more nearly cubic, with the edges sharp and scratchy or somewhat rounded.

Variations in bond composition and/or heat-treatment react on bond characteristics, making it hard and brittle, or tough, and controlling its mechanical strength.

By varying the proportion of bond to abrasive grain in a given volume of mixture, the *average* spacing of the individual grains may be varied within limits. By varying bond ingredients the consequent bond structure is, in turn, variable, and the foregoing variations are under a reasonable degree of control in grinding-wheel manufacture.

Taking extremes, a vitrified bond wheel has a bond of some kind of glass, sometimes water-white in color – and on the other end of the scale is a rubber wheel. The glass bond pretty well covers the abrasive grains by wetting them during heat-treatment, but it is rather unlikely that the rubber bond vulcanizes tightly to the grains when rubber is used.

Following inspection for grade, soundness, and balance the wheel may be used.

It must be recognized that, although the grains are arranged systematically within the wheel itself, it does not follow necessarily that each individual grain is so placed as to be best positioned for good cutting action when its turn comes to be used. In other words, the grinding wheel in this respect is by no means as uniform as is an inserted-tooth milling cutter.

The Materials Ground

The materials ground may be of almost any solid kind. Confining ourselves to metals, these have varying degrees of hardness, ductility, strength, elasticity, heat conductivity, and coefficient of expansion.

They may or may not be homogeneous, the sizes and prop-

erties of the grains composing them may be unlike and, in general, the material may be received for grinding in *any* condition of internal stress.

Accidental variation of any of these characteristics from those generally expected or encountered is sufficient to upset a nicety of balance of operating conditions for, when obtaining the *absolute* maximum result per dollar in grinding, conditions are somewhat like those of walking a tight rope. Plenty of minor influences are waiting in ambush to push one off, as it were.

Geometrical Considerations

Since wheel and work are both in motion in the customary machine-grinding processes, the motions of considered cutting points of the wheel, through and with respect to the work, are not simple circular motions.

Neglecting for the moment surface-grinding with a cup-wheel face and confining our attention to the use of the periphery of cylindrical wheels, we may as we choose cut "up," or "climb-cut," to borrow terms from milling practice.

The ordinary surface grinder, with the work reciprocating past the wheel, indulges alternately in both habits.

The length of cutting path of the individual grain as it penetrates the work is *not* the same for both directions of traverse, the shapes of the path are unlike, and the order in which things occur is to some extent reversed.

In cutting "up," a curved wedge-shaped chip is removed, *ideally* starting at the thin end of the wedge and increasing in thickness almost to the breaking-out place, when it rapidly diminishes in thickness.

In climb-cutting, a *different* wedge-shaped chip is removed, starting at its big end and building up rapidly to maximum thickness and then proceeding toward the small end.

Conformity of Surfaces

Before continuing discussion, a few words on conformity of surfaces may not be amiss. The usual convention in analysis is to define "curvature" as the reciprocal of radius of curvature and, when applied to the surface of a body, to consider curvature positive when the center of curvature of the surface lies *under* that surface.

Thus a 4-in. diameter shaft has a transverse curvature of $+\frac{1}{2}$, and a 4-in. diameter hole surface has a transverse curvature of $-\frac{1}{2}$, because its center of curvature lies *above* or outside of the material of which the hole surface is the bounding surface.

Imagine such shaft fitted in the hole, and the surfaces of both shaft and hole are in contact all over and conform to each other infinitely.

If r_1 be the radius of curvature of one body and r_2 that of the second, and K is the "conformity factor," we have:

$$K = \frac{r_1 r_2}{r_1 + r_2} \tag{1}$$

and, numerically, with due regard to signs

$$K = \frac{2 \times (-2)}{2 + (-2)} = \infty$$

That is, the conformity is infinite and the fit, theoretically at least, is perfect.

Conformity between a 12-in. grinding wheel and the plane surface it has surface-ground is better understood as $K = \frac{1}{1/r_1 + 1/r_2}$, a variant of the preceding. Inserting numerical values: $K = \frac{1}{1/6 + 1/\infty} = 6$

That is, for surface grinding, the conformity factor is the wheel radius.

Calculation of Length of Interference Path and Average Thickness of Chip Per Grain

Let R = wheel radius, in.
S = surface speed of wheel, ft. per min.
T_s = traverse speed of table (surfacing), ft. per min.
d = radial depth of cut, in.
N_γ = wheel speed, r.p.m.
r = work size radius, in.
N_a = work speed, r.p.m.
L = length of interference path, in.
t = traverse per work revolution (cylindrical), in.
τ = average thickness of chip per grain, in.

Subscripts are $_s$ for surface grinding, $_{ep}$ for external plunge cut and $_{et}$ for external with traverse.

Then, approximately:

$$L_s = \left(1 \pm \frac{T_s}{S}\right)\sqrt{2Rd} \tag{2}$$

$$L_{ep} = \left(R \pm \frac{rN_a}{N_\gamma}\right)\sqrt{\frac{2rd}{R(R+r)}} \tag{3}$$

$$L_{et} = \sqrt{1 + \frac{t^2}{4\pi^2 r^2}} \times \left(R \pm \frac{rN_a}{N_\gamma}\right)\sqrt{\frac{2rd}{R(R+r)}} \tag{4}$$

$$\tau_s = \frac{T_s d}{S \pm T_s} \tag{5}$$

$$\tau_{ep} = \frac{N_a d}{RN_\gamma \pm rN_a} \tag{6}$$

$+$ for up cut, $-$ for climb cut in all cases. Demanded rate of metal removal is proportional to $d \times T_s$ for surface-grinding, to $2\pi rdN_a$ for plunge-cut grinding, and to

$$\sqrt{4\pi^2 r^2 + t^2} \times d \times N_a$$

for external grinding with traverse.

The individual wedges removed per grain have the same areas for both "up" and "climb" cut, but those for up cut are longer than those for climb cut, hence the *average* and *maximum* thickness of chip per grain is greater for the "climb" cut case.

For conformity changes only, due to changes in ratio of wheel to work sizes:

$$L \overset{\sim}{\propto} \sqrt{K} \quad \text{for all cases} \tag{7}$$

Heating of the Work

Consider for the up-cut case the sequence in which things happen in building up a chip.

Ideally the cutting particle immediately enters the work at the size point at the thin end of the wedge, increases depth of cut per grain until almost through, and rapidly diminishes this depth as it breaks out.

A radial force is required to make the grain penetrate the stock at the thin end of the wedge. Once this penetration has occurred the material piles up ahead of the grain and cutting proceeds, the parent metal failing by compression and shear. The duller the edges, the greater the required force for penetration. This radial force is produced by the elastic reaction of the grain, its bond support, the surface of the work, the workpiece as a whole, and the grinding machine itself.

The combined rigidity of all these things is akin to that of a very stiff spring, and a very slight linear yield of all these parts may be accompanied by a relatively great change in force acting along the line joining centers of wheel and work; conversely, a slight spread in centers requires a fairly large change in the force acting.

So, if the work be not penetrated immediately by an abrasive particle, such a particle slides along the surface of the work until, by interference, a sufficiently large force has been built up to start cutting.

The "up-cut" wedge is very slender at its thin end, a minute change in centers then corresponds to a fair proportion of the

total length of theoretical cut being skidded over by the grain under continually increasing normal force.

Further, what happens *first* to the wheel, happens last to the work hence, in using a wheel with somewhat dulled abrasive, the work is scrubbed frictionally by such grains instead of being cut very lightly. This scrubbing occurs as the work swings *away* from the wheel.

As the grain edges dull still more, the forces exerted on them by the work become great enough and so directed as to either shear the grain or tear it loose from its bond.

This action occurs continually in use but, when it occurs too rapidly or if the lack of absolute uniformity of structure of the wheel is such that it occurs non-uniformly around the periphery, in shop terms the wheel is said to "break down."

It is difficult, if not impossible, with data at hand to give any quantitative treatment of this phase of the work but, by taking rather extraordinary liberties with the Herz theory of bodies in contact, a guided guess may be made as to what qualitatively may be expected to happen.

Mentally replace the abrasive grain by a something having a spherical end, radius r_s, which is very small in comparison with the radius of the work being ground, and assume this something pressed normally to the work surface.

Then the Herz theorem gives a maximum compression stress at the center of the spot of contact expressible (strictly only for static conditions) as

$$F \propto \frac{P^{1/3} H^{1/3}}{\lambda^{2/3}} \tag{8}$$

where P is the normal force.

λ is a factor depending on conformity between equivalent sphere and work.

H is a function of the elastic properties of the materials in contact.

In the limit, λ becomes proportional to $\frac{r_s}{2}$ for the case of sphere and plane.

We do not know what F is in force per unit area but, for convenience, we may call it unity.

$$\text{Then, Unit Stress} \propto \frac{P^{1/3}}{r_s^{2/3}}$$

And next suppose the unit stress to be just high enough to cause surface failure of the work, permitting the sphere to dig in deep enough that material will pile up ahead of it if it be moved sideways.

Then $r_s^{2/3} \propto P^{1/3}$

And, cubing both sides:

$$P \propto r_s^2$$

Now let δ be called the dullness, and take this factor as proportional to r_s, when

$$P \propto \delta^2 \tag{9}$$

that is doubling the dullness quadruples the normal pressure, so the effective spread of wheel and work centers is proportional to P, hence to δ^2. And the length of path of skidding along the thin end of the wedge becomes likewise proportional to P and, assuming constant coefficient of friction between dull abrasive and work, the frictional work done during skid becomes proportional to the product $\frac{P\delta^2}{2}$, so that the temperature rises of both work and wheel are likewise so proportional, neglecting the effect of coolant.

The H function includes the stiffnesses of the abrasive and the work themselves, independent of their manner of support. For given work material these stiffnesses are constant, and ease of penetration may be encouraged by increasing that of the abrasive itself.

In the climb-cut case, the grains start to cut at the big end of the wedge, and a relatively large geometrical interference is rapidly established and, arguing from other metal-cutting experience, once a chip is started it is not especially difficult to maintain it, as happens, for example, in the hand-scraping of surfaces.

Next, consider the effect of grain spacing or wheel structure. The chip material is composed of small pieces which have parted from the parent metal by a combination of shear and compression failure, just as the chip is deformed from any other metal-cutting tool edge. The material of the chip, being harshly worked, becomes quite hot, and its net volume increases. The volume entirely enveloping the chip is much greater than that of the chip itself and, as the chip grows due to the abrasive edges continually advancing through the metal, so must the overall volume of the chip *ideally* remain smaller than the intergranular volume at the wheel perimeter. As the overall volume of the chip increases above this value, the chip itself is forged between the wheel and parent material and heated still further, even to its melting point. With relatively large chips and a somewhat dovetailed intergranular space, it is easily possible to forge the chip into this space. When the bond is sufficiently strong, this forged-in chip stays in the wheel – the "loaded" condition.

As the forging work done on the metal removed increases, the heating of the chip and the consequent additional friction work done by scrubbing the hot chip over the parent material yet to be removed combine to still further heat the wheel, the chip, and the parent material. A slight heating effect occurs in the parent material itself due to metal severance, neglecting the foregoing effects, but this effect is of itself very minute. Imagine a large vertical boring mill on which a ring 85 ft. or so in diameter is being turned with a cut ½ in. deep, and the steel chips are coming off blue. It is about, to scale, the same as grinding a 1-in. shaft, removing 0.001 in. diameter. The big chips are hot but the temperature of the parent metal is raised relatively slightly.

So, the two principal sources of heat in the work are those due to friction prior to the establishment of cutting, and friction due to entrapment of chips after establishment of cutting.

It has been shown that *both* of these sources depend on some kind of surface conformity – the first on that between grain surface and work surface, and the second on that between wheel surface and work surface.

The effect of these things, for conditions of infinite wheel surface conformity and variable length of chip path, may be examined readily as follows:

Prepare a series of samples of, say medium steel, having cross-sections, say ¼ in. wide and thicknesses varying by sixteenths between 1/16 and ⅜ in. Successively clamp them in a vise on the table of a cutter-grinder so that their various thicknesses are in direction tangential to the motion of the grinding wheel and push them up toward and against the wheel with forces proportional to their thicknesses.

Collect samples of the grindings on filter papers close to the work, taking for observation samples ground when the wheel just fits the work. Examine them at say 50 diameters magnification. The character of chips changes remarkably as the length of chip increases. For short lengths of path the chips are curly but, as length of path increases, the curls are less frequent and, instead, little balls of material replace them. With great lengths of chip path the curls are almost absent. The sphericity of the ball particles indicates that they have been either molten or very close to it, their shape being consequent to the joint action of gravity and surface tension. Repeat with "harder" and "softer" wheels. The "balls" increase in frequency with the "harder" wheel.

Heating in Shoulder-Surface Grinding

Shoulder-surface grinding or, as loosely termed in the shop, "bumping" a shoulder, frequently and often is done successfully. With carburized parts it may, however, be a risky procedure, a not infrequent consequence being the loss of a complete ring of the hardened material.

The situation is really as follows: The periphery of the grinding wheel, used for grinding the diameter adjacent to the shoulder, is that of a wheel whose properties are selected primarily to grind the cylindrical portion of the shaft satisfactorily. When the wheel is moved sidewise to finish the adjacent shoulder with the side of the wheel, the length of chip path increases enormously, and the conformity between wheel surface and work is infinite.

The depth of cut is now measured endwise of the work. The intergranular volume is not increased hence, with any but the lightest cuts, a chip may very easily overfill the chip space and tremendous heating may result. When internal stress conditions are just right, chunks of work may, and do, jump right off.

Instances exist where lumps of heat-treated steel-rolling-mill rolls as large as the palm of one's hand have come off under such misgrinding.

The shop often blames the heat-treat department, perhaps with some justification, but the fact remains that an excess in depth of cut of 0.0002 in. may, in critical cases, make all the difference between trouble and no trouble.

Dressing the side of the wheel slightly concave does not change the "cut-per-grain" situation much, but it does permit the wheel corner to break down somewhat faster and so promote effective openness of structure, and obviously promotes ease of grinding square-shoulder faces. So, in the end, this trick is of help.

So, where shoulder-bumping is anticipated, the wheel structure should obviously be as open as possible consistent with the satisfactory and economic production of the adjacent cylindrical surface. Then, to reduce hazard due to misfeeding, the machines themselves should be kept in smooth running order so that the cut may be felt by the operator.

Wheel Keenness and Scratchy Work Surface

A freshly trued grinding wheel may present a substantially cylindrical surface. During truing, certain abrasive particles may have been sheared through, others torn bodily from their holding bond and removed, and still others merely loosened, displaced perhaps from their original position and jammed into the wheel. Such latter particles are retained only by the friction of their setting. Particles encountering the work at high velocity are held radially inward by centripetal tension in the wheel and by the radial component of the resisting forces encountered at the grinding location. They are kept from shifting tangentially off the wheel by their own inertia and by the mechanical strength of the supporting bond.

In common with other tools, the grains dull with use, their *rate* of dulling seeming to depend upon their degree of dullness at the moment. Thus, they dull at first relatively slowly, and more rapidly as time goes on.

When the forces acting are great enough to free the grains, obviously they come out, those not firmly fixed emerging first.

Such emergence may permit other grains, perhaps partially shielded earlier by their departed fellows, to get into action so, to a certain extent, the grinding wheel is somewhat self-sharpening. However, it is *not* perfectly uniform in make-up and, as the foregoing phenomena occur, the originally round wheel becomes enough out of round to be useless for precise work and it must be re-trued.

The particles merely loosened and jammed into the wheel are by no means so firmly fixed as those properly bonded. As grinding proceeds, they often loosen and roll between the wheel and work, leaving scratches whose lengths approximate those of the interference paths.

Such scratches may be reduced in frequency of occurrence by giving the rotating wheel a slight rub with an oil stone just after diamond truing. Many loosened particles may be so removed.

Dirty grinding compound may contribute to scratch production, but it must really be fairly filthy to exert any controlling influence on this phenomenon.

A further cause of scratching will now be touched upon. In turning a cylindrical wheel with a diamond, the latter may be sharp-cornered or rounded. The feed of the diamond in truing may be say 1/32 in. per wheel revolution. With a sharp-cornered diamond a fairly sharp thread is chased on the wheel and, as the table reversal takes place, an opposite-hand thread is produced, making a series of protrusions on the wheel surface. This method makes a very keen wheel which initially cuts very freely but, as the protruding particles dull, their securing bond is not so strong and they readily break away and are carried on sometimes between the work. As the wheel loses these protuberances, it is still in fairly good shape and produces less scratchy surfaces. The smoothness of surface finish produced by a newly dressed wheel and the same wheel ready for re-dressing varies between wide limits under these circumstances.

The use of too sharp cornered a diamond is wasteful of grinding-wheel material.

Influence of Wheel Diameter on Dressing

Let $D = 2R$ = wheel diameter, in.
f = wheel face, in.
$f+a$ = dressing face, that is, wheel face plus overtravel for reversal of truing direction, in.
N_γ = wheel speed, r.p.m.
t_d = feed of dresser per revolution of wheel, in.
T = time for 1 pass across the dressing face, sec.

Then
$$T = \frac{f+a}{t_d} \times \frac{60}{N_\gamma} \text{ sec.} \qquad (10)$$

Now the area of the wheel periphery is πDf, and the diamond may or may not entirely dress the wheel surface depending upon whether or not it is rounded or sharp-cornered, and upon the depth of cut into the wheel when dressing occurs.

With *constant* surface speed and *constant* time allowance for dressing, a sharp-cornered diamond may contact a greater proportion of the wheel face for the product $N_\gamma t_d$ must be constant for given initial conditions, that is, the traverse in dressing is reducible as the wheel speed increases to maintain constant surface speed as the wheel radius is reduced by continued use.

With a more rounded diamond of curvature such that it will envelope say three times the wheel feed, the traverse per wheel revolution need not be reduced so fast if at all.

The sharper-form diamond will leave more ridges and pimples on the wheel so, for finishing, a rounder form usually is preferred.

It is desirable to reduce the depth of cut of such diamond to a minimum (a) to conserve wheels; (b) to conserve diamonds; and (c) to dress cleaner leaving fewer incipient scratch producers on the wheel surface.

Stuck particles may have been pushed out of the way by the diamond during one wheel revolution and missed in the next revolution if the diamond be of too sharp a shape. The rounder diamond has more likelihood of contacting them on a subsequent revolution and either picking them loose or pushing them in tighter.

The centrifugal force on a stuck particle, tending to throw it out and thereby cleaning the wheel, is proportional to its mass, the radius of the wheel, and the square of the wheel speed in r.p.m.

For *constant wheel r.p.m.* both the likelihood of stuck particles existing and the self-cleaning tendency of the wheel are proportional only to the wheel diameter for a given shape diamond, depth of cut, and dressing time.

However, for *constant surface speed* and time of dressing cycle, as the wheel diameter is reduced, the likelihood of stuck particles existing is reduced, and the centrifugal self-cleaning tendency of the wheel varies inversely as the wheel radius, halving the wheel size doubling the self-cleaning tendency.

As the wheel is reduced in size and its surface speed maintained, the lengths of such scratches as may occur are reduced with the consequent reduction of the grain interference path *L*.

Hence, for freedom from scratches, small diameters of wheels are indicated.

However, since a wheel can never be used up completely and since the part discarded is just as expensive per unit volume as that productively used, wheel cost per article made is often lessened as wheel size increases and, finally, wheel sizes are set by commercial compromise.

Function of the Coolant

Grinding compound pumped onto the wheel performs two principal functions: (1) reducing the *average* temperature of the wheel, and (2) reducing the *average* temperature of the work.

Of the 20 or so gal. per min. which the pumps of many machines can furnish, it is extremely debatable whether more than a very few cubic inches per minute are taken through between the wheel and work in the intergrain spaces of the wheel.

With up-cutting the provision of a large supply of coolant, when grinding materials which have been subjected to drastic heat-treatments, may be hazardous in the following respect:

Suppose the grinding wheel to be in an intermediate stage of dullness, that is, the edges of the average abrasive grain are dulled to such an extent as to ride the work fairly hard before penetration builds up, but not so dull as to involve rapid wheel breakdown.

As the work swings up away from the wheel, the hot scrubbed surface immediately is subjected to a violent quench with consequent sad results in the formation of grinding cracks, in some cases.

It is not necessary to have a liquid coolant to produce such cracks, for the heat in the surface layer is dissipated two ways, inwardly to the colder core of parent material, and outwardly to the environment, be it air or a liquid.

The *material* characteristics encouraging crack production are low strength coupled with high hardness and high coefficient of expansion coupled with low thermal conductivity.

Coolants with ability to wet the work surface readily, and those of relatively high heat conductivity will surface-quench the work faster than others, encouraging cracking. Again some emulsified-oil coolants break down chemically and form a kind of sticky mess on the face of the wheel in certain instances.

During experimentation with wheels of differing qualities, it is most convenient to be able to see the work as it comes up from the grinding wheel in external grinding. This point argues for as transparent a coolant as possible.

Initially the writer believes a mixture of 5 lb. of ordinary automobile soap jelly to 40 gal. of water to be a satisfactory starting mixture for coolant purposes. It is cheap, available anywhere, and can be seen through quite readily. As a production coolant it has been used fairly extensively with satisfaction.

The thought has been advanced that, were a portion of the coolant delivered radially through the wheel, it might, in the intergrain spaces, quench the chip as it is being removed, and so reduce the tendency to heat.

A number of obstacles lie in the way of the accomplishment of this desideratum. They are:

(1) Only wheels of porous nature could be used.

(2) To be sure the coolant was present in each intergrain space, the wheel would have to run full of coolant.

(3) To run full of coolant, without cavitation, the meridian section of the wheel would have to be hyperbolic.

(4) The sides of the wheel would have to be watertight under the centrifugal pressure of the coolant.

(5) Assuming that all these requirements were met, the volume of coolant available per chip would be insufficient to more than partially quench the chip as it was made.

(6) Coolant at the intergrain space at the work would not necessarily reduce the tendency of a dulled grain to skid in "up"-cutting.

(7) Coolant at the intergrain space behind an active grain would tend to overquench the immediately pre-ground or pre-rubbed surface.

But the centrifugal pressure of the coolant would tend to clean the wheel face.

The wheel itself must be cooled for at least the reason of reduction of stress at the inner diameter. Mounting wheels with blotter washers between wheel and flange, although providing adequate traction for the drive, does not really restrain the wheel to a great extent.

Assuming the wheel elastic, it is an interesting fact that the circumferential stresses are greater than the radial stresses, both are tensile, increasing toward the center. High rim temperature will, by expansion, increase the stress at the inner diameter until finally the wheel tears open from the inside, under the joint influence of temperature and speed.

The individual grains and their bond are highly heated locally by friction with respect to the work piece and hot severed chips and their temperature must be controlled reasonably to prevent loss of grains still satisfactorily sharp by undue weakening of the bond due to temperature.

Cam-Profile Grinding

Cam-profile grinding is discussed here because it has been until very recently about the only instance in which it was possible to observe a cycle of operations of similar kinds, during which cycle the wheel becomes progressively duller and, at the same time, the curvatures of the work produced and, hence, the degrees of conformity between wheel and work vary systematically over a wide range. The adoption of the full-automatic cam grinder which, for a given setting of controls, eliminates the human factor in introduction of variables makes this phase of commercial grinding still more interesting and informative.

A typical "three-arc" cam shape is shown in Fig. 1.

Initially, we shall assume the angular velocity of the work to be uniform. For certain types of cam grinders this is not an exact truth.

As the profile is ground, conditions vary from zone to zone, and may be considered as though at the same work and wheel speeds, in succession are ground shafts of 1⅜, 1.4, 6 3/32, 2 1/16, and 0.17 in. diameters as grinding proceeds from heel to tip of cam.

The demanded rate of metal removal per abrasive grain will be inversely proportional to the ratio of wheel perimeter passing the work per unit length of work.

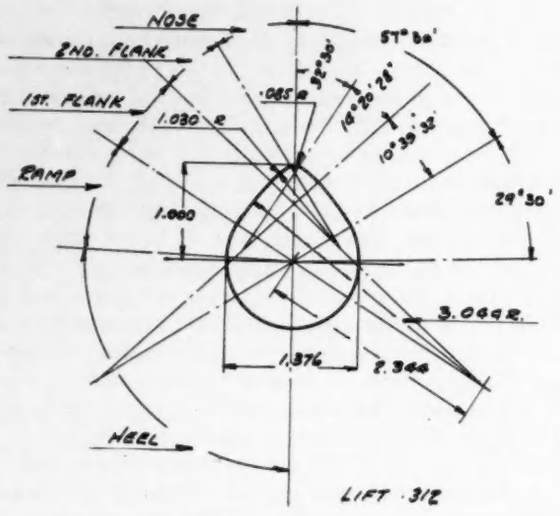


Fig. 1—Typical Cam Shape

Hence, from the standpoint of *minimum* rate of heat production in the work, this item should be somewhat proportional to product of depth of cut times radius of curvature, and is so illustrated in Fig. 2. This chart is very idealistic and, in ordinary manufacture, can only be approached but never fully realized.

Among upsetting influences are local deviations from expected amounts of finish to be removed, local variations in hardness and strength of material ground, and the continual dulling of the grains with use.

Suppose that we have rough-ground the present example and, in roughing, have produced an accurate profile leaving some 0.005 in. material uniformly distributed around the theoretical shape and size of finished cam. Now, suppose we finish-grind on a second machine which, in turn, will produce accurate profiles, but that the work driver is mis-set 0 deg. 20 min. ahead of its best position.

In the conventional up-cut case, the effect on the work is as tabulated in Table 1. Angular deviations of 0 deg. 20 min. from perfection are by no means uncommon in manufacture, especially in the shifting of work from roughing to finishing operations from machines of one type to those of another.

When, additionally, profile errors from roughing machines are included, and mostly these errors are on the "fat" side; to be sure the work subsequently will clean up, but the charted presentation of such effects is rather disturbing.

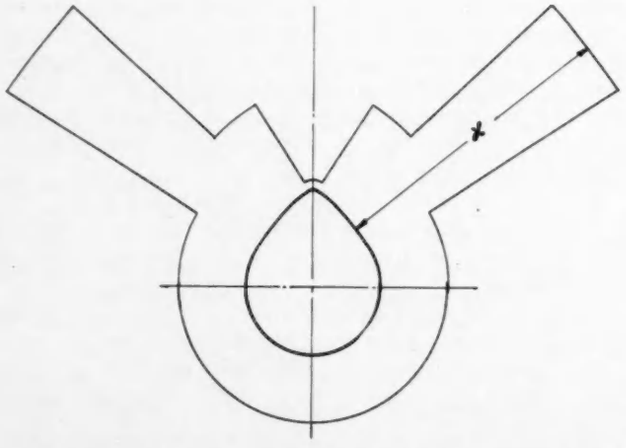


Fig. 2—Ideal Case—Distribution of Minimum Tendency for Heating Work Surface—Tendency Measured as *X* Normal to Work Profile

Fig. 3 shows the difference between anticipated and realized stock distribution on one cam on a shaft, roughed, measured, finished and measured. The designed profile of this freak was *not* that of the example dimensioned in Fig. 1. Investigation showed three contributing causes for this peculiar shape, namely, too fat a shape from the roughing master, mis-index of the roughing and finishing work drivers, and a small flat on the master roll of the roughing machine. The two latter items were fixed up, and the distortion reduced to quite a reasonable shape.

Table 2 shows the ideal interrelationship between length of wheel rim swinging past the various zones of the illustrated cam profile.

For uniformly distributed stock, the last column gives figures proportional to the amount of material removed per grain. The high demanded duty of the wheel in the first

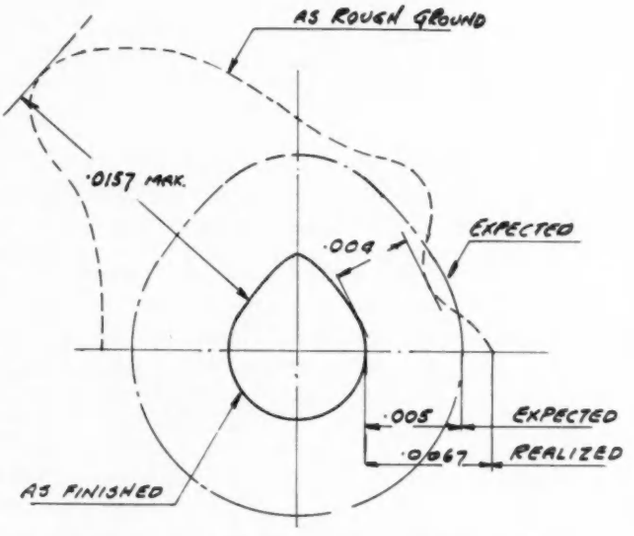


Fig. 3—Erratic Stock Distribution Due to Careless Rough-Grinding

flank zone is plain to see, and this normal high demand is increased by the presence of excess stock due to poor work in preparation of the shape for grinding.

The higher the demanded duty of the wheel, the more rapid its rate of dulling, local surface temperature rise, and consequent rate of breakdown.

The duty demanded of the wheel in the critical region, the first flank, can, in the practical case, be reduced in two ways:

(1) By locally reducing the amount of finish in this region.

(2) By increasing the wheel speed.

General Wheel Characteristics

In any wheel application we find certain wheel characteristics consequent to the peculiarities of the specific application.

Fig. 4 qualitatively shows a typical characteristic chart, the upper portion showing bond strength and necessary bond strength as a function of time, assuming continuous grinding, or as number of similar parts ground. The difference in ordinate at any point represents the margin of bond strength available, and the rate of breakdown of surface may, as a first approximation, be taken inversely as the margin, the time integral of which rate determines the extent of breakdown; curves on the middle portion of the chart exhibit these prop-

LOCATION ON CAM.	EXTRA ″ UP SIDE	EXTRA ″ DOWN SIDE	% OF EXPECTED UP SIDE	% OF EXPECTED DOWN SIDE
HEEL	0	0	0	0
RAMP	+.00018	-.00018	+ 2.66%	- 2.66%
START OF 1ST FLANK	+.0002	-.0002	+ 4%	- 4%
START OF 2ND FLANK	+.0024	-.0024	+ 48%	- 48%
START OF NOSE	+.0028	-.0028	+ 56%	- 56%
TIP OF NOSE	+.00006	-.00006	+ 0.3%	- 0.3%

Table 1—Finish Allowance Deviations Due to Mis-Indexing of Work Driver

ZONE	PROFILE LENGTH-IN	ROTATION RADIANS	TIME IN SEC.	WHL. TRAVEL FT.	WHL. TRAVEL UNIT LENGTH	UNIT TRAVEL FT WHL TR.
HALF HEEL	1.11072	1.61448	.51388	51.388	46.265	.02161
RAMP	.33372	.51847	.16389	16.389	49.866	.02180
1ST. FLANK	.36333	.18703	.05953	5.9533	10.497	.09363
2ND. FLANK	.25731	.25040	.07970	7.9703	30.906	.03236
HALF NOSE	.04821	.56723	.18053	18.053	374.47	.00267

S = 6000 FT/MIN.

N_w = 30 WORK R.P.M.

Table 2—Wheel Travel per Unit Profile Length for Cam of Fig. 1

erties. For each job there exists a practical limit to the extent of permissible breakdown beyond which the work surface is geometrically unsatisfactory. The intersection of this horizontal practical limit with the "extent" curve gives the usable time between dressings, or the number of pieces per dressing.

The lower curve represents the rise in temperature of the work surface with time of use. Its characteristic is that of a logarithmically increasing function snubbed to some extent by a rate-of-increase reducing tendency consequent to the somewhat self-sharpening tendency of moderate wheel breakdown.

On certain work, a maximum practical work surface temperature exists beyond which the thermal stresses induced in the work by the extremely rapid surface-heating and surface-quenching occurrences induce surface-cracking.

Ideally this temperature maximum should be reached at the time at which the practical breakdown limit of extent should occur, and both should come as late as possible, with least loss of abrasive grain.

Fig. 5 depicts roughly the working characteristics of a wheel successfully used in finish-grinding an eight-cylinder engine camshaft with heat-treated cams, and incorporating a chill-cast eccentric which is the fifteenth cam ground. The wheel is dressed immediately before grinding the first cam on each shaft ground.

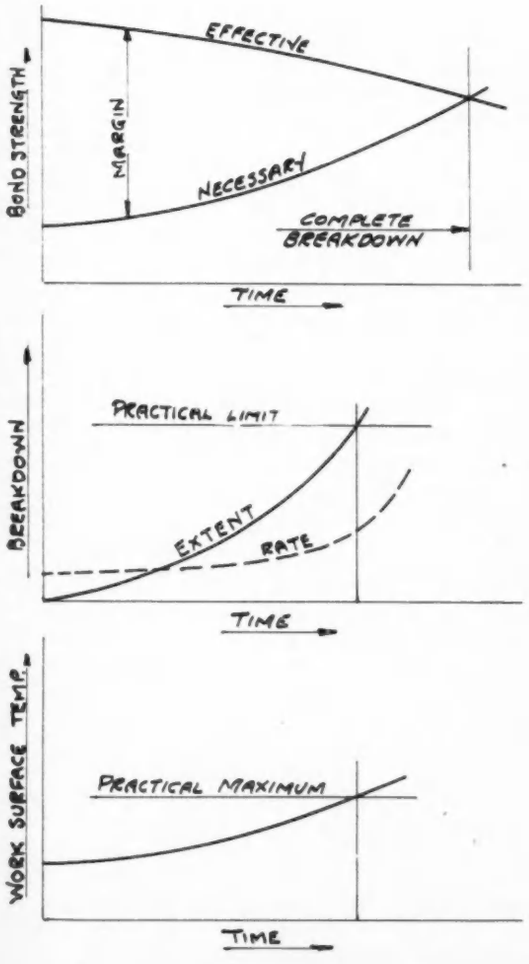


Fig. 4—Typical Wheel Characteristics

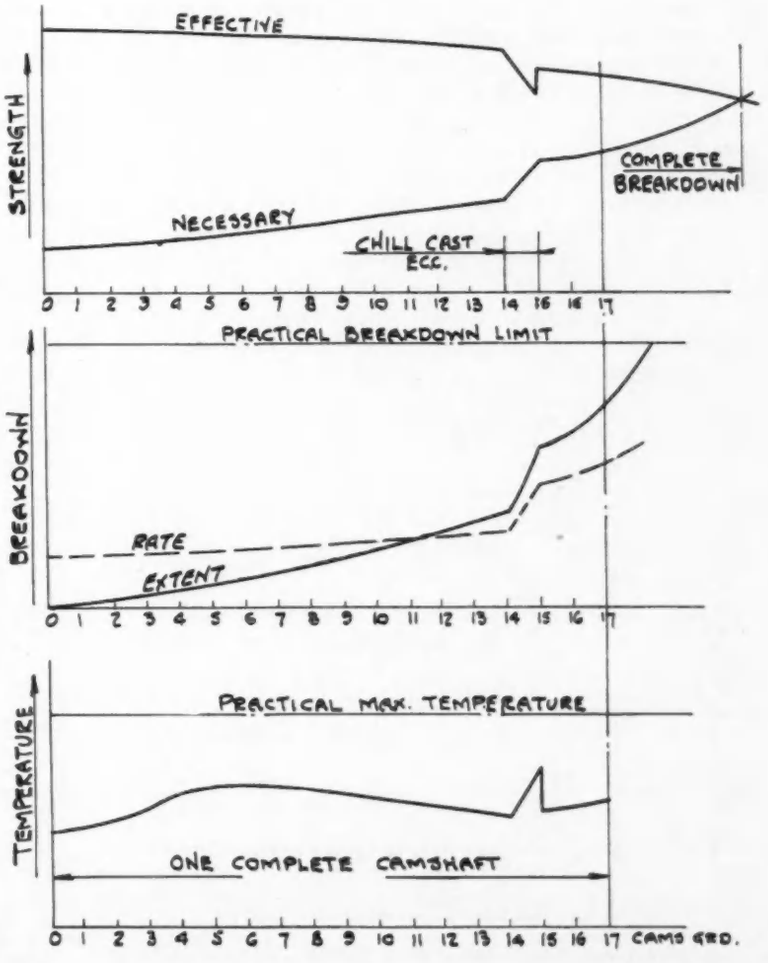


Fig. 5—Rough Depiction of Wheel Characteristics for Finish-Grinding Camshafts on Automatic Machines

Recapitulation

From the foregoing we may argue qualitatively:

(1) Heating of both work and wheel increases at a much faster rate than does the dulling of the grain.

(2) Heating of the work surface may be reduced by: (a) stronger abrasive, (b) weaker bond, (c) wider spacing of existing abrasive grains, (d) increasing wheel speed combined with the foregoing.

(3) Heating of work surface is partially proportional to length of chip path for given sharpness of grain. Length of chip path is proportional to square root of radial depth of cut, that is, quadrupling the cut depth doubles the chip path.

(4) Average and maximum thicknesses of chip per grain are roughly proportional to radial depth of cut; doubling the cut depth almost doubles the chip thickness.

(5) Tendency to wheel breakdown increases with depth of cut.

(6) Heating of the wheel depends jointly upon the rate of dulling of the wheel, and upon the extent of filling of the intergrain cutting clearance spaces in the active wheel face.

(7) Heat developed at the work surface may be somewhat reduced by climb-cut grinding.

(8) Friction between wheel and work due to scrubbing as the work passes the "size-point" is potentially reducible in climb-cut grinding.

(9) In grinding materials prone to heat checking, it is of first importance to keep the work surface cool and free from violent temperature changes. Rather than attempt this cooling with copious use of coolant, in some instances, it may be better to cool the wheel separately, say by air blast or spray, at the same time using a very hard and strong abrasive grain in a fairly weak bond, with structure open enough to easily accommodate the chip volume per grain. To attain good wheel life it may be necessary to run at high surface speeds.

(10) Where large-percentage variations of local finish allowance may be anticipated, as in second-operation cam grinding, it is sensible to: (a) use maximum possible surface speeds, (b) use wheels of very open structure, and (c) use (a) and (b) in combination.

(11) In finish-grinding cams, especially on full-automatic grinders, it is essential that adequate infeed control be had at all times.

Erratic infeed and disposition of finish have as major effects: (a) actual depths of cut during first few work revolutions per cam may become excessive; (b) consequent oversize chip per grain may result in undue work and wheel heating and acceleration of wheel breakdown rate; and (c) cam profile grinding may be incomplete.

(12) In cylindrical grinding with power traverse, the requisite openness of wheel structure required for optimum results increases with traverse speed.

(13) If traverse rate alone be increased, the resulting greater chip volume per grain stimulates wheel heating, while the more attenuated chip wedge encourages work heating as the grains dull.

(14) For constant length of chip path, given wheel and work diameters, the relationship between traverse rates and work revolutions are as given on Fig. 6.

(15) In finish-grinding, minimum disturbance of the wheel surface in dressing is desirable.

(16) Minimum conformity between wheel and work will result in finished surfaces most free from scratches due to "dragged-through" particles from the dressed surface of the wheel.

(17) Where conformity varies, as in cam-grinding, greatest freedom from scratches will result when the finish-grinding infeed occurs while the nose is being ground.

(18) Climb-cut grinding in finish-grinding is probably attended by fewer surface scratches than is "up-cut" grinding.

(19) Expressions have been given permitting the fairly ready calculation of the geometrical interference properties for surface and external grinding. By suitable changes of sign the analysis equally applies to the internal-grinding case.

(20) Approximate relative lengths of chip interference path for plunge-cut grinding may be obtained from Fig. 7, really in terms of wheel and work conformity.

Closure

This whole business is a study of transients. The grains dull, tending to heat the work and increase the force required to cut until the duller and less well supported ones leave the wheel.

These things increase the duty on the good grains remaining, the keener ones of which cut freely and tend to run cooler. But there are fewer of them, as the wheel is slowly growing out of round, so they have each heavier duty and the whole cycle of interrelated dulling, heat production, wear, and breakdown is cumulative at an increasing rate with time. Since the bond may weaken with increased temperature, and its *effective* strength certainly reduces as less of it is available for usefully supporting such grains as are in action, the *effective* strength *must* become lowered as time goes on.

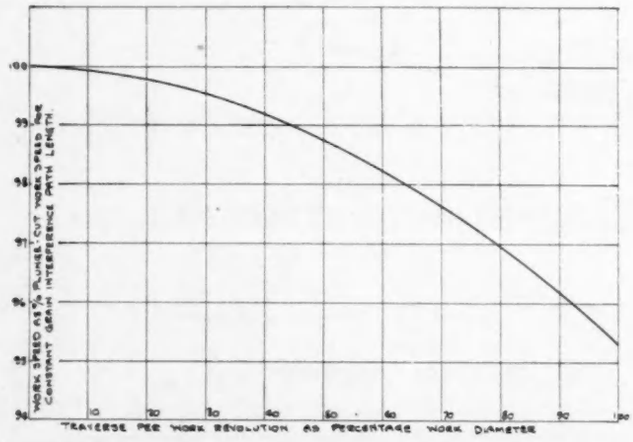


Fig. 6—Interrelation between Work R.P.M. and Traverse Speed To Maintain Constant Length of Grain Interference Path—External Cylindrical Grinding

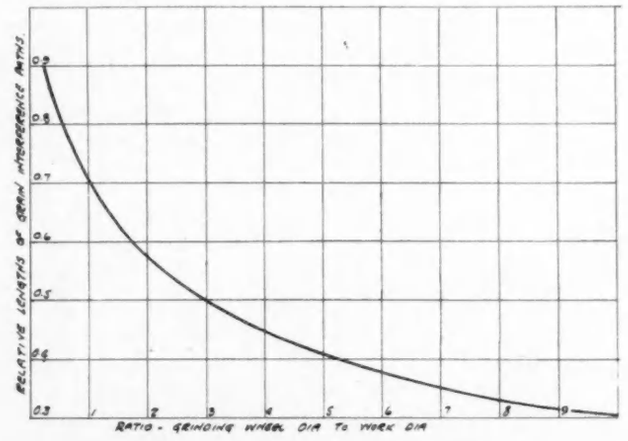
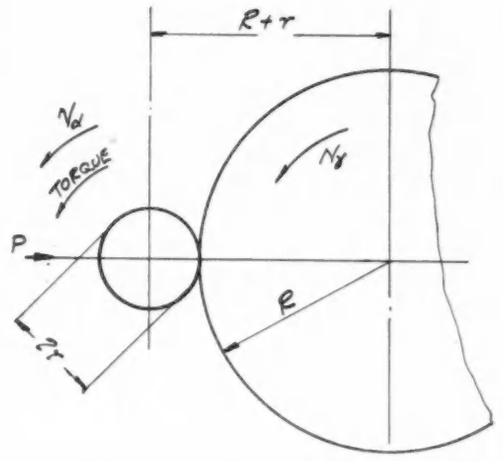


Fig. 7—Approximate Relative Lengths of Grain Interference Paths—Plunge-Cut External Grinding—This Approximation Neglects Rotational Senses and Speeds and Is a Conformity Function Only



MECHANISM ARRANGED TO CONTINUOUSLY MEASURE AND RECORD
$2r$, $R+r$, N_γ, N_w, P & TORQUE.
OUT OF THE TIME RATES OF CHANGE OF THESE QUANTITIES THE LIFE-CONSTANTS OF THE WHEEL MAY BE OBTAINED.

Fig 8—Basic Principle of One Form of Grinding-Wheel Dynamometer

Further, since the force per remaining active grain increases with time, the necessary strength to maintain cutting at the demanded volume removal rate must go up.

In an ideal application, once the wheel is dressed, its self-sharpening tendency due to breakdown should just compensate its dulling tendency due to work done, and such a wheel could, in theory at least, go on until quite worn out with *no* redressing.

A grinding-wheel dynamometer is sketched schematically in Fig. 8. Its existence may as yet be a mere figment of the imagination. Did such a machine exist and were it satisfactorily used, out of data taken with it, the constants governing life and suitability of a given wheel for a given job could, with sufficient background, be fairly well predicted.

Difficult things to measure directly are the surface temperature of the work and its rates of heating and cooling. The writer has devoted considerable thought to accurate determination of these important quantities, with no success. The experienced temperatures, so far as he knows, can only be inferred qualitatively and by indirect methods. The *average* temperatures are easy enough to measure, but often mean exactly nothing.

Wheels for an appreciable time to come will, in all probability, be selected by trial and error on the job, though dynamometry may provide a series of guide posts to steer one's guessing, so to speak.

So, really, after all, the writer knows very little in detail about the grinding process! Do you?

Appendix A

Expression for Conformity Factor K

An expression is required to exhibit, in terms of the radii of wheel and work, the closeness of fit of their surfaces when in contact, their axes being parallel.

A suitable expression, by inspection, is of the form

$$K = \frac{1}{1/r_1 + 1/r_2} = \frac{r_1 r_2}{r_1 + r_2}$$

This expression fits all cases. Take general and limiting examples:

(a) Cylinders $r_1 = 2$ $r_2 = 12$

$$K = \frac{2 \times 12}{2 + 12} = 1.714+$$

(b) Cylinder and plane $r_1 = 2$ $r_2 \rightarrow \infty$

$$K = \frac{1}{1/2 + 1/\infty} = 2$$

(c) Cylinder and hole $r_1 = 2$ $r_2 = -4$

$$K = \frac{-4 \times 2}{-4 + 2} = 4$$

(d) Knife edge and plane $r_1 = 0$ $r_2 = \infty$

$$K = \frac{0 \times \infty}{0 + \infty} = 0$$

(e) Two knife edges imagined in line contact $r_1 = r_2 = 0$

$$K = \frac{1}{1/0 + 1/0} = 0$$

(f) Two planes $r_1 = r_2 = \infty$

$$K = \frac{\infty \times \infty}{\infty + \infty} = \infty$$

Note in the case of cylinder and plane $K = r_1$

This expression is the same as saying in surface grinding that the conformity of the wheel cylinder with the surface ground is equal to the radius of the grinding wheel.

Scale effect is as follows:

Let r_1, r_2 be radii of curvature of a pair of cylinders in line contact, and pr_1, pr_2 those of a geometrically similar pair, then

$$K_1 = \frac{r_1 r_2}{r_1 + r_2} \quad \text{for the first set.}$$

$$K_2 = \frac{p^2 r_1 r_2}{p(r_1 + r_2)} \quad \text{for the second.}$$

$$\frac{K_2}{K_1} = p$$

That is, the ratio of new to old conformity equals the scale.

Appendix B

Surface Grinding Case

Characteristics of Wheel Interference
Point Motion With Respect to the Work

When the work is moved tangentially to the wheel, the path of a cutting point through the work may be equally represented by imagining the wheel to roll on a track so that its translational speed is that of the work traverse.

The interference-path shape is a looped trochoid, the rolling circle for which is set in size by the angular velocity of the wheel and the rate of traverse, since the circumference of such circle must equal the traverse per wheel revolution.

Let R = wheel radius, in.
T_s = traverse speed, ft. per min.
S = wheel surface speed, ft. per min.
N_γ = wheel speed, r.p.m.
n_s = radius of rolling circle, in.

Then $N_\gamma = \frac{6S}{\pi R}$ r.p.m.

Traverse per wheel revolution $\frac{T_s}{N_\gamma} = \frac{2\pi R T_s}{S}$ in.

Traverse per wheel revolution = circumference of rolling circle

Hence $\frac{2\pi R T_s}{S} = 2\pi n_s$

$\therefore\ n_s = \frac{T_s}{S} \times R$

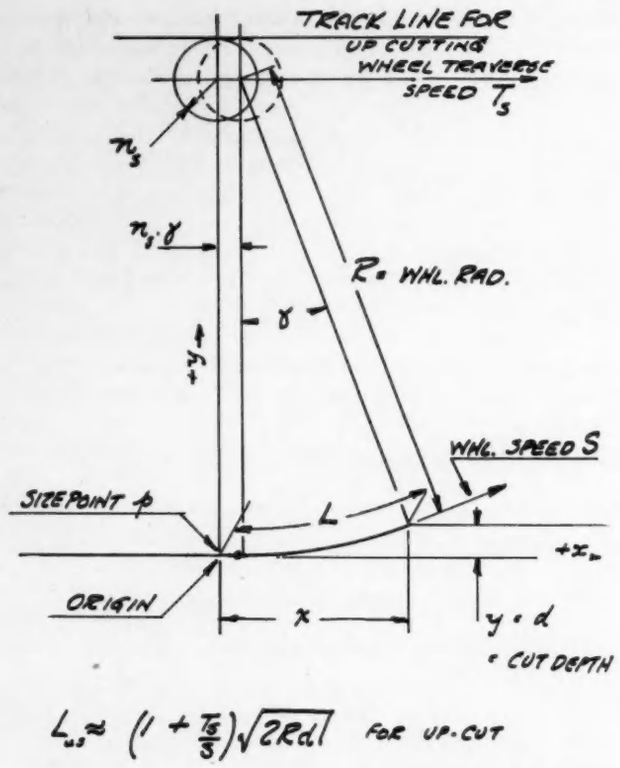


Fig. 9—Diagram for Calculation of Interference Path

Refer now to Fig. 9 in which origin of coordinates is the size point p. The circle, radius n_s, rolls without slipping on the line $y = R + n_s$ and its radius extended to length R represents the radius of the wheel, and traces out the trochoidal path pd, cutting through the work-surface line $y = d$ *at* d. Then

$$x = R\left(\sin\gamma + \frac{T_s}{S}\gamma\right)$$

$$y = R(1 - \cos\gamma)$$

$$\frac{dx}{dy} = R\left(\cos\gamma + \frac{T_s}{S}\right)$$

$$\frac{dy}{d\gamma} = R\sin\gamma$$

$$\left(\frac{dx}{d\gamma}\right)^2 + \left(\frac{dy}{d\gamma}\right) = \left(\frac{dL}{d\gamma}\right)$$

$$\frac{dL}{d\gamma} = R\left[\frac{S^2 + T_s^2}{S^2} + \frac{2T_s}{S}\cos\gamma\right]^{\frac{1}{2}}$$

$$\int_o^L dL = R\int_o^\gamma \left[\frac{S^2 + T_s^2}{S^2} + \frac{2T_s}{S}\cos\gamma\right]^{\frac{1}{2}} d\gamma$$

an elliptical integral of the second kind. The indicated integration becomes a trifle unwieldy, and for the practical small values of γ, the following approximation is simple, convenient, and expresses the curve length in terms of familiar dimensions.

Suppose the length of the curve to consist of the sidewise shift of the rolling circle plus the arc $R.\gamma$.

γ is given *exactly* as

$$\gamma = \cos^{-1}\frac{R - d}{R} = \sin^{-1}\frac{\sqrt{2Rd - d^2}}{R}$$

and for the experienced small γ values

$$\gamma \approx \frac{\sqrt{2Rd - d^2}}{R}$$

and $$L \approx (R + n_s)\frac{\sqrt{2Rd - d^2}}{R}$$

But $$n_s = \frac{RT_s}{S}$$

$$\therefore\ L \approx \left(1 + \frac{T_s}{S}\right)\sqrt{2Rd - d^2}$$

When d, the cut depth, is small, again as an approximation d^2 may be neglected and we have

$$L \approx \left(1 + \frac{T_s}{S}\right)\sqrt{2Rd}$$

for the "up-cut" case. For the climb-cut case the curve length approximates the arc $R.\gamma$ minus the sidewise shift, which makes

$$L \approx \left(1 - \frac{T_s}{S}\right)\sqrt{2Rd}$$

Thus

$$\left.\begin{aligned} L_{su} &\approx \left(1 + \frac{T_s}{S}\right)\sqrt{2Rd} \\ \text{and}\quad L_{sc} &\approx \left(1 - \frac{T_s}{S}\right)\sqrt{2Rd} \end{aligned}\right\} \qquad (2)$$

and L_{su} and $L_{sc} \propto \sqrt{R} \propto \sqrt{K}$ where K = the conformity.

The approximate average thickness of chip per grain may be estimated as follows:

Length of traverse per grain cut $n_s\gamma$

$$n_s\gamma = \frac{T_s}{S}\sqrt{2Rd - d^2}$$

Volume removed per grain cut

$$\frac{T_s}{S}\sqrt{2Rd - d^2} \times d$$

Length of interference path L per cut is

$$L \approx \left(1 \pm \frac{T_s}{S}\right)\sqrt{2Rd - d^2}$$

Hence average chip thickness = Volume ÷ Length

$$\left\{\frac{T_s\sqrt{2Rd - d^2}}{S} \times d\right\} \div \left\{\left(1 \pm \frac{T_s}{S}\right)\sqrt{2Rd - d^2}\right\}$$

$$= \frac{T_s d}{(S \pm T_s)} \qquad (5)$$

This expression assumes that one grain comes into action exactly as its predecessor finishes. With closer grain spacing the chip per grain reduces proportionately. Treating the chip as approximating a triangle, the maximum chip thickness will be about twice the average thickness, and the increase of these dimensions for "climb" cut over "up" cut cases will approximate $\left(\frac{S + T_s}{S - T_s} - 1\right) \times 100$ per cent.

Appendix C

External Grinding Case – Plunge Cut

Neglect the minor effect of infeed, or assume infeed to be intermittent, and consider the stable condition between successive feed motions.

The interference motion is, for the conventional or "up"-cutting case, a portion of a looped hypotrochoid, and for the climb-cutting case a portion of a looped epitrochoid.

Fig. 10 illustrates the "up"-cut case.

Let R = wheel radius, in.
S = wheel surface speed, ft. per min.
N_γ = wheel speed, r.p.m.
N_a = work speed, r.p.m.
r = size radius of work, in.
d = radial depth of cut, in.
m = radius of track circle, in.
n = radius of rolling circle, in.

Then C = centers = $R + r$

$$N_\gamma = \frac{6S}{\pi R}$$

$$\frac{m}{n} = \frac{N_\gamma}{N_a}$$

In the figure $\alpha = \frac{n}{m} \cdot \gamma$

We have given the three sides of a triangle, respectively

C, R, and $r + d$

$$\text{Then } \gamma = \cos^{-1} \frac{C^2 + R^2 - (r+d)^2}{2CR}$$

$$= \cos^{-1} \frac{(R+r)^2 + R^2 - (r+d)^2}{2CR}$$

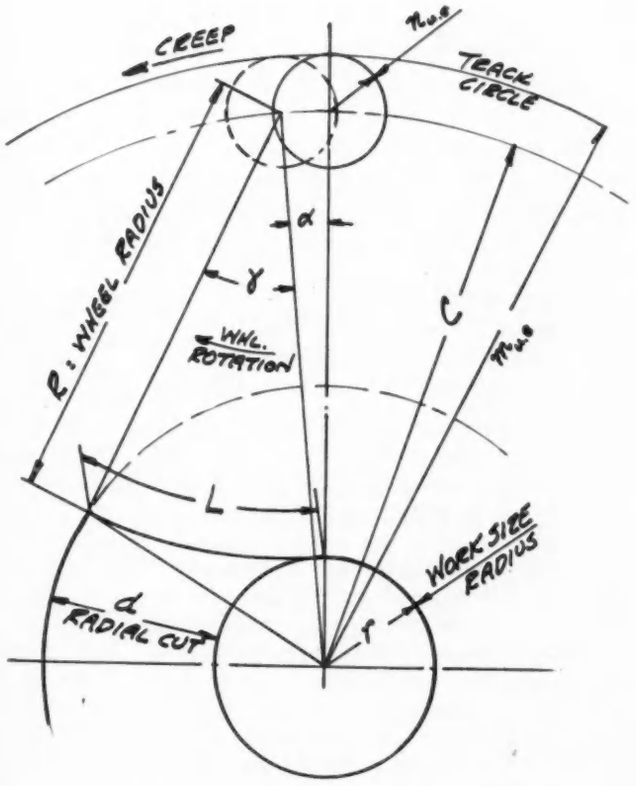
$$\approx \cos^{-1} \left\{1 - \frac{rd}{R(R+r)}\right\}$$

$$\approx \sin^{-1} \sqrt{1 - \left(1 - \frac{rd}{R(R+r)}\right)^2}$$

$$\approx \sin^{-1} \sqrt{\frac{2rd}{R(R+r)} - \frac{r^2d^2}{R^2(R+r)^2}}$$

The second term in the radical is small, so neglecting it and setting $\gamma = \sin \gamma$, for small angles we have

$$\gamma \approx \sqrt{\frac{2rd}{R(R+r)}}$$



UP-CUT (CONVENTIONAL) CASE +

$$L_c \approx \left(R \pm r\frac{N_a}{N_\gamma}\right)\sqrt{\frac{2Rd}{R(R+r)}}$$

Fig. 10—Diagram for Calculation of Interference Path

Hence approximately

$$L_{uc} \approx R\gamma + \frac{rn}{m} \cdot \gamma = \left(R + r\frac{N_a}{N_\gamma}\right) \cdot \gamma$$

$$\approx \left(R + r\frac{N_a}{N_\gamma}\right)\sqrt{\frac{2rd}{R(R+r)}} \quad (3)$$

But, for climb cutting, the rolling circle rolls on the outside of the track circle, which is of a new size, and angle α in the sketch shifts to the opposite side of the vertical, this changing expression (3) to:

$$L_{cc} \approx \left(R - r\frac{N_a}{N_\gamma}\right)\sqrt{\frac{2rd}{R(R+r)}} \quad (3)$$

and

$$\frac{L_{uc}}{L_{cc}} = \frac{N_\gamma R + N_a r}{N_\gamma R - N_a r} \quad (11)$$

Neglecting the term $\frac{rN_a}{N_\gamma}$ in the first factor of the right-hand side of expression (3), we have approximately the *ratio* of length of interference path to conformity factor K as proportional to

$$R\sqrt{\frac{r}{R(R+r)}} \div \frac{Rr}{R+r}, \text{ i.e. } \propto \frac{\sqrt{R+r}}{\sqrt{Rr}} \propto \frac{1}{\sqrt{K}}$$

The approximate average thickness of chip per grain may be estimated as follows:

Volume of metal removed per grain passage $\approx r \times a \times d$

$$\alpha = \frac{N_a}{N_\gamma} \cdot \gamma \quad \text{and} \quad \gamma \approx \sqrt{\frac{2rd}{R(R+r)}}$$

The approximate length of interference path is

$$\left(R \pm r\frac{N_a}{N_\gamma}\right)\sqrt{\frac{2rd}{R(R+r)}}$$

Hence average thickness of chip per grain

$$\left\{\frac{N_a}{N_\gamma}\sqrt{\frac{2rd}{R(R+r)}} \times d\right\} \div \left\{\left(R \pm r\frac{N_a}{N_\gamma}\right)\sqrt{\frac{2rd}{R(R+r)}}\right\}$$

$$= \frac{N_a}{N_\gamma} \times d \times \frac{N_\gamma}{RN_\gamma \pm rN_a}$$

$$= \frac{N_a}{RN_\gamma \pm rN_a} \times d \quad (6)$$

Treating the chip shape as approximately triangular, the maxima thicknesses are about double the average, and the increase of these dimensions for "climb" cut over up cut cases is

$$\left(\frac{RN_\gamma + rN_a}{RN_\gamma - rN_a} - 1\right) \times 100 \text{ per cent.}$$

Appendix D

Effect of Traverse in External Cylindrical Grinding

Let t = traverse per work revolution in in. Other nomenclature as before, see Fig. 11. The cutting points of the wheel now operate on helicoidal surfaces coaxial with the work instead of in planes normal to the work axis. Approximately the wheel may be imagined to cut in a series of planes inclined from normal to work and wheel axes by the angle

$$\Theta, \text{ where } \Theta = \tan^{-1}\frac{t}{2\pi r}$$

The wheel and work sections in such plane are elliptical, their minor axes colinear, and they contact on these axes.

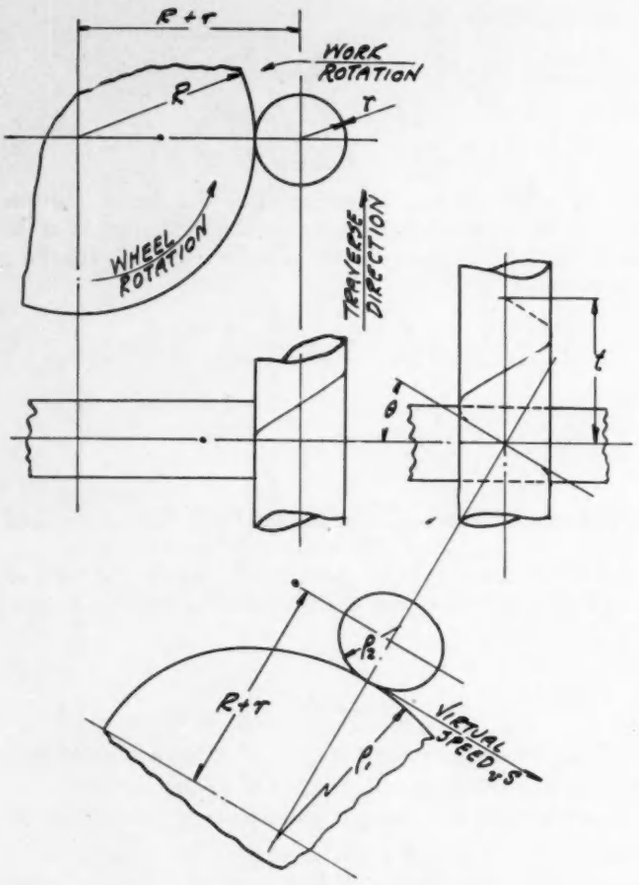


Fig. 11—External Grinding with Traverse

Their radii of curvature at the contact points may easily be shown to be

$$\rho_1 = R \sec^2 \Theta \quad \text{and} \quad \rho_2 = r \sec^2 \Theta$$

see Fig. 10.

Let ${}_vS$ = apparent wheel surface speed in the inclined plane.

Then ${}_vN_\gamma = \dfrac{6S \cos \Theta}{\pi R \sec^2 \Theta} = N_\gamma \cdot \cos^3 \Theta$

= apparent wheel r.p.m.

And ${}_vN_a$ = apparent work r.p.m. $= N_a \cdot \cos^3 \Theta$

Radial depth of cut, d, is unmodified.

Substituting ${}_vN_\gamma$, ${}_vN_a$, ρ_1, and ρ_2 in expression (3) we have

$${}_iL_e \approx \left(R \sec^2\Theta \pm r \sec^2\Theta \frac{N_a \cos^3\Theta}{N_\gamma \cos^3\Theta}\right) \sqrt{\frac{2rd \sec^2\Theta}{R \sec^2\Theta(R+r) \sec^2\Theta}}$$

which reduces to

$${}_iL_e \approx \sec \Theta \left(R \pm r \frac{N_a}{N_\gamma}\right) \sqrt{\frac{2rd}{R(R+r)}}$$

+ for "up" cut, − for climb cut and, since $\sec \Theta = \sqrt{1 + \tan^2 \Theta}$, this relationship means that the length of cut is increased by the presence of the traverse to $\dfrac{\sqrt{4\pi^2r^2 + t^2}}{2\pi r}$ times its value for the plunge-cut case.

The virtual conformity factor ${}_vK$ changes for

$${}_vK = \frac{\rho_1\rho_2}{\rho_1 + \rho_2} = \frac{Rr \sec^4 \Theta}{(R + r) \sec^2 \Theta}$$

$$\therefore \frac{K}{{}_vK} = \cos^2 \Theta = \frac{4\pi^2r^2}{4\pi^2r^2 + t^2}$$

That is, plunge-cut conformity is less than the conformity with traverse in the preceding ratio.

For the plunge-cut case:

$$\frac{L}{K} \text{ is proportional to } \frac{1}{\sqrt{K}}$$

But now ${}_vL$ becomes $L \sec \Theta$ and ${}_vK$ is $K \div \cos^2 \Theta$

So that $\dfrac{{}_vL}{{}_vK}$ is proportional to $\dfrac{\cos \Theta}{\sqrt{K}}$ and since $\cos \Theta$ can never be greater than 1 the conformity in this case increases faster than L increases.

Appendix E

Interrelation Between Work Speed and Traverse For Constant Length of Chip Path

Let $\pi\Delta$ = work circumference.
t = traverse per work revolution.

Then $L \propto \sqrt{\pi^2\Delta^2 + t^2}$ and L is constant

Let $\dfrac{t}{\Delta} = n$

$\therefore L \propto \sqrt{\pi^2\Delta^2 + n^2\Delta^2} \propto \Delta\,(\pi^2 + n^2)^{1/2}$

Calling L = unit length of chip path

$L(\pi^2 + n^2)^{-1/2} \propto \Delta$

and, since Δ in the actual case is constant, the effect of changing Δ to suit is made by changing its rotational speed, that is,

$L(\pi^2 + n^2)^{-1/2} \propto N_a$ where N_a = work r.p.m.

This plots as shown in Fig. 6.

Appendix F

Derivation for Plot of Relative Grain Path Length

$$L \approx \frac{R\sqrt{2rd}}{\sqrt{R} \times \sqrt{R + r}} \quad \text{and} \quad K = \frac{Rr}{R + r}$$

$$\frac{L}{K} \approx \frac{R\sqrt{2rd}}{\sqrt{R} \times \sqrt{R + r}} \times \frac{R + r}{Rr} \approx 2d \times \frac{1}{\sqrt{K}}$$

$$\therefore \quad L \propto \sqrt{K} \propto \sqrt{\frac{Rr}{R + r}}$$

Let $n = \dfrac{R}{r}$

Then $L \propto \sqrt{\dfrac{R^2/n}{R + R/n}} \propto \sqrt{\dfrac{R}{1 + n}}$

For convenience let R have unit value

Then $L \propto \sqrt{\dfrac{1}{1 + n}}$

and L is plotted versus n, that is versus $\dfrac{R}{r}$, as shown in Fig. 7.

Classification of Machine Tools

THE machine tools used currently in the automotive industry fall into three general types: special or single-purpose machines, unit-type machines, and standard or universal-type machines.

The first type is well known and has a strong hold on many unique operations in many plants. Its use today is just as basic to automotive production as ever before.

The second type finds wide adoption not only in mass-production plants but for job-lot work as well. This type of machine has been gaining steadily in importance due to its flexibility. Its versatility lies in the adaptability of a given machine to a variety of similar products or to complete change-over on a seasonal basis, simply by altering some of the basic heads or fixtures.

The third type is commonly used in all small-lot production and in tool rooms.

Excerpt from the paper: "An Appraisal of Current Progress in Automotive Manufacturing," by Joseph Geschelin, Automotive Industries, presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1938.

What Fleet Operators Should Know About Tires

By J. E. Hale

Manager, Development Department, The Firestone Tire & Rubber Co.

THIS paper is a non-technical review of an up-to-date survey of the lines of tires needed in all types and classes of fleet operation. To understand better how to get the best results from their operations, operators must know the proper type of tire to use.

The author first describes and catalogs the principle forms of tire failures, then reviews the characteristics of the fundamental lines of tires available at the present time.

Next, an attempt is made to classify the different types of fleet operation so that definite recommendations can be made as to the most appropriate tire equipment for these vehicles. The types of tire trouble most commonly encountered in each group are brought out with suggestions on how to avoid them. This part is followed by a section giving advice on the care of tires.

The paper concludes with a brief survey of worthwhile facts about repairs and retreading. An appendix contains the load-inflation tables which are most widely used.

DURING the past few years the automotive industry has put pneumatic tires to work in several new fields and has extended greatly the functions of tires in the older established operations. All kinds of freight are being transported faster and for longer distances. Passenger traffic is being speeded up and greatly extended. The latest developments have been in the moving of "tonnage" and "cubic-yardage" materials, such as earth, rock, coal, ore, and so on. The tire designer has labored steadily to increase the capacity of pneumatic tires so as to enable them to function efficiently in these new and difficult fields.

This paper is intended to be a matter-of-fact presentation of the present trend in tires as they pertain to fleet operations. The tire business is no longer simple. Each year it becomes more and more complex. As a consequence, the fleet operator, to get the most out of tires, has to specialize in his selection of tire equipment and must, too, understand the problems which develop in the use of tires in connection with his particular type of operation.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1938.]

Principal Lines of Tires

The tire manufacturers have not been able to design a universal tire that would give general all-around satisfaction in all classes of service. To cover the field effectively several different styles and types are offered. I will start by describing the features and characteristics of the most widely used fundamental lines. These lines group themselves into different tire classifications as follows:

Regular Passenger-Car Tires – Passenger-car tires are made in 4- and 6-ply construction and in several different grades but, for the purpose of this discussion, I will refer to the grade known as the first-line tire. This is the quality always found as original equipment on automobiles and commercial trucks. Passenger-car tires have been perfected to a point where, for average owner service, they may be expected to deliver from 16,000 to 20,000 miles of non-skid life and still be free from body ply failures. For the last few years most all cars have had adequate tire equipment but, in some cases, the pressures suggested are too low. The present tendency is toward tread designs of all-rib type, and of a rather flat profile. This design is used to satisfy the car manufacturers' insistence for quiet running, even though there is a slight loss of traction and non-skid ability. The public, however, is still very much safety-minded, consequently there continues to be a demand for tread designs which have more actual and visual non-skid properties.

Regular Truck-Bus Tires – These tires are divided into two types – balloon and high-pressure. (The pressures used in the balloon type are about two-thirds of those used in the high-pressure tires.) Either of these lines ordinarily may be expected to have averaged from 35,000 to 50,000 miles when they are worn out. These tires must deliver this service either in city or inter-city transportation work, and they must be rugged enough to withstand reasonable neglect and abuse, and also occasional overloading and occasional short-distance high-speed operation.

The tread is designed primarily for pavement operation to give the highest composite effect of long wear, traction, and non-skid on wet and dry pavements with no tendency to separate or develop sufficient heat to break down the tire body. To combat the tendency to uneven wear from "tread-wiping," commonly observed on front wheels, due to im-

proper alignment and underinflation and sometimes due to sagging rear axles on dual trailer wheels, there is a tendency to use narrower, flatter treads with more circumferential ribs. The longest-wearing designs have all the traction non-skid elements connected to the circumferential ribs since isolated non-skid units tend to wipe away.

The body structure of the tire uses from 6 to 16 plies according to the section size and air pressure recommended to carry the load. Beads, reinforcements, extra gum plies, gum strips, high-grade cord, graduated count of the cord, and so on, are all very important details of the specifications necessary to produce a balanced body construction that will not fail prematurely from blowout, separation, or bead trouble. These tires are used on flat-base truck rims in rim diameters of 17 in. and larger.

Experience indicates that the use of the high-pressure tire should be discontinued in favor of the lower-pressure balloons. Of course, I do not mean that we can make this move over night, but the move can be brought about gradually by equipping new trucks with balloon tires. The obvious advantages of the lower pressure sum up in:

(1) Reduced maintenance.
(2) Protection to fragile goods.
(3) Faster average speed.
(4) Easier on the driver.
(5) Desirable on rough streets.
(6) Less tendency to sink in soft soils.
(7) Better all-around service.
(8) Tires run cooler.
(9) Better equalization of load on duals.
(10) Longer tread wear.
(11) Less destructive to pavements.
(12) Better traction non-skid from larger area of contact.

Commercial-Delivery Tires – During the past year commercial-delivery truck developments in the ¾-ton and 1-ton classes have encouraged the introduction of a line of tires to fit 15-in. and 16-in. diameter rims. Generally speaking, these tires fit in the gap between passenger-car tires and truck-bus tires. The tires are made in 6.00 to 7.50 cross-sections of truck tire characteristics and have tapered beads designed to fit the "shallow drop-base" rims. This line of tires is intended for use in place of passenger-car sizes where a more rugged tire is needed.

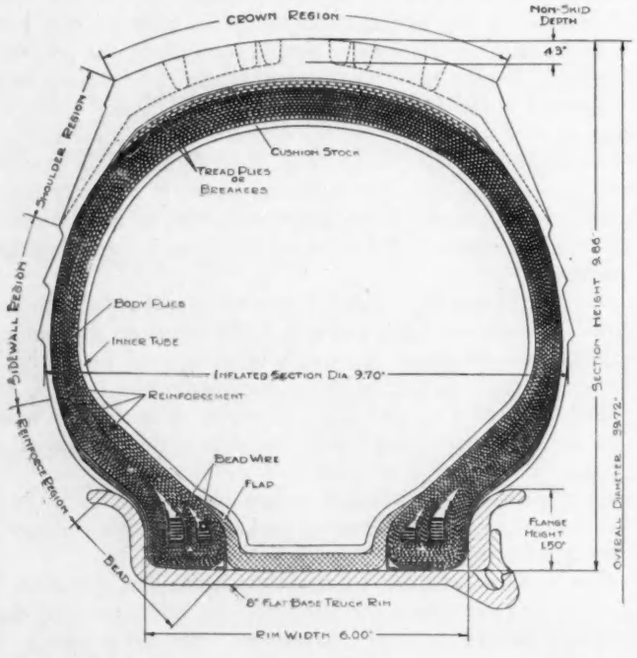


Fig. 1 – Cross-Section of 9.00-20-10-Ply Truck-Bus Tire, Showing Tire, Tube, Flap, and Rim with Principal Structural Elements and Points of Measurement

Inter-City Bus Tires – Although, in size and external characteristics, these tires generally resemble the regular truck-bus tires, their internal structure and the materials used in making them are highly specialized.

The features of inter-city bus operation that are difficult to combat center around the high degree of heat developed throughout the body of the casing from the high average speed and the long runs without stops. Of course the really tough conditions develop in summer, particularly in those parts of the country where the atmospheric temperatures average the highest. Design and construction details are all directed to the use of materials that will develop less heat and possess the ability to withstand it. Recently rayon cord has come into the market for tires in this class, to resist heat.

Mud and Snow Tires – Until the mud and snow lines were put on the market, chains had to be used on the regular tires, and they were not any too effective, to say nothing of being expensive and troublesome to put on.

These tires are made in the complete range of passenger-car sizes and truck-bus balloon and high-pressure sizes. In addition to having sturdy body construction, their treads are designed so as to deliver a maximum of traction in "off-the-road" service, on unimproved muddy roads or in deep snow. Traction bars radiating from the center of the tread at a 45-deg. angle, deep cut and with wide spacing between the bars, characterize these designs. The all-important feature is to have the design "clean" in muddy operations. Those patterns that are designed most effectively will provide traction to take the truck or passenger car through any kind of mud road or field that will offer enough support to keep the axles and motor pan off the ground.

The mud-and-snow type tires are not suitable for continuous use on pavements. The traction bars generally wear unevenly, to an extent that makes them noisy, and often they are rough enough to cause the car or truck to vibrate. The tires have arrows on their sidewalls indicating the direction of rotation that minimizes the wiping tendency. In those cases where the operators find it necessary to travel a certain percentage of the time on hard roads, the life of the mud and snow tires varies considerably. Being used under such varied operating conditions, they are more likely than other types to be subject to abuse and neglect to a degree that affects their life.

Stop and Start Tires – This type of tire has a larger volume of tread rubber in proportion to the rest of the tire than has any other line. Tires of this type were introduced for city deliveries because the frequent repetition of brake application to stop and power application to accelerate was responsible for fast tread wear. This difficulty could be corrected only by designing a tire having approximately double the tread volume of the so-called "regular tire." Tires of this character cannot be overloaded nor can they be operated at high speeds. They are made in sizes for small delivery trucks, passenger cars, and taxicabs.

Rock-Resisting Truck Tires – Regular truck-bus tires, when used in heavy hauling on poor roads, construction jobs, contracting operations, rocky roadways and the like, will be cut and snagged to a degree that will put them out of service long before they have delivered reasonable service, so we have brought out this line.

The characteristics of these tires are: first, extra wide and heavy treads with massive non-skid elements made of "cut-resisting" rubber and, second, heavy sidewalls to stand the abuse of rocky and uneven road operations. They are made



Fig. 2 – Truck-Bus Tire That Has Worn Through the Tread to the Light-Colored "Cushion" Stock



Fig. 3 – Passenger-Car Tire Worn to the Point Where the Non-Skid Design Is About Gone



Fig. 4 – This Truck-Bus Tire Had Its Tread and Sidewall Cut and Snagged from Rough Roadways; Consequently It Did Not have a Chance To Wear Out by Normal Wear



Fig. 5 – Showing the Tread "Separated" from the Cord Body When the Tread Is Hardly One-Half Worn

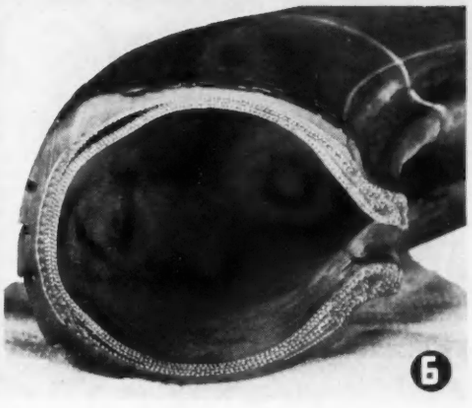


Fig. 6 – An Example of "Ply Separation" – A Passenger-Car Tire that Ran Soft at High Speed for a Long Distance, Thus Developing Too Much Heat

in truck balloon sizes. Also, they are made commonly with "extra-ply" body construction.

Earth Moving Tires – These tires are always used on trailers or their equivalents, on "off-the-road" operations. Consequently, they are free-wheeling, have only a moderately prominent tread pattern and are generally made in a size large enough to be operated at pressures low enough to assure flotation (25 to 35 lb. per sq. in.).

Road Grader Tires – These tires are of two classes: high-pressure (80 to 100 lb. per sq. in.) used on flat-base truck rims and low-pressure (25 to 30 lb. per sq. in.) used on tapered-bead seat rims. The latter type of rim is necessary with low-pressure tires to prevent the tire from creeping on the rim. In either case the tread design is aggressive (similar to the mud and snow design) to provide traction, and the body construction of the tire is such as to resist injury by rough road conditions as encountered in some types of road construction work. The road grader front tires have prominent circumferential ribs which have been found necessary to facilitate steering control.

Classification of Tire Failures

Inasmuch as I constantly refer to the various forms of tire troubles in what follows, it seemed advisable to illustrate them at this point in the paper. The shop names or terms that we tire people use are certainly not always self-descriptive. Furthermore, I doubt if many operators ever really have a clear understanding of the things that happen to tires to put them out of service.

First, I will introduce an outline drawing, Fig. 1, showing the cross-section of a pneumatic truck tire, tube, and flap on a rim with the principal regions of the tire named. I will also give a brief reference to the principal effects observed in tires that have completed their service.

The term "treadwear" generally is used to express rubber actually worn away by normal wear and not wear by cutting, chipping and snagging. See Fig. 2. In referring to treadwear on passenger-car tires one generally has in mind the points at which the tread has lost its non-skid design or gone smooth. See Fig. 3. On the other hand, with respect to truck tires, the worn-out point almost always refers to the wear being carried down to the breakers or tread plies as shown in Fig. 4.

"Tread separations" or "ply separations" are somewhat colloquial terms indicating very destructive forms of tire trouble. Pneumatic casings are designed and manufactured with each structural and wearing element an integral part of the whole and are so vulcanized. Furthermore, if the casing is to function as a tire, it must remain so during its operating life. If it happens that the casing in service is subjected repeatedly to stresses beyond the natural strength of the materials, particularly through shear and the weakening effect of heat, then these forces may, by destroying the binding qualities of the rubber stocks, break this unit construction and the tread will "separate" from the tire body as shown in Fig. 5, or there will be a separation between cord plies of the body itself as illustrated in Fig. 6.

"Cord body breaks," also serious forms of premature failure, show up in any of several different forms. When they occur, the tire is definitely finished regardless of the state of wear of the tread. They cannot be repaired but, by proper

treatment of the tire, these forms of tire trouble may be prevented. They are not the result of manufacturing defects but the consequence of operating conditions. Figs. 7, 8, 9, and 10 show examples of the different characteristic failures. We recognize flex breaks in the shoulder (Fig. 7), mid-sidewall (Fig. 8), and reinforce region. The term "flex" in this connection is fairly self-explanatory; that is, the flexing of the cord body is so excessive that the cords break by flexing. Impact or concussion breaks occur in three forms, one long diagonal break (Fig. 9), an L-shaped break, and an X-shaped break (Fig. 10). These breaks, the result of too much stress on the cord body as a whole, are due generally to striking some obstacle which indents excessively into the tire when it is overheated with the result that the tire bursts or blows out through all plies.

Rim bruises are a form of abuse common in the 4-ply passenger-car tires and occasionally seen in 6-ply tires. They are really concussion breaks resulting from an underinflated tire bumping over a curb or rock and in such a manner that the casing flattens out crushing the sidewall between the rim flange and the curb as shown in Figs. 11 and 12.

There is an infrequent type of failure in which the tire beads may fail from external chafing at the top of the rim flange. This chafing happens if the tire is not used on the proper rim or if the rim flange has been battered down. See Figs. 13 and 14. Sometimes the beads fail from too much internal action caused by severe overloading or underinflation, particularly in use upon rough or winding highways which develop stresses in the plies around the bead wire sufficient to loosen the adhesion.

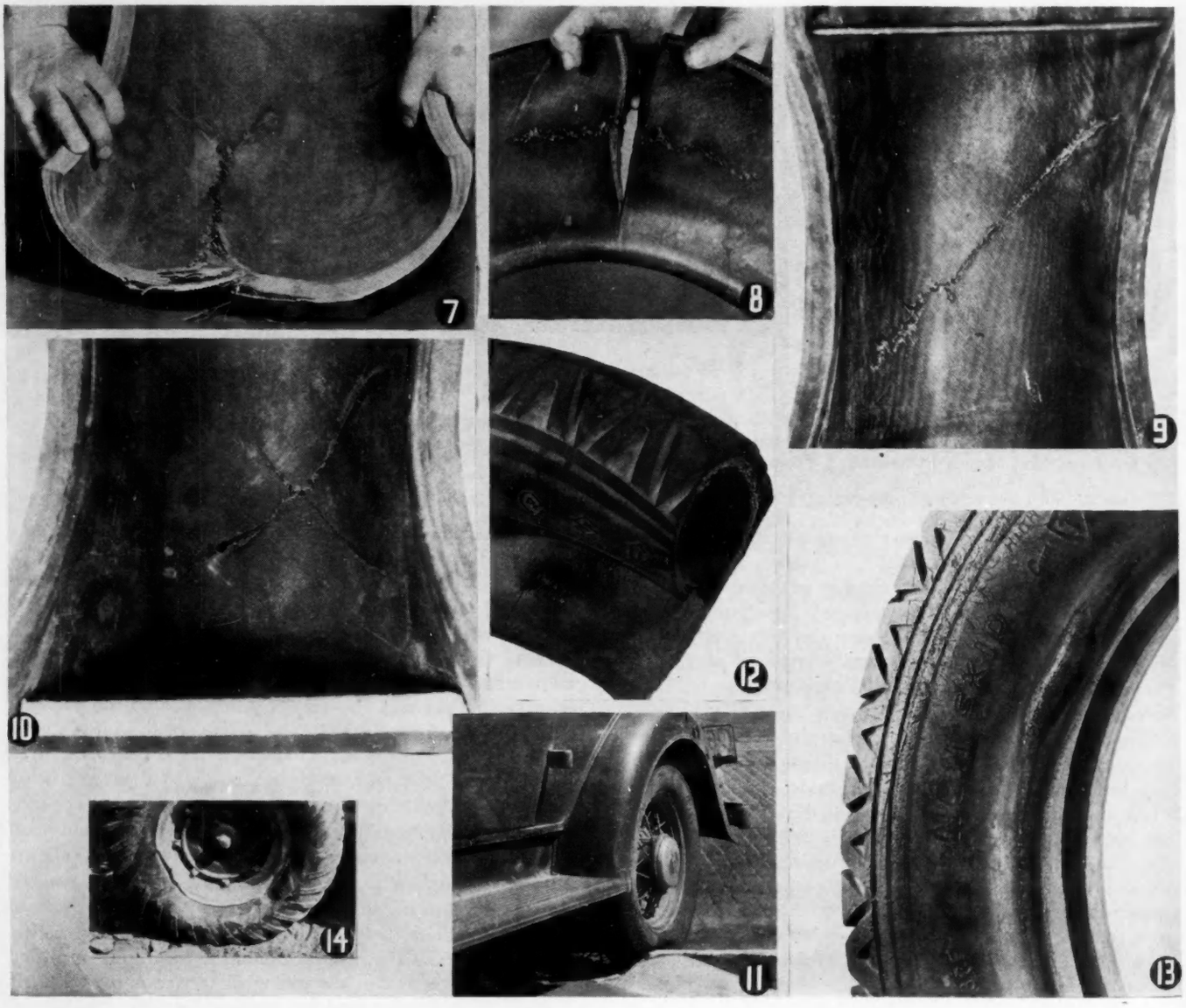


Fig. 7 – A "Shoulder Break" Through All Cords – High Sustained Speed in Hot Weather – The Tread Probably "Separated" First Which Added to the Heat at This Spot

Fig. 8 – A Mid-Sidewall Flex Break – Note the Cords Broken and Frayed by Too Excessive Flexing Due to Overloading or Underinflation

Fig. 9 – "Diagonal Impact Break," the Result of Hitting a Hole or Ridge in the Highway

Fig. 10 – An "Impact Break" in the Form of the Letter X – The Tire Burst when It Hit Some Protruding Object in the Roadway

Fig. 11 – Showing How an Under-Inflated Tire Can Be Doubled Up and Crushed between the Corner of a Curb and the Rim

Fig. 12 – "Rim Bruise" Resulting from the Action Shown in Fig. 11

Fig. 13 – Truck-Bus Tire Prematurely Removed from Service Because Rim Flange Was Battered Down at One Place Causing Bead To Chafe Through

Fig. 14 – An Example of a Battered Rim Flange on Road-Grader Operation

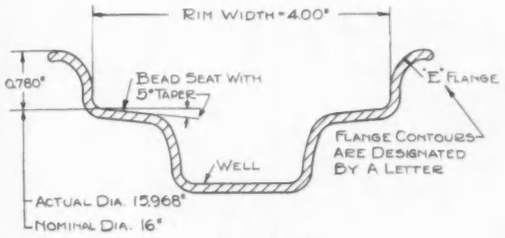


Fig. 15

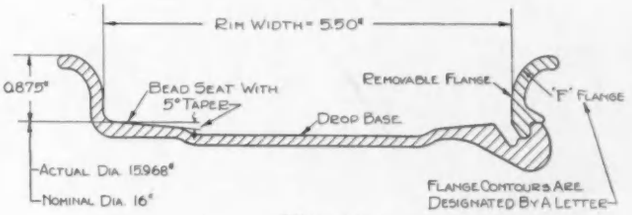


Fig. 16

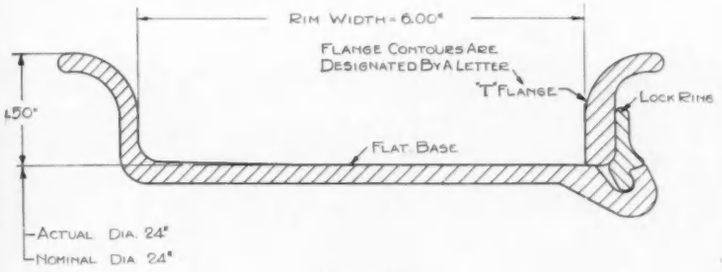


Fig. 17

Fig. 15 – Passenger-Car Full Drop-Center Rim – 16 x 4.00-E

Fig. 16 – Two-Piece Commercial-Delivery Shallow Drop-Base Rim – 16 x 5.50-F

Fig. 17 – Flat-Base Truck Rim – 24 x 8 (6.00-T)

Fig. 18 – Four-Piece Road-Grader and Agricultural Tractor Shallow Drop-Base Rim – 24 x 8.00-T

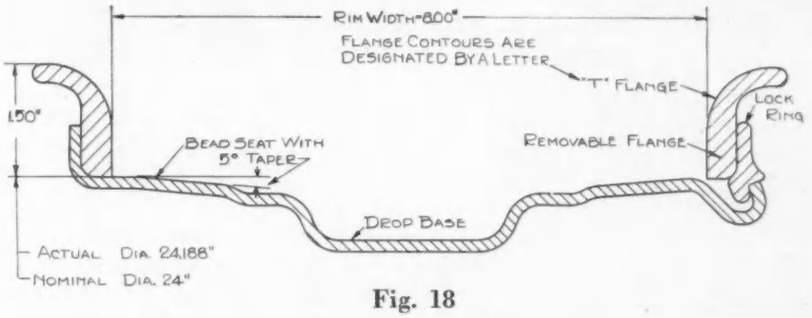


Fig. 18

Principal Lines of Rims

A complete paper could be written on the subject of passenger-car and truck-bus rims. However, I will simply list and illustrate in Figs. 15, 16, 17 and 18, the more important basic types in the nature of an accompaniment to the tire descriptions.

Vehicle Classification by Operations

One objective of this paper is an attempt to make recommendations as to the preferred type of tire for the different classes of operation which might serve as a guide in choosing tire equipment. To cover this subject completely would require too much detail for a paper of this length but, by classifying similar operations in suitable groups, it seemed possible to give a fairly comprehensive line-up. Allowance must be made for cases where the classifications overlap because obviously there are many borderline examples.

(A) *City Operations – Heavy Freight and Passengers* – In this group are listed city freight, city buses, wholesalers, city coal deliveries, meat and provision deliveries, building materials, beverages, moving vans, and so on. Characteristics of these operations are heavy hauling with medium- and large-size units, both trucks and truck-tractor-trailer units, practically all on paved or improved streets. Fig. 19 shows an example of this group.

The preferred tire recommendation for vehicles in this group is the regular truck-bus balloon. For replacements on existing vehicles already equipped with regular high-pressure truck-bus tires, the latter type is logical but it is not recommended for new jobs.

Overloading, under-tiring, neglect of inflation, bumping curbs, injuries from bad pavements, and litter in the streets, are the prevailing forms of abuse or neglect that damage the tires. The driver can do a lot to favor the tires and the truck dispatcher can too, if they will refrain from overloading.

Fast treadwear is, of course, principally the result of frequent brake and power applications from starting and stopping. Overload and underinflation also accentuate treadwear.

(B) *City Operation – Commercial Deliveries* – In this group I include what might be put in the class of commercial-deliveries service, such as department stores, bakeries, newspapers, groceries, plumbers, laundries, small coal trucks, milk, grain, and feed, hardware, express, and various forms of construction and repair service. Fig. 20 shows an example of this group. A large percentage of these operations is in city deliveries and practically all of them are on city pavements which normally should allow the tires to wear out naturally. Any shortcomings from this service could be attributed largely to the start-stop nature of the operation, but there is obviously a fair amount of neglect of inflation pressure which is also responsible for shortening the tire's life. The smallest of these vehicles generally has a passenger-car chassis and should be equipped with 4- or 6-ply passenger-car tires or start-stop tires. The larger vehicles of this group (¾-ton – 1-ton – 1½-ton trucks) should be equipped with either stop-start tires, commercial-delivery tires, or the regular truck-bus tires of the smaller sizes.

The tire problem throughout this group is neglect of inflation which leads to early treadwear and rim bruises. Backing into curbs, clipping curbs at street corners, bumping over





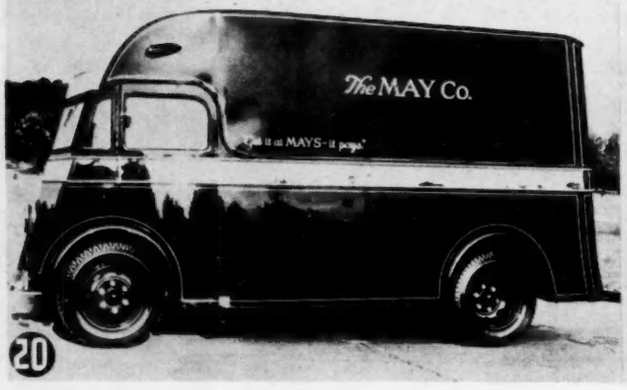




Fig. 19 – Example of City Operations – Heavy Freight and Passenger Group

Fig. 20 – Example of City Operations – Commercial Deliveries Group

Fig. 21 – Example of Inter-City Operation – Heavy Freight and Passenger Group

Fig. 22 – Example of Passenger Cars Used in City and Inter-City Passenger-Car Group

holes in pavements, these abuses combined with pressures low enough to let the rim strike through, and the result is that all cords are crushed and broken. Overloading also will be found to be common with the operations of this group that deliver the heavier classes of goods and merchandise.

(C) *Inter-City Operation – Heavy Freight and Passengers* – In this group I have put inter-city motor freight, inter-city buses, automobile transport units, oil tankers, and so on. An example is shown in Fig. 21.

The features which characterize this group are heavy hauling for long distances over good highways. Most operators aspire to high sustained speeds, which overheat the tires. Overloading is also more or less prevalent except with buses, the result being that body ply failures (both impact and flex break) are the most annoying type of premature tire trouble. For inter-city freight where speeds are only about two-thirds those of the bus lines, the regular truck-bus tire in either balloon or high-pressure type has been found satisfactory. But, where the motor-truck speeds are the same as those of the inter-city buses, the special heat-resisting bus tires are preferable.

Some operations between cities include considerable stretches of gravel roads, a class of operation which is particularly hard on tires. In addition to the crown and flex breaks mentioned previously, we must add impact breaks and severe tendencies to tread-cracking due to cuts which grow into large cracks on gravel roads.

(D) *City and Inter-City Passenger-Car Use* – This group I have reserved for the passenger cars used by salesmen in covering their territory. In most cases these are small popular-priced cars as shown in Fig. 22. The most appropriate tire equipment is to duplicate first-line passenger-car tires provided as original equipment. In most cases they are operated over good pavements or at least hard-surfaced roads. There is probably an occasional case where the mud-and-snow type tire would be recommended.

Neglect of inflation, clipping curbs, hard fast driving, and violent application of brakes, (the human-element factor) are the abuses to watch in this group: the result, rim bruises and fast treadwear.

(E) *Company or Contract Operations – Extra Heavy Trucking* – The classes of fleet operations included in this group are coal strip mining, quarrying, lumbering and logging, ore mining, oil-field operations, and city contracting. The predominating characteristics of these operations are heavy hauling, short or medium distances, large-capacity units of trucks, and truck-tractors with semi-trailers, generally operating over poor roadways for at least a part of their operation. Figs. 23 and 24 show examples.

Such work calls for tires that can stand the abuse sustained in rolling over bad roads, holes, uneven places in the roadway, and the injuries due to generally unfavorable operating conditions.

Traction is seldom an acute problem in this type of service. Overloading is also quite common in operations of this group and, in combination with rocks, holes, or stumps in the roadway, it will produce impact breaks in the cord body of the tires. As the overloaded tire rolls over these rocks, holes or uneven spots, the object indents the tire too far, the tension on the cords is carried beyond their strength and there is a burst through all plies. Lower air pressure, by putting less tension on the cords, reduces or eliminates these impact breaks, but lowering the pressure may be resorted to only after a thorough check by a qualified tire man to determine whether the loads carried will permit such a move without inviting flex breaks.

The tires seldom wear out through normal use. They are often cut and chewed to pieces before they are worn down to the base of the non-skid. The cuts, gradually getting worse, penetrate into the tire body, allowing water, sand, and gravel to enter the openings causing body ply injuries which put the tire out of commission.

The type of tire most generally suitable for this class of operation is the "rock-resisting" type. There are certain exceptions, however. In some operations, notably in oil fields and lumbering where maximum of traction is needed, the mud-and-snow type should be used.

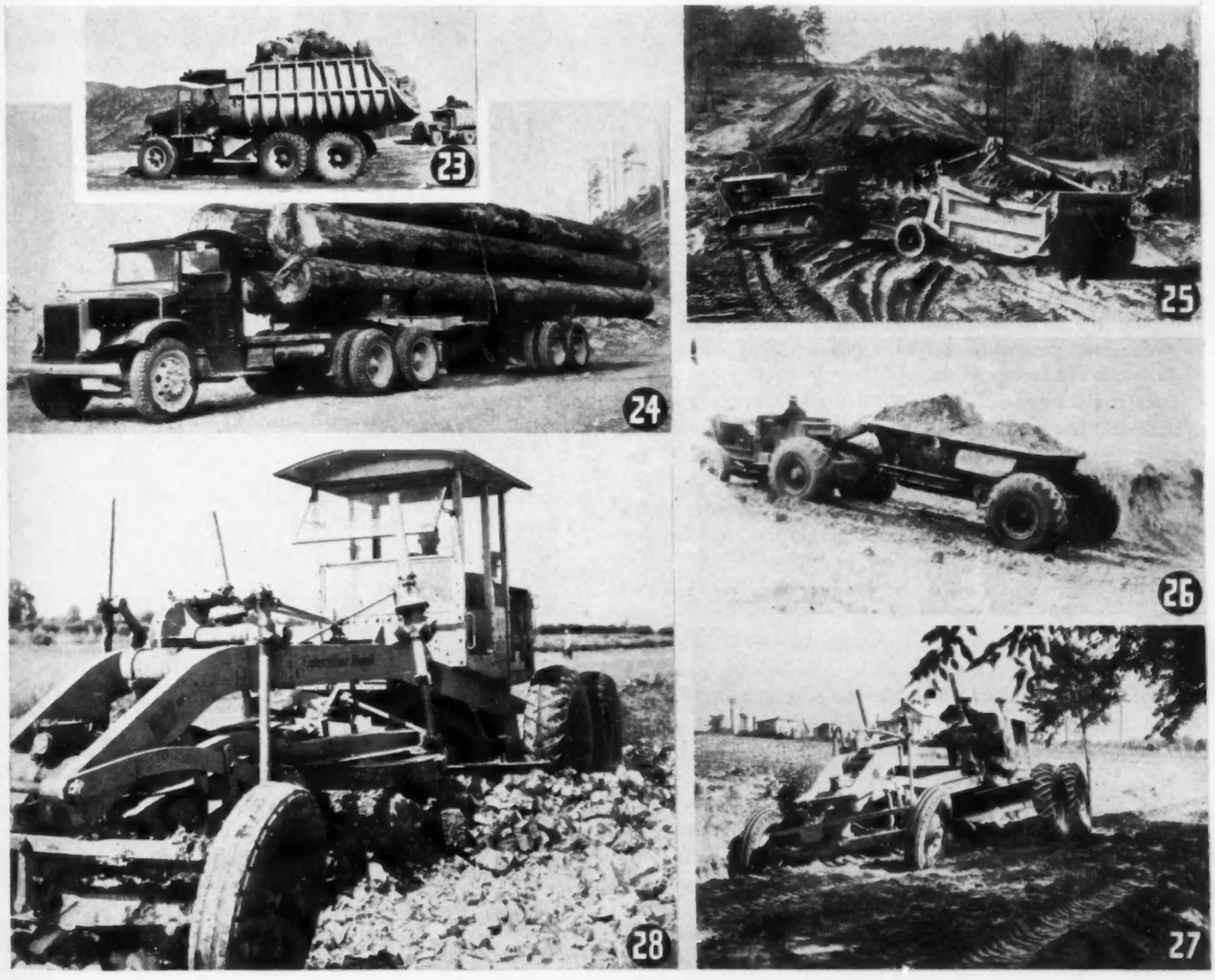
Quite often we find operations that are unfavorable even to the "rock-resisting" tires, that is, where loads are excessive, the roadways particularly bad, or the vehicle restricted in a manner to prohibit oversizing. For these cases we suggest "extra-ply" tires of the same "rock-resisting" type. A study of the operation has to be made to determine the most suitable air pressure to make this move successful.

(F) *Earth-Hauling Operations* – In this group I have classified the earth-moving vehicles which include scrapers and tractor-trailer wagons. These operations are always on new construction jobs, involving the moving of earth and rock and, consequently, operating over very unfavorable ground surfaces. Two of these vehicles are shown in Figs. 25 and 26. One important problem is to provide flotation. This factor requires low air pressure ranging from 25 to 40 lb. per sq. in. In this service the free-wheeling tires should be of the "earth-mover" type and, on those units employing a wheel tractor, the driving wheels should be equipped with the "rock-resisting" tires in rocky operations and the mud-and-snow type in conditions requiring a maximum of traction.

Although the speeds are low, the loads are heavy. Since most of these vehicles have no springs, failures by impact breaks of the body plies from rolling over rocks, stumps, pot holes and uneven roadways are, by far, the most common form of tire trouble. Cutting and snagging of the tread and sidewalls are also common in rocky operations.

In operations of this character there are two features that must receive more consideration: First, in order not to overload the tires, the weight per cubic yard of the material being hauled must be ascertained and the yardage moved in each load governed accordingly. Second, if the operation is over rocky ground, the yardage carried must be reduced by one-third to one-half of what would be technically allowable with dirt, sand, or gravel, and the air pressure must be reduced by the same amount. Even with these precautions it will pay the contractor to put on extra men and equipment to keep the runways free from rocks.

Another thing that we learned is that large singles are preferable to duals in rocky operations. It is a case of momentary double load for one of the two tires of a dual combination as it rolls over the rocks.



Figs. 23 and 24 – Examples of Company or Contract Operations – Extra-Heavy Trucking Group
Figs. 25 and 26 – Two Types of Vehicles Used for Earth-Hauling Operations
Figs. 27 and 28 – Road-Graders Showing Types of Tires Used

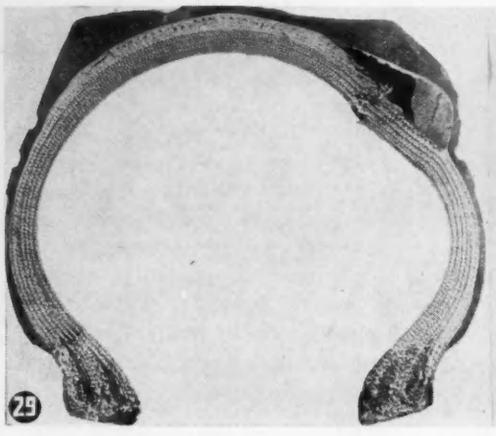


Fig. 29 – "Tread Separation" along the Shoulder – The Loose Tread Rubbing Against the Cord Body Ground Up the "Cushion" Stock into a Soft Bulky Mass (Called "Cushion-Bunching") – This Action Developed Enough Heat in the Cord Body so that It Blew Out Through All Cords

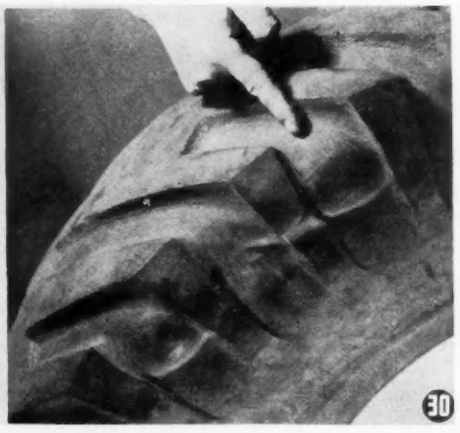


Fig. 30 – How "Cushion-Bunching" Looks from the Outside – Whenever this Bunching Happens the Tire Should Be Removed from Service Before It Blows Out

We found some cases of tractor-trailer wagon operation in which it was possible to get up considerable speed over uneven and rocky runways. Of course, this combination just popped the tires open.

(G) *Road Grader Operations* – In this group I place road graders. These vehicles are used particularly on gravel roads and in road-construction work, generally under the authority of public highway departments. The loads are heavy, the road surfaces often unfavorable, and the operation calls for maximum traction. Also most road graders have no differential action between the driving wheels – whether there be two or four driving wheels. Two of these machines are shown in Figs. 27 and 28.

Up until a year and a half ago, high-pressure (80 to 100 lb. per sq. in.) tires were considered necessary for road graders because lower pressures resulted in bouncing of the grader with uneven "washboard" grading as a result. We are now able to use low-pressure (25 to 30 lb. per sq. in.) tires with more aggressive traction tread designs. The result is that road graders now have tractive ability to do road-construction work and more effective snow plowing as well as road maintenance – just another automotive advancement contributed by low-pressure tractive tires.

Tire troubles on road graders manifest themselves mostly as the result of excessive traction scrubbing the tread off. The rougher the operation, the more this type of wear takes the aspect of small, medium, or large cuts, or snags. Front tires on road graders must be designed especially of very heavy ribs to provide effective steering. Some graders have leaning wheels to prevent slipping sideways, as well.

In road-construction work the road graders are handled more or less like a gigantic plow. The blade is set to gouge out the soil or rocks from the banks at the side of the road. It often happens that the grader blade is set in such a position that the discharge from the blade leaves a stream of rocks right in front of the rear tires so that service on these tires is particularly severe and the liability of trouble from impact breaks is excessive.

Choose Adequate Tire Equipment

Fleet owners should realize that, to get economical and trouble-free tire service, their efforts should start with the selection of the proper size and type of tires as original equipment on all new units. Let the buyer be frank with the truck salesman, describing to him precisely the service in which the equipment is to operate, the loads to be carried, the roads to be traveled, the speeds, the nature of the material to be hauled, and so on. The vehicle salesman will, if he wants to have a satisfied customer, discuss the optional tire equipment listed for the job in question and urge the buyer to specify a tire and rim size adequate to do the job.

It is poor economy to skimp on a tire and rim size when purchasing new vehicles. Sooner or later the correct size will have to be installed. Rim sizes on passenger cars, as the cars are built at present, are universally adequate and satisfactory to support properly the beads of the tires. The situation on trucks is generally satisfactory but, occasionally, in selecting tire and rim optional equipment for new trucks, a mistake may be made by choosing an undersized rim whose flange may be just low enough to permit bead chafing. Our industry standards (see Table 1, a reproduction of page 35 of the 1937 Tire & Rim Association Hand Book in the Appendix to this paper) show a "recommended original equipment" rim size (and center spacing) for the truck manufacturer to use when he delivers the truck. In addition, the table shows a suggested "consumer oversize" tire that may be used by the truck operator on this rim if, later, he finds the oversize tire necessary to get satisfactory service. If the rims are too much undersized, particularly in rough heavy-trucking service, there may be actual rim breakage. The drop-base and drop-center rims used on small trucks should be according to the industry recommendations to make sure the bead seat of the rim is of proper width to support the bead without tube chafing.

Never forget that tires are articles that either wear out or fail prematurely.

To avoid the confusion which would arise from claims of competitive makes, the tire industry has agreed (through the medium of the Tire & Rim Association) upon a few technical fundamentals which are common to all makes. These fundamentals are:

(1) The system of name sizes is standardized.

(2) The rims on which the tires are mounted are standardized, thus providing complete interchangeability between makes on a given rim.

(3) The cross-section measurement of tires of a given name size are standardized.

(4) In the case of dual tires the center spacing is standardized.

(5) The load limits and inflation pressure recommendations are standardized.

The automotive industry simply could not function without all these standards; yet they leave plenty of latitude for the various tire companies to display individuality in the appearance and quality of their respective lines. Also these standards are designed so that individual tire companies have full and ample opportunity to pursue their efforts to develop and improve their own products unhindered by these standards.

Our various types of tires require at least an equal number of load-inflation tables. Inasmuch as these tables are of such

prime importance and, I find, are often discounted or the wrong one used, it seems worth while to list the complete tables in the Appendix of this paper (Tables 2 to 10). These tables all show a definite load maximum or limit for each tire size with a corresponding inflation pressure recommendation. Note that each table has a caption describing fully the use of that particular table. These tables are significant in three ways:

(1) They indicate the condition of operating load at which the tire may be expected to give the most economical service and wear out without premature failure.

(2) The air-pressure recommendations are of equal importance to the load limit since tire technicians know that either underinflation or overinflation affects tire durability adversely.

(3) They serve as a technical basis for deciding adjustments in the event of claims for unsatisfactory service.

Advice on the Care of Tires

"Your tires are too soft . . . this casing looks as though it had been overloaded . . . we will adjust this tire provided you will put on the proper size to carry the load . . . our inspection shows underinflation . . . suggest you insist that your drivers check air pressure once a week regularly . . . this casing blew out because it was driven at high speed over some obstruction in the highway . . . this is a case of tread separation caused by a combination of overinflation and overload."

The tire manufacturers have been using just these phrases and a hundred others of the same import for many years. Papers and articles – dozens each year – have been promulgated by the tire companies in an earnest effort to educate the tire user to his own advantage. Only a couple of years ago I myself wrote a lengthy paper on the single subject of overloading truck tires. Here I am writing another paper and I find myself trying to think of new and better ways to drive home the same thoughts. I think of new tests, new photographs, and curves – new tabulations, of how easy it is to prove these things, and how difficult to convince the operator! I really have nothing new on the care of tires, so I list as briefly as possible a rehash of the salient points:

Effect of Heat and Seasons – Rubber being a poor conductor of heat and also physically under the handicap of rather high hysteresis when subjected to flexing, tires gradually build up heat through internal friction to a substantial degree when subjected to the simple action of flexing as the tire rolls along. The thicker the tire, the greater the heat that is developed, and the slower it is in dissipating it.

On the other hand, the thin 4- and 6-ply tires, although they do not have so much tire body to hold heat, also develop a high degree of heat because they are used on vehicles capable of high speed which generates the heat so rapidly that it does not dissipate it effectively by windage.

The high sustained speed of inter-city bus operation raises the heat level in the tires to a degree greater than in any other form of service. This excessive tire heat has two very definite effects tending to take mileage out of the tire: First, the tread wears away much faster when in a heated condition. Second, the cord body is much more liable to premature failure. The effect of heat on the cord is to lower the tensile strength, and repeated heatings combined with flexing gradually deteriorate the cord. The result is that the tire finally blows out through a circumferential flex break in the shoulder which we ordinarily call a "crown flex break," or the tire may burst through all plies when it rolls over an obstacle or hole in the road which results in a blow in the nature of an impact. These effects naturally are more pronounced in summer and in hot climates.

Heat also accentuates tread and ply separation. The rubber stocks in a heated condition lose some of their physical characteristics when stresses of the sort causing shear action occur, the result is "separation" as shown in Figs. 29 and 30.

Treadwear is much faster under hot weather conditions than in cold weather. The rate of wear may be up to double that of cold weather.

Fig. 31 shows summer mileage compared with winter mileage in bus operation. The curve shows quite vividly the extent to which hot-weather operation reduces tire life. Tires worn out and tires going out of service prematurely from all causes are included in the data used to draw the curve.

All vehicles should be constructed to provide the best possible wheel-housing ventilation, and the relationship between the rim and the brake drum should be such that there will be maximum ventilation to keep brake heat from building up too much around the beads of the tires.

The degree to which tires heat up can be visualized in the pressure and temperature increase of the air in the inner-tube. The following are examples:

Size	Speed, m.p.h.	Air Temperature in Tube Increase, deg. fahr.	Air Pressure in Tube Increase, lb. per sq. in.
7.00-16 4-Ply	67	84	5
7.50-16 6-Ply	67	92	7
9.75-20 12-Ply	50	155	25
11.25-18 12-Ply	50	138	19

However, regardless of the pressure increase, the driver of the vehicle should not under any circumstances let air out of the inner-tube or "bleed" the pressure as it is sometimes called. All tire-company pressure recommendations are intended to be taken "cold," that is, when the temperature of the tire is at approximate equilibrium with the prevailing atmospheric temperature.

As a means of further illustrating the heat and pressure build-up the set of curves in Fig. 32 is included.

Heat has a deteriorating effect on the inner-tubes. After

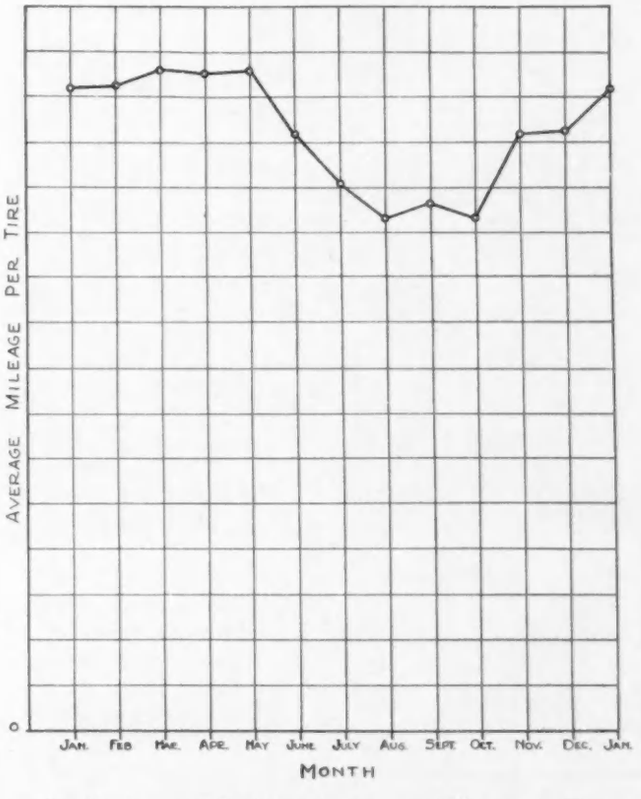


Fig. 31 – Seasonal Variation in Average Mileage per Tire – Inter-City Buses

many months of repeated heat runs the stock loses its character, becomes lifeless and flabby, and takes a permanent set in a stretched-out condition so that it is almost impossible to put the tube back into the casing without buckles or folds once it is taken out. It is poor economy to put old tubes into new casings.

Recently rayon has come into the picture for tires intended for use under heat conditions. This application is because rayon does not lose tensile strength when heated, to the extent that cotton does. Rayon is still expensive enough to justify an increased cost of rayon-cord tires over tires made with cotton. However, it is likely that sooner or later this expense will be reduced and we will see a much more extensive use of rayon for heat service.

Effect of High-Speed Driving – In addition to the development of heat, high-speed driving is, in itself, responsible for accentuating treadwear. Long, straight, smooth concrete highways invite high-speed driving which is most effective in scrubbing off tread rubber. It is possible for hard drivers to wear their tires smooth in one-third to one-half the mileage that more conservative drivers are getting. This statement, of course, pertains particularly to passenger-car tires and is reasonable since all the power of the motor has to go through the tires. Whatever the car does is done through the tires. Another thing to remember is that, when a car is being driven fast, it must be able to stop quickly and brake applications at high speeds are especially effective in abrading the tread. It is an interesting fact that bus tire mileage is always less than that of trucks, which are run at slower speeds, for the reasons just mentioned, although the buses are seldom overloaded.

The heat from high-speed driving, particularly on rough roads, gravel roads, or unimproved roads, increases the probability of impact or concussion breaks. If the roads are rough enough to cut the tread through to the breaker or cord fabric, tread separation and ply separation can be started readily as shown in Fig. 33. Separation in isolated spots, even though small in area, creates extra frictional heat at that point which in turn makes this particular spot more susceptible to blowout.

In studying the effect of speed of driving on non-skid life of passenger-car tires, we investigated a large number of one model of a certain make of car giving especially good car performance. We interviewed the owners and checked the mileage and wear. Some drivers wore their tires smooth in less than 6000 miles; some got about 25,000 miles; whereas the rest got all sorts of mileages in between. We used the same car, same territory, same tire – everything was the same except the character of driving.

Effect of Overloading – Overloading brings about excessive deflection and, of course, excessive flexing. Flexing in the tire brings about a rapid deterioration of the individual strands of the cord, breaking them at isolated spots. This action progresses until here and there a cord breaks through and, if in one region enough cords have broken, a weakness develops at that point, resulting finally in a blowout.

Overloading not only accentuates treadwear but brings about uneven wear or what we call "wiping" (see Figs. 34,

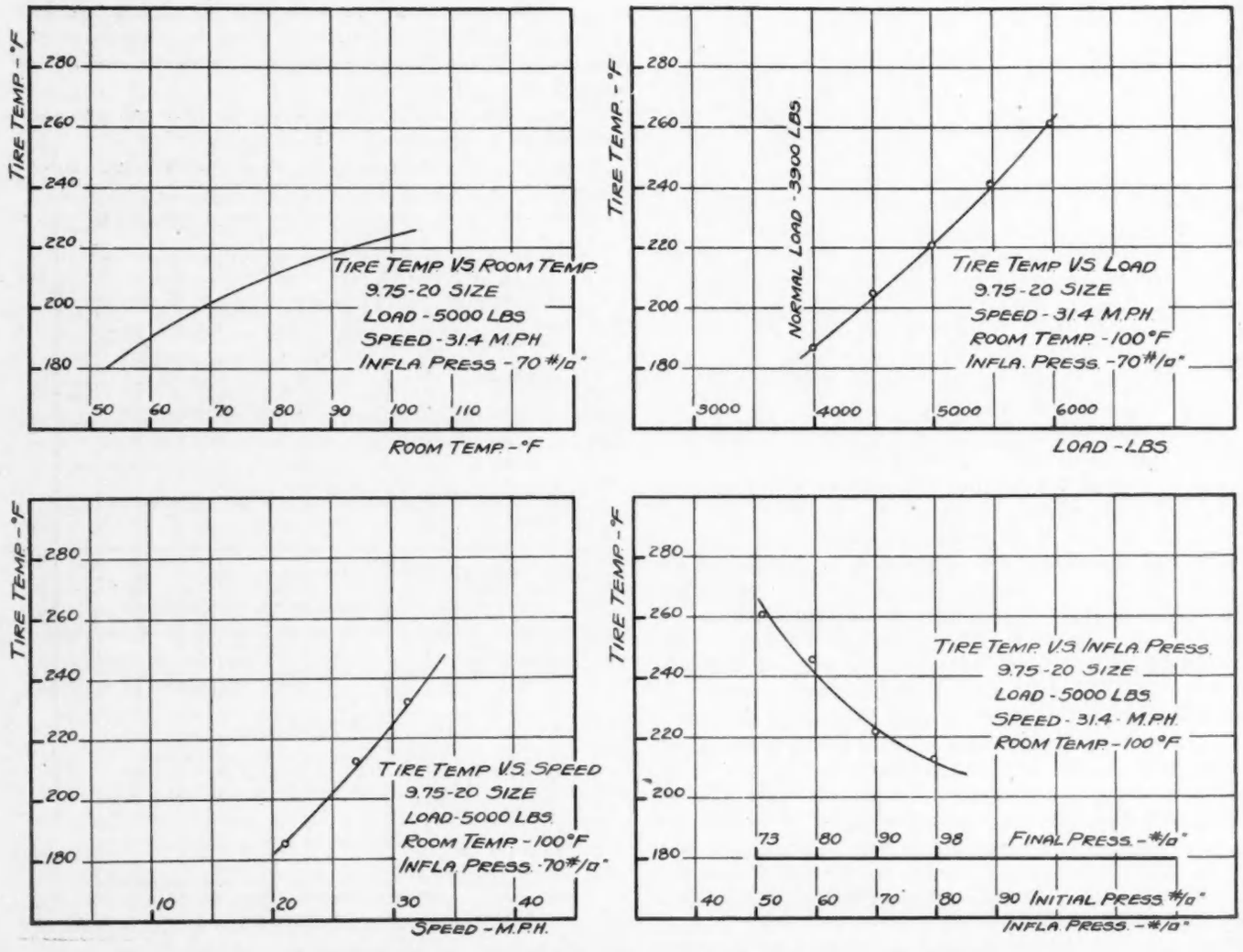


Fig. 32 – Tire Temperature Versus Room Temperature, Load, Speed, and Inflation Pressure

















Fig. 33 – This Case of "Tread Separation" Was Caused by Some Sharp Object Piercing Through the Tread and Starting Separation, also Damaging the Cord Body

Fig. 34 – Characteristic Type of "Tread-Wiping"

Fig. 35 – Sometimes on Front Wheels or Axles the Bearings Get Loose or the Bushings or the Kingpin on Steering Arms Wears, with the Wiping Resulting as Shown

Fig. 36 – One of the Many Types of "Wiping" Observed on Dual Trailer Wheels Where the Axles Sag with Heavy Loads and Crowned Roads

Fig. 37 – These Tires Were All Run on the Same Car at the Same Time, and Show Effectively the Benefits of Proper Air Pressure in Giving Longer Treadwear

Fig. 38 – Low-Pressure Road-Grader Tires Have Given Trouble from "Impact Breaks" from Abuse of this Character

Fig. 39 – Example of "Off-the-Road" Service Among Rocks or Stumps – Sidewall Snags Will Sooner or Later Ruin the Tires If They Are Neglected

Fig. 40 – Example of Bad Case of Front-Wheel Misalignment – The Tread Has Been Worn Through To the Cords at a Low Mileage – Note the "Wiping" Effect as Evidence of Sidewise Scuffing Action

35, and 36) on front wheels, and on free-wheeling dual wheels on trailers. Overloading combined with relatively high speeds induces conditions favorable to impact breaks, particularly if the operation is one tending to develop a high degree of heat.

Effect of Underinflation and Overinflation – It is the air that carries the load in a pneumatic tire. This fact is easy to demonstrate with a passenger car tire casing off the rim. Hold it upright on the floor and just step on the beads. You will notice that it flattens out even under your own weight. To give the maximum possible mileage without premature failure, the proper pressures, as specified in the load-inflation tables (see Tables 2 to 10 in Appendix) is necessary to limit the deflection. Generally speaking, most of the troubles resulting from underinflation occur in a similar manner or in a similar degree from overinflation, with the exception of much less liability to impact breaks. In addition to these troubles we find in passenger-car tires another form of premature failure which we call rim bruises, particularly in 4-ply tires of the smaller sections. The casing is crushed between the rim and such obstacles as curbs or rocks, in a manner in which the cords are broken.

The extent to which underinflation causes tread wear is realized seldom. The average non-skid life of tires on salesmen's passenger cars could be increased from 15 per cent to 30 per cent by simply maintaining the recommended pressures. See Fig. 37. If we express the underinflation in percentage it tells the story more graphically. For example, the recommended inflation is say, 25 lb. per sq. in.; assume that the pressure drops 4 lb. per sq. in. Now, a 4 lb. per sq. in. drop does not sound like much but it is 16 per cent, with a corresponding loss in treadwear. Another important caution to keep in mind is the accuracy of the pressure gage used in checking tires. A gage reading a few pounds off correct will obviously be serious.

It is quite common to find an operation where the truck supervisor decides that he will be able to overload if he overinflates. This is definitely bad practice. It may work for a while but, sooner or later, the tires will fail prematurely by either impact breaks or tread separation. In the first place the high pressure makes the cords of the body construction too taut, and they will rupture by striking a road obstruction. Tread separation is brought on by too high intensity of pressure in the center of the tread which is transformed into a shear action, finally fatiguing the cushion stock to a point of failure.

Effect of Weather – Although not of major importance, we should mention that rubber is a perishable material, subject to deterioration under the actinic rays of the sun. The sidewall of the tire, particularly where it bulges from flexing, develops weather-checking or cracking. Ozone is particularly bad for tires. Any new tire that is inflated on the rim and unduly exposed to sunlight, air currents, or ozone will be affected seriously. It has been found that, by running the new tires to take the stretch out of them for a few hundred miles as soon as they are mounted, they will be much freer from this danger. In storing tires, it pays to keep them in a cool, dark place, free from drafts.

Effect of Roads – Obviously the condition of the roadway over which tires are operated is of fundamental importance in the life of the tire. All through this paper reference has been made constantly to road-surface conditions. The operator should keep constantly in mind the possible effect of all roads upon the tires and, although it may seem strange, this recommendation includes good pavements as well as poor pavements, gravel, and unimproved roads. See Figs. 38 and 39. Each of these surfaces injures in its own way and it requires the habit of vigilance on the part of the driver to favor the tires with very worthwhile results.

Miscellaneous Effects – Curb-scuffing, which is quite com-



Fig. 41 – Casing Worn Out but Apparently in Good Condition, Suitable for Retreading

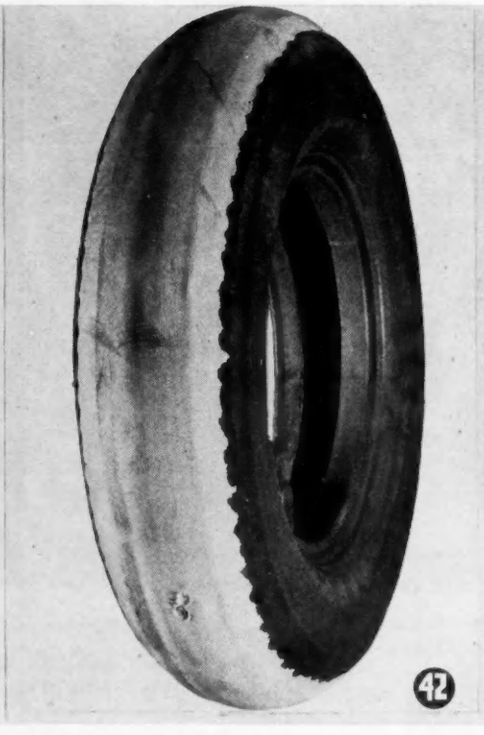


Fig. 42 – Casing Shown in Fig. 41, with Tread Removed and Buffed Ready for "Complete Retreading" – Note That Casing Is O.K. with Only Punctures To Be Treated



Fig. 43 – Same Casing – After Retreading Has Been Completed – Note Shoulder Tread Design of Original Tire as Shown in Fig. 41 Is Entirely Gone – The Evidence of Complete Retread Stands Out in the Complete New Design

mon, fortunately leaves its tell-tale marks easy to check. Curb-scuffing suggests the possibility of rim bruises which are, of course, more serious.

Dual tires on any vehicle should have the same pressure in both tires, the only possible exception being that in some bus operations special pressure differentials have been agreed upon. It is always desirable to mate two tires having the same amount of treadwear on dual wheels, otherwise there may be a noticeable difference between their diameters. In the case of dual tires on trailers where the axles sag or are out of alignment, a tremendous loss of premature treadwear can be avoided if the axles are watched and kept in good shape.

Proper alignment of front wheels will pay dividends any time. Every fleet operator should watch this item carefully and check frequently because, if wheels are out of alignment, treadwear can be run off at an astonishing rate as shown in Fig. 40.

Dragging brakes or brakes which heat up are, of course, bad. The heat radiates from the brake drums keeping the rim hot which, in turn, heats the beads, flap, and tube with a deteriorating effect. If this heating goes too far, there may be actual bead trouble in the tires.

If the rims become battered or bent, they should be straightened if possible; otherwise, they should be replaced. The operator will find it worth his while to spend the necessary money for an adequate size rim, particularly in case that he has to oversize his tires.

Tire records are desirable in most operations. They need not be so complicated as to be burdensome but, properly posted and followed, will justify the expense by revealing abuse, neglect, and improper servicing.

Retreading and Repairs

I want to treat briefly the subject of repairs and retreading. Whether to repair and retread worn casings depends, first, on the condition of the casing, second, on the quality of workmanship that the retread shop turns out, and lastly, on the service to which the reconditioned tire will be subjected. The technique, materials, equipment, and methods have all been developed and perfected by the tire and equipment manufacturers, and they have conducted classes to teach the retreaders and repairmen how to turn out first-class work. There are well-equipped shops in every community, so that it is easily possible to secure satisfaction from repaired or retreaded tires provided everything is handled according to approved practices.

Patronize a reputable retread shop that is equipped with up-to-date equipment, uses up-to-date methods and a good quality of materials, before proceeding to retread. The retread man first should examine the casing thoroughly to determine its condition. If the cord body is intact – no holes, broken cords, loose cords, no ply separation, and no trouble around the bead and no evidence of excessive flexing – the tire is O. K. to retread by any one of several methods. But, if the inspection discloses that a casing is not in perfect condition, then it is necessary first to decide whether the cord body can be repaired with reasonable assurance that the repair will hold up during the life of the retread.

Passenger tires may be retreaded if there are small cord body injuries, but truck-bus tires must have almost perfect cord bodies and are usually rejected if they have anything more serious than nail holes.

Second- and third-line tires are not a very good investment as far as retreading is concerned. There is not enough quality built into the original cord body to wear out the second tread, even if the body appears perfectly sound.

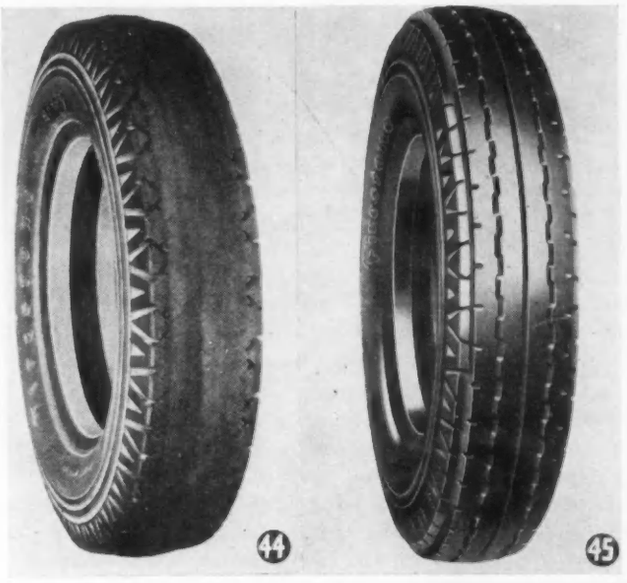


Fig. 44 – A Casing Removed When Apparently in Good Condition and Buffed Ready for "Tread-Capping"

Fig. 45 – Same Tire as in Fig. 44, after "Tread-Capping" – Note that the Original Tread Design Still Shows on the Shoulder of the Casing – The "Tread-Capping" Is Confined to the 4 Ribs

There are several systems of retreading in general use. I will describe the best system and mention one or two others:

The Complete Retread – This is the most satisfactory method, particularly for truck tires. Successive steps are shown in Figs. 41, 42, and 43. It is the only method that can be used with tires that are worn through the breakers or tread plies, or that have suffered from cuts or snags through the tread, which makes it necessary to remove the tread and breakers to determine if the body of the tire is in condition to retread. Passenger car tires may be retreaded by this system without replacing the breakers but, in retreading truck-bus tires, it has been proved repeatedly that retreads with the breakers replaced give best results. Where strictly high-grade materials and workmanship are available and the casing is in good sound condition, the retreads from this system may be expected to deliver approximately 70 per cent to 80 per cent of the mileage obtained from the original tread. The cost for this system averages about one-half the cost of new tires.

With a little study and consultation with the retreader, the operator can learn to size up his tires himself, and decide with reasonable certainty whether they are worth retreading. Or better still, by systematically checking, he can have them removed from service when they are in just the best condition for retreading.

After the tires are retreaded, favor them by putting them into easy service. Do not overload them, and do not subject them to the heat of high sustained speed. Remember that the cord body has already given a lot of service and, if you want it to stand up while the retread is wearing out, it must be given a chance. Retreaded passenger-car tires should not be pushed at the same high speeds as are new tires. On the other hand, we find that a very nice saving can be had if the retreads are used in average operations, as proved by taxicab practice, where it is expected that casings will be retreaded at least two or three times.

Secondary Systems of Retreading – I use this caption although it may not be fair to do so, inasmuch as it may suggest the wrong thought. The intention is to emphasize the fact that a complete retread, as briefly described, is definitely superior to any other system.



Fig. 46 – A Cut in the Sidewall Which Did Not Appear To Be Serious Until It Was Checked by Prying It Open with a Screwdriver



Fig. 47 – Same Cut as Shown in Fig. 46 After Being Repaired

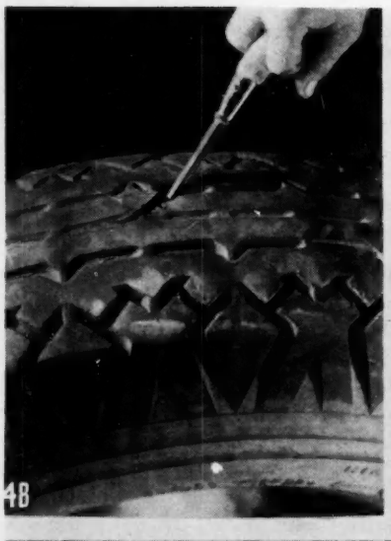




Fig. 49 – Same Tire as In Fig. 48 – Looking Inside





Fig. 50 – Same Tire with the Injured Spot Trimmed and Cleaned Ready for Repairing – The Conical Opening Will Be Filled with Gum and a Cord Cross-Patch Cured Inside

Fig. 48 – An Injury that Looked Like a Rather Small Cut in the Tread, Upon Checking with a Screwdriver, Disclosed a Fair Sized Hole Through All the Cord Plies

Fig. 51 – Same Tire Showing Cord Cross-Patch Cured Inside Casing – The Patch Shows Faintly in the Center of the Picture

Listing them in order of cost and effectiveness, they are:
(1) A retread process known as "semi-capping."
(2) A retread process known as "recapping."
(3) The lowest form, known as "tread-capping."
The characteristics of these systems are as follows:

In the "semi-cap" system the worn tire is buffed all over the complete tread and shoulders and material removed in the center of the tread, down through the breakers or tread plies, and replaced with the new tread stock.

The "recapping" system buffs thoroughly over the complete surface of the tread and shoulders of the tire just enough to remove the non-skid design, but makes no attempt to go through the remaining tread rubber down into the cushion breaker combination.

The "tread-capping" system, as shown in Figs. 44 and 45, simply buffs smooth the tread remaining between the shoulders to prepare a region to receive the new tread stock, confined to the region between the shoulders of the tread.

It can be appreciated readily that these various stages of retreading are merely more economical, short-cut methods of trying to accomplish certain results. The satisfaction the operator will secure from retreading under these different systems, has a decided element of chance for reasons I have mentioned. It all sums up that, in lighter service, on older vehicles, and in easier operating conditions, the various retread methods are probably good enough to justify themselves.

Tire Repairing – It is quite important for the fleet operator to understand just what can be repaired and what cannot be repaired in tires that go out of service from injuries or neglect. A great deal has been accomplished by the tire manufacturers and the companies making repair equipment in developing methods and equipment to produce the most satisfactory repairs.

Here again, as in the case of retreads, only reliable repairmen should be patronized. Repairmen who have modern equipment and properly trained workmen using only the highest-grade materials are able to turn out high-grade work.

There are two major divisions of repair activities: one is the treatment of cuts, snags, and so on, which affect only the external rubber and perhaps one or two cord plies, whether it be sidewall or tread, and the second treats injuries which have ruptured through most or all of the cord body plies.

The external injuries are very common, particularly in the heavier off-the-road operations, described earlier. Often the tread and sidewalls are a mass of cuts and openings more or less serious. See Figs. 46 and 47. These defects should be watched, and it is money in the operator's pocket to have the bad cuts repaired, to keep water out of the cord body, and to prevent the cut from gradually getting worse.

Cord breaks through the majority or all the plies, as shown in Figs. 48 and 49, are of course the most serious problems to tackle because the tire has gone out of service and we face the question as to whether the rupture is too serious to repair. Obviously there is a limit to the size of the hole which can be handled satisfactorily: large X breaks in the crown, diagonal breaks, crown breaks, and flex breaks cannot be treated successfully. In a general way we might say that, if the length of the ruptured spot exceeds 3 to 4 in., the tire is not repairable. Although it is quite common for repairmen to suggest that a larger repair can be attempted if the tire definitely is going to be put in easy, slow-speed service without any overload, they are not disposed to guarantee their work.

The general method of bringing about these repairs is to cut away all the frayed cord, skive a neat hole, filling in the hole with gum and repairing the strength of the broken cords with a suitable number of plies of new cord in the form of a cross-patch, as shown in Figs. 50 and 51. After the cure, we have a repair which can be guaranteed and is definitely superior to the older methods of cementing in a "boot" or building in the replacement cords ply by ply.

Neither ply nor tread separation in any casing can be repaired satisfactorily. Bead troubles are rather limited also. No internal bead troubles are worth while repairing although, where the bead has been chafed externally to a limited extent due to a bent-down rim flange for instance, a patch can be put on where the chafer has not been disturbed and will probably justify the cost.

A temporary repair by putting a "boot" in the casing can be considered only in the light of an emergency to keep the vehicle running until it can be withdrawn from service long enough to remove the tire.

Regrooving the tread to restore non-skid traction effect is quite common, particularly on public carriers which are subject to these regulations by law. Several different forms of regrooving equipment are available and are satisfactory to use, provided the regrooving does not penetrate below the tread stock into the cushion stock.

In conclusion, let me impress upon operators that tires today are made under systems of inspection and controlled methods of production which turn out a uniform product with none of the factory imperfections so common fifteen or twenty years ago. The mileage is built into the tires, and the tire manufacturers would like nothing better than to see their product used under conditions which would deliver more universal satisfaction.

Appendix

Table 1 – Rim and Dual Center Spacings for Truck Tires

THE TIRE AND RIM ASSOCIATION, INC. 35

TRUCK AND BUS TIRES, RIMS, AND DUAL SPACINGS
RECOMMENDED PRACTICE

Tire Size		Original Equipment Recommendation		Consumer's Oversize
Nominal Cross Section	Maximum Cross Section	Recommended Rim Size	Dual Spacing Center to Center	Maximum Oversize Tire That May Be Used by Operator
BALLOON TIRES				
5.50	5.95"	3.75P (5")	7¼"	6.00
6.00	6.20"	3.75P (5")	7⅝"	6.50
6.50	6.75"	3.75P (5")	8¼"	7.00
7.00	7.45"	4.33R (6")	9"	7.50
7.50	7.95"	5.00S (7")	10"	8.25
8.25	8.60"	5.00S (7")	10½"	9.00
9.00	9.70"	6.00T (8")	11½"	9.75
9.75	10.10"	6.00T (8")	12"	10.50
9.75	—	7.33V (9-10")	12¾"	10.50
10.50	11.10"	7.33V (9-10")	13¼"	11.25
11.25	11.60"	7.33V (9-10")	14"	12.00
12.00	12.95"	8.37V (11")	15¼"	12.75
12.75	13.55"	8.37V (11")	16"	13.50
13.50	14.10"	8.37V (11")	16"	No Oversize
HIGH PRESSURE TIRES				
5"	6.10"	3.75P (5")	7⅝"	6" 8-Ply
6" 8-Ply	6.60"	3.75P (5")	—	—
6"	7.15"	4.33R (6")	9"	7"
7"	8.15"	5.00S (7")	10"	8"
8"	9.30"	6.00T (8")	11½"	9"
9"	10.90"	7.33V (9-10")	12¾"	10"
10"	11.75"	7.33V (9-10")	12¾"	No Oversize

NOTE 1 —Chains cannot be used with oversize tires. If it is necessary to provide for chains, use tires shown in first column with corresponding rims and spacings shown. Revised April 6, 1934.
NOTE 2:—The load carrying capacity of a truck rim is the equivalent of the load carrying capacity of the maximum size tire which may be used on the rim as set forth in the above table. Where higher rating for tires is established for slow moving vehicles the oversizing of tires on rims is eliminated, the recommended normal size rim must be used. Approved January 17, 1936
NOTE 3 —Section measurements taken at recommended inflation pressures See page 31.

Table 2 – Load-Inflation Table – PC-1

Type of Tire — Regular Passenger-Car Balloon
Mud-and-Snow Balloon
Class of Service — Passenger Transportation, Mail, Light Delivery, and so on
Type of Vehicles — Passenger Cars and Half-Ton Trucks
Speed Limit — None
Industry Status — Compiled from Several Tire and Rim Association Standards

Nominal Tire Size	Maximum Loads for 4-Ply Tires 32 lb. per sq. in. Inflation 15 in.	16 in.	17 in.	18 in.	19 in.	Maximum Loads for 6-Ply Tires 36 lb. per sq. in. Inflation 15 in.	16 in.	17 in.	18 in.	19 in.
5.50	....	840	955	1000	1040	....	900	1075	1125	1170
6.00	940	990	1025	1070	1115	1010	1065	1155	1205	1255
6.25	1015	1065	....	....	....	1090	1140	....	....	....
6.50	1085	1135	1175	1225	1270	1165	1215	1320	1375	1430
7.00	1240	1295	1330	1355	1405	1330	1395	1465	1525	1585
7.50	....	....	1460	1520	1580	1500	1560	1645	1710	1775

The Firestone Tire & Rubber Co.

Table 3 – Load-Inflation Table – TB-1

Type of Tires — Balloon Regular Truck-Bus
Balloon Trailer
Balloon Intercity Bus
Balloon Mud and Snow
Balloon Rock Resisting
Class of Service — Transportation of Freight, Goods, and Passengers
Type of Vehicles — Regular Trucks, Truck-Tractors, Trailers, Semi-Trailers, City Buses, Intercity Buses
Speed Limit — None
Industry Status — Tire and Rim Association Standard

Size Tire	Standard No. of Plies	Inflation Pressure, lb. per sq. in.	Maximum Load Per Tire in Pounds 15 in. Rim	16 in. Rim	17 in. Rim	18 in. Rim	20 in. Rim	22 in. Rim	24 in. Rim
5.50	6	45	....	....	....	....	1225	....	....
6.00	6	45	....	1160	1210	....	1400	....	....
6.50	6	50	....	1410	1480	1550	1700	....	....
7.00	8	55	....	1640	1725	1800	1950	....	2200
7.50	8	55	....	1860	1950	2025	2200	....	2500
8.25	10	60	....	2240	....	2450	2650	2850	3025
9.00	10	65	2650	2770	....	3000	3250	3500	3650
9.75	12	70	3175	3320	....	3600	3900	4200	4400
10.50	12	75	....	....	....	4400	4700	5000	5200
11.25	14	80	....	....	....	5100	5450	5800	6050
12.00	14	85	....	....	....	....	6250	....	6950
12.75	16	90	....	....	....	....	7200	....	8000
13.50	16	95	....	....	....	....	8200	....	9100

The Firestone Tire & Rubber Co.

Table 4 – Load-Inflation Table – TB-2

Type of Tires — High-Pressure Regular Truck-Bus
High-Pressure Mud and Snow
Class of Service — Transportation of Freight, Goods, and Passengers
Type of Vehicles — Regular Trucks, Truck-Tractors, Trailers, Semi-Trailers
Speed Limit — None
Industry Status — Tire and Rim Association Standard

Size Tire	Standard No. of Plies	Inflation Pressure, lb. per sq. in.	Tire Load, lb. 18-in. Rim	20-in. Rim	24-in. Rim
5 TT	6	70	....	1600	....
5	8	75	....	1700	1950
6 TT	8	75	....	1950	....
6	10	80	....	2200	2500
7	10	85	2550	2800	3200
8	12	90	....	3600	4000
9	14	95	....	4500	5000
10	16	100	....	5500	6000

The Firestone Tire & Rubber Co.

Table 5 – Load-Inflation Table – TB-3

Type of Tires — Start and Stop
Class of Service — City Commercial, City Delivery
Type of Vehicles — Light and Medium Capacity Delivery Cars and Trucks, City Buses, and Taxicabs
Speed Limit — Moderate Speeds, No Long Runs
Industry Status — Commercial Standard

Size Tire	Standard No. of Plies	Tire Load, lb.	Inflation Pressure, lb. per sq. in.	Equivalent Balloon Size
10	6	1130	40	6.00-16
11	6	1140	40	5.50-17
14	6	1400	60	6.00-20
17	8	1700	65	6.50-20
19	8	1950	65	7.00-20
22	10	2200	65	7.50-20
28	10	2800	65	8.25-20

The Firestone Tire & Rubber Co.

Table 6 – Load-Inflation Table – TB-4

Type of Tires — Balloon Passenger Car
Class of Service — Commercial Delivery
Type of Vehicles — One-Half Ton, Three-Quarter Ton, and One-Ton Trucks
Speed Limit — None
Industry Status — Tire and Rim, Experimental Practice

Tire Size	Maximum Rating per Tire 4-Ply Air Pressure, lb. per sq. in.	4-Ply Maximum Load, lb.	6-Ply Air Pressure, lb. per sq. in.	6-Ply Maximum Load, lb.
5.50-16	32	840	..	....
6.00-16	32	990	40	1130
6.00-17	32	1025	40	1230
6.25-16	32	1065	40	1210
6.50-16	32	1135	40	1290
6.50-17	32	1175	40	1400
7.00-15	..	....	40	1415
7.00-16	..	....	40	1485
7.00-17	32	1300	40	1560
7.50-15	..	....	40	1595
7.50-16	..	....	40	1660

The Firestone Tire & Rubber Co.

Table 7 – Load-Inflation Table – EM-1

Type of Tires — Mud-and-Snow Balloons
Regular Truck-Bus Balloons
Rock-Resisting Balloons
Class of Service — Earth Moving
Coalstrip Mining
Quarrying
Lumbering and Logging
City Contractors
Type of Vehicles — Trucks, Truck-Tractors, with Semi-Trailing Units
Speed Limit — 25 M.P.H.
Industry Status — Tire and Rim Association, Experimental Practice with Additions

Tire Size	Standard No. of Plies	Inflation Pressure, lb. per sq. in.	Maximum Load, lb. 20-In.	24-In.
7.50	8	55	2640	3020
8.25	10	60	3180	3630
9.00	10	60	3720	4180
9.75	12	70	4680	5230
10.50	12	70	5230	5930
11.25	14	75	6300	6980
12.00	14	75	....	7740
13.50	16	75	8560	9560
15.00	18	75	....	11940
18.00	20	75	....	16750

The Firestone Tire & Rubber Co.

Table 8 – Load-Inflation Table – EM-2

Type of Tire — Earth Mover, Low Pressure Regular Truck-Bus Balloon
Type of Vehicle — Trailing Units such as Buggies, Wagons, Scrapers
Class of Service — Earth Moving
Speed Limit — 10 M.P.H.
Industry Status — Tire and Rim Association Experimental Practice with Additions

Size Tire	No. of Plies	Load, lb., at Inflation Pressure, lb. per sq. in. 20	25	30	35	40	45	50	55	60	65	70
9.75-20	12	2520	2870	3200	3510	3790	4060	4320	*4570*	....	....	....
9.75-24	12	....	....	3610	3940	4270	4570	4860	5150	....	....	....
13.50-20	10	4450	5050	*5650*	....	....	....	....	....	....	....	....
	12	4450	5050	5650	6160	*6680*	....	....	....	....	....	....
	16	4450	5050	5650	6160	6680	7150	7600	8030	*8450*	....	....
	18	....	....	....	....	....	....	....	....	....	8850	9250
13.50-24	16	....	....	6250	6850	7400	7950	8450	8950	9400	....	....
15.00-20	12	5500	6300	6950	*7600*	....	....	....	....	....	....	....
	16	5500	6300	6950	7600	8250	8900	9500	*10000*	....	....	....
	18	....	....	....	....	....	....	....	....	10500	11000	....
15.00-24	16	6200	7050	7850	8550	9250	9950	10550	11200	....	....	....
18.00-24	12	8700	9850	*11000*	....	....	....	....	....	....	....	....
	16	*8700*	9850	11000	12100	13070	*14000*	....	....	....	....	....
	18	....	....	....	....	....	....	14900	....	....	....	....
	20	....	....	....	....	....	....	....	15750	16500	....	....

Note: Italic figures denote maximum load per tire.
Basis: Truck-Bus Table increased 35 per cent.

The Firestone Tire & Rubber Co.

Table 9 – Load-Inflation Table – RG-1

Type of Tires — High-Pressure Road Grader
Class of Service — Road-Building and Maintenance
Type of Vehicles — Motor and Pull-Type Graders
Speed Limit — 25 M.P.H. Maximum
Industry Status — Tire and Rim Association Experimental Practice

Balloon Marking	High-Pressure Marking	Standard No. of Plies	Maximum Load, lb.	Inflation Pressure,* lb. per sq. in.
7.00-16	....	8	2145	55-80
7.00-20	32x6	..	2340	55-80
7.00-24	36x6	..	2680	55-80
8.25-24	38x7	..	3630	60-85
9.00-20	36x8	..	3720	60-85
9.00-24	40x8	..	4180	60-85
9.75-24	42x9	..	5230	70-95

The Firestone Tire & Rubber Co.

*To control blade chatter, the inflation pressure may be varied to suit operating conditions between the minimum and maximum shown.

Table 10 – Load-Inflation Table – RG-2

Type of Tires — Low-Pressure Road Grader
Class of Service — Road-Building and Maintenance
Type of Vehicles — Motor Graders
Speed Limit — Slow (No Specific Figure)
Industry Status — Tire and Rim Association Experimental Practice

Tire Size	6-Ply Air Pressure, lb. per sq. in.	6-Ply Maximum Load, lb.	8-Ply Air Pressure, lb. per sq. in.	8-Ply Maximum Load, lb. per sq. in.
7.50-24	28	1580	38	1890
9.00-24	24	2040	34	2500
11.25-24	20	2420	28	2940
11.25-28	20	2580	28	3120
12.75-24	18	2960	24	3510
12.75-28	18	3150	24	3740
12.75-32	18	3350	24	3970

The Firestone Tire & Rubber Co.

Discussion

Reviews Tire Experience in C.C.C. Fleets

– B. J. Lemon

U. S. Rubber Products, Inc.

THE first point that I wish to discuss is the question of causes and locations of ultimate tire failure in operations where supervision of the service is not so well established, such as with smaller fleet operators and in such larger operations as that, for example, of Federal control of the large fleets quickly assembled for C.C.C. field work. In operations of these types, experience in tire service is likely to be limited due to a number of important reasons.

Tire mileage on C.C.C. vehicles, for example, was startlingly low, particularly in the early stages of the work. An examination of 4000 tires discarded by camps located in every region in the United States disclosed that, in some regions, as few as 20 per cent of such tires were worn out from normal tread wear. 80 per cent of the tires failed prematurely due, as near as could be diagnosed, to:

(1) Shortage of experienced drivers.
(2) Rough theatre of operation.
(3) Lack of inflation equipment and of experienced maintenance and service personnel.

The first and most important cause of premature tire failure in C.C.C. operation – the influence of the driver – probably will not be subject to any substantial remedy because of the inexperience of the ever-changing youthful enrolees; yet the conclusions drawn from such C.C.C. experience, of the importance of driving and care in vehicle operation, are applicable to all large and small fleet operations. Causes of tire failure (2) and (3) namely, rough going and lack of maintenance supervision and equipment, in C.C.C. operation have shown steady improvement as the complicated problems of supervising such scattered units have yielded to Federal efforts toward safety, education, and correlation of maintenance problems.

Incidentally, this tire survey disclosed little differences in the service rendered by high-pressure and by balloon types of tires in rough operation despite the prevailing practice in some departments of using only high-pressure tires.

The second point, touched on by Mr. Hale, and covered by the government tire survey, was: How economical is the mud-and-snow type of tire in truck operation?

It appears from answers to the questionnaire that the mud-snow tire had been put into service in many localities of C.C.C. operation where conditions did not warrant its use, and where such adaptations became uneconomical. This condition is not at all surprising because, when any new special service product is offered in a large field, such as motor transportation, the operator must learn by experience just where this new product fits into the transportation set-up.

The C.C.C. motor transport system pioneered the extensive use of the mud-snow tread tire for trucks. No such previous experience was available. But the experience and knowledge so gained should not be lost, but utilized by fleet operators generally. Every special service tire, offered by the tire industry, is economical if confined as far as practicable to that special field. It is such efforts as those of Mr. Hale's paper that bring out information which, if digested and applied, will result in saving many tire dollars in fleet operation.

Modern Conditions Demand Modern Trucks

By B. B. Bachman
Vice-President, The Autocar Co.

AFTER tying in truck development with that of passenger cars, this paper differentiates between "engine-under-the-seat" and "cab-over-engine" types, lists the advantages and disadvantages of each, and discusses their design features. A comparison of these types with trucks of conventional design is made by contrasting specific data, such as capacities and dimensions.

SOME forty odd years ago the dream which a few pioneers had of building an automobile started to become a reality. A number of unrelated factors made this realization possible. The internal-combustion engine had been produced in a size suitable for use in a light vehicle. Refining petroleum to produce kerosene yielded gasoline as a byproduct. Rubber tires for bicycles pointed the way to provision for larger sizes and heavier loads.

At the turn of the century a new industry had been born and, after countless years, during which man was dependent on his own legs or those of an animal, a power-operated vehicle for individual transportation became available.

The phenomenal development of the vehicle from a horseless carriage to a multicylinder streamlined car has occupied only a few short years. The development and acceptance of this new device have affected every phase of life.

The steel industry has developed ferrous metals to meet both structural and production requirements. Light-weight alloys of aluminum and magnesium have been developed. Glass sheets which would not shatter have been put into use, and paints and lacquers capable of producing and retaining sightly and serviceable finish have been made available.

So that cost might be reduced while quality was being improved, the construction and design of machine tools were changed completely to provide speed and accuracy never previously conceived. Methods of progressive handling in use in other fields were adopted and developed into an astounding system of mass production.

The suburbs of every city sprang into activity and provided new homes for thousands of people liberated from the limited areas practical when they were dependent on older forms of transportation. Naturally these vehicles required new roads over which to roll, and highway construction became a matter of first importance. The private ownership of roads and bridges could no longer be countenanced and local, county, state, and Federal governments had to learn to build, operate and police highways which would provide adequate facilities.

The automobile has affected the home, the school, the church, and the state. It has made profound changes in politics and economics and the importance to each one of us can best be grasped if we try to visualize what the results would be if it were taken from us.

While we can thus epitomize the service which the automobile has contributed we must, at the same time, see that the motor truck was needed to make the service possible. The truck helps to build the roads; hauls material to construct homes, schools, churches and public buildings; distributes the fuel for our cars to the service stations which dot every highway, as well as the fuel for heating our houses; carries the product of farm and factory and mine; and thus ministers in as great a degree to our well-being and comfort as does the passenger car.

During the time when the automobile was progressing from a toy to a tool, the truck grew from a makeshift to a highly developed and specialized vehicle. Someone early in that day put a box in the tonneau and started to haul goods instead of people.

The first automobiles were called horseless carriages and were built that way, the single- or two-cylinder engine being mounted under the seat. The importance of accessibility to the temperamental machinery of that early day and the requirements of additional space for the larger engines which

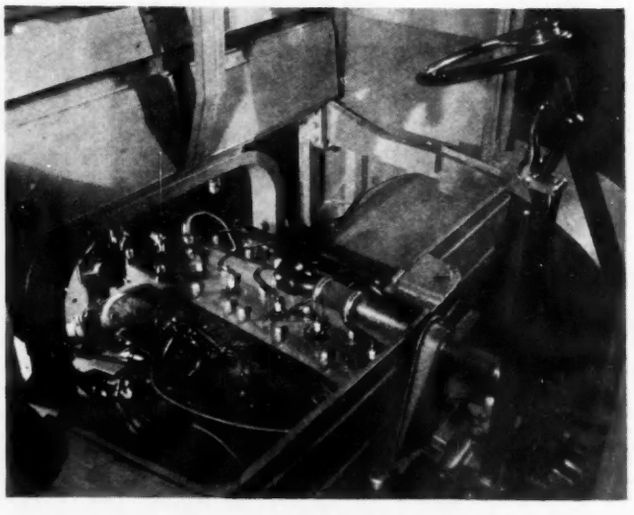


Fig. 1 – Engine-under-the-Seat Construction

[This paper was presented at the Pittsburgh Section Meeting of the Society, Pittsburgh, Pa., Nov. 16, 1937.]

soon were needed, brought about the relocation of the engine and the development of the general arrangement which has persisted to date. Within recent years a number of people have criticized this arrangement and have argued the advantages which would result from a relocation of the powerplant but, to date, no one in this country has risked staking money or reputation in attempting to market such a car.

The truck and bus have followed much the same path, starting with the engine under the floorboards or seat, and changing to a location under a hood. However, both forms have continued their course of evolution and have relocated the engines in a manner more consistent with practical requirements.

In the truck this development has resulted in placing the engine in the cab either ahead of or under the driver's seat. A number of factors have brought the need for this change to the recognition of the industry:

First, the necessity for reducing the length of the vehicle to the minimum needed to carry a body of required length. Several factors influenced this requirement. The increasing amount of traffic on streets and highways placed a premium on space, and regulations limiting the length of single vehicles and combinations made it necessary either to restrict the length of the body or the vehicle.

Second, a short wheelbase is a distinct advantage. The turning-circle dimension is affected very greatly by the length of the wheelbase. For 1 in. taken off the wheelbase, the diameter of the turning circle, as measured at the outside front wheel, is reduced approximately 4 in., keeping the wheel cut the same. We will show in detail later the difficulty in obtaining a similar reduction on a conventional truck.

Third, the general adoption of pneumatic tires made it desirable to modify the load distribution generally convenient and practicable on the conventional vehicle. The use of pneumatic tires makes it desirable, in most cases, to use the same size tire on all wheels. With dual tires on the rear wheels this arrangement results in a tire capacity on the rear axle which is twice the capacity of the tires on the front axle. For a number of reasons which will be detailed later this distribution of weight is difficult to attain on a conventional truck, but is the ideal distribution in the "engine-under-the-seat" or "C.O.E." truck.

Fourth, the limitation for wheel or axle loads contained in many state regulations operates to reduce the gross vehicle weight permitted in the same regulations. As an illustration, Pennsylvania permits 26,000 lb. on a four-wheel truck, but limits the maximum weight on the axle to 18,000 lb. With a conventional truck having a weight distribution of one-fourth front and three-fourths rear, which can be taken as fairly representative, this distribution limits the gross weight to 24,000 lb. To reach the 26,000 lb. permitted, the distribution should be 31 per cent front and 69 per cent rear. Although this distribution can be effected on a conventional truck, it can be made much more readily on an "engine-under-the-seat" truck.

The other advantages which are present in this type are distinctive appearance, reduced frame stresses, and greater driver comfort.

Balanced against these advantages some criticisms and objections have been advanced. They include (1) inaccessibility for daily service, inspection, and repair; (2) unsatisfactory riding qualities; (3) unsatisfactory vision; and (4) unsatisfactory driving control.

An attempt will be made to discuss these alleged advantages and disadvantages from the basis of experience extending over a period of thirty years during which this type of vehicle has been built continuously by our company. The trucks built up to 1932 did not have all the characteristics of those of modern type, but did establish principles of design and afford experience upon which present designs are based.



Fig. 2 - Cab-over-Engine Construction

Many attempts have been made to name this type of vehicle, but in this discussion two names will be used: "engine under the seat" applies to the construction in which the engine is carried under the driver's seat and located back of the front axle as shown in Fig. 1 and "cab over engine" or "C.O.E." in which the engine is mounted ahead of the front axle and, therefore, is located under or projecting through the floorboards as illustrated in Fig. 2.

Our company builds both types. The engine-under-the-seat type is used in heavy-duty units where the engines and transmissions are of our own design and manufacture. The size of the engine used in these heavier units is such that, if they should be mounted forward, they would either crowd the driver or require a very wide cab. Mounted under the seat the only encroachment on the driver or rider is a narrow tunnel which affords a passage for the water outlet pipe and also an air passage from the fan to the engine compartment. It is obvious that this construction calls for some detail modification of the engine, transmission and rear axle which we, as the maker of these parts, can control.

On our lighter units where cost considerations are of great importance, we use the C.O.E. construction. Here the engines are smaller and can be housed conveniently and we can use standard production-run engines, transmissions, and axles which, if they were changed to meet the requirements of the engine-under-the-seat type, would be affected seriously in cost.

The problem of reducing the overall length of a truck is one of reducing the amount of this length occupied by the powerplant and the driver. The ideal solution is illustrated in the modern bus where the powerplant is located at the rear. Except for a very limited field this construction is not applicable to trucks. Vehicles for handling liquids and the smaller type for parcel delivery could be adapted to such a

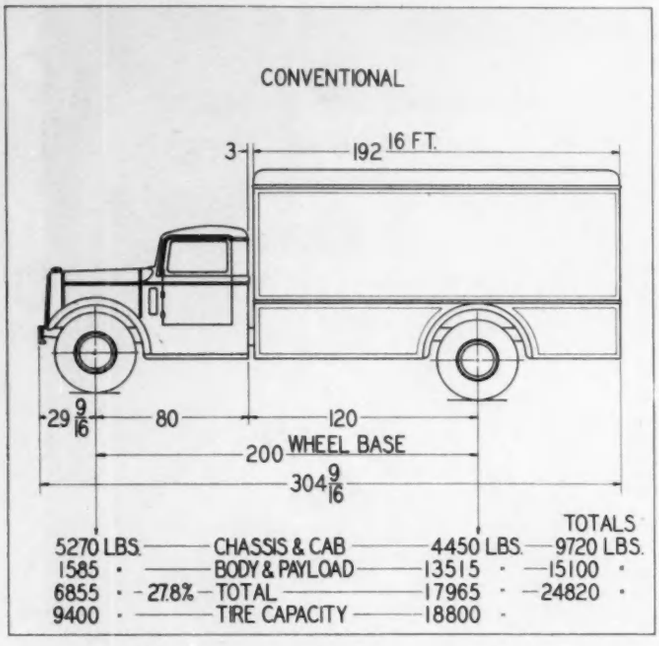


Fig. 3 – Illustration of Conventional Truck

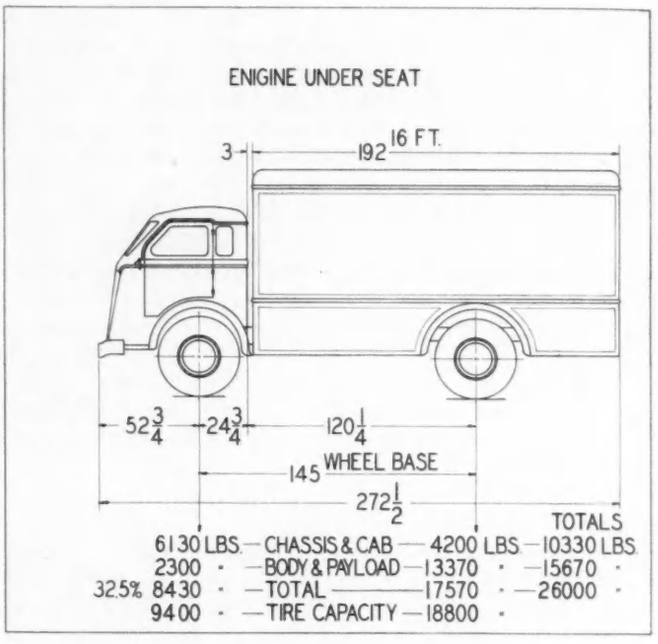


Fig. 4 – Illustration of Engine-under-the-Seat Truck

design but, in general, end-loading is desirable if not necessary with most materials and this loading is not possible with the powerplant occupying one end and the operator, the other.

Following the plan which seems feasible – to bring the operator's compartment and the powerplant space together – we follow a procedure which leads logically to the types under discussion.

We will assume that it is desirable to provide equipment to meet the regulations of Pennsylvania which will carry the maximum payload in a body 16 ft. long. From the conventional truck illustrated in Fig. 3 it will be noted that the wheelbase is 200 in. and the overall length, 304 9/16 in. From the front of the truck to the back of the cab is 109 9/16 in., or about 36 per cent of the total length. Next, turning to the illustration of an engine-under-the-seat truck shown in Fig. 4, it will be seen that the wheelbase is 145 in. and the overall length 272½ in., while the distance from the bumper to the back of the cab is 77½ in., or 28.4 per cent of the total length.

Reducing the total length of the vehicle 32 in. is certainly worth while and, in combination with the 55-in. reduction in wheelbase, it contributes a valuable improvement in operating characteristics.

Both of these vehicles are well within permissible lengths but, in the case where maximum body length is desirable, the 32 in. in total length difference between the two types makes the engine-under-the-seat construction particularly valuable.

Next, turning to the factor of reduction in length of wheelbase, it was noted that this dimension was 145 in. instead of 200 in., or 55 in. less. The diameter of the turning circle on the conventional truck is 74 ft., whereas, on the engine-under-the-seat truck, it is 50 ft., a reduction of 24 ft., or 32.4 per cent. Such a reduction in turning-circle dimension is of the greatest importance in improving the mobility of the vehicle, particularly in city traffic.

Fig. 5 illustrates the factors controlling the diameter of the turning circle. The length of the wheelbase and the angle through which the outer wheels are turned determine the radius of the turning circle for the outside front wheel. The angle through which the wheels can be turned is controlled by tire size, frame width, spring location, front-wheel tread, location of drag-links, and so on. In addition there is the limitation imposed by the axle itself which is established by the clearance in the yoke end of the spindle and the angular relation of steering arms and tie-rod.

Data sheets of the Timken-Detroit Axle Co. show a maximum wheel cut of 37 deg. for most of their axles and 40 deg. and 42 deg. for the two largest axles. In order to satisfy the conditions shown in the figure, the outer wheel turns through an angle about 10 deg. less than the inner wheel. As a consequence the maximum turning angle practicable for the outer wheel is 27 deg. to 32 deg.

In the case just cited other factors limited this angle to 27 deg. on the conventional truck while the engine-under-the-seat truck was 29 deg. Removal of these limitations and the use of an axle having a maximum wheel cut of 42 deg. would reduce the turning circle on the conventional truck to 63-ft. diameter, which is still 13 ft. greater than the "engine-under-the-seat" truck.

To reach the diameter of 50 ft. it would be necessary to have a wheel cut of 42 deg. for the outer wheel and 52 deg. for the inner wheel. Not only would it be impracticable to use such an extreme cut, but it would require a considerable increase in front-wheel tread to provide the needed clearance.

Although the clearance required for the outside front wheel has been under consideration in the previous discussion, it must be remembered that, above the curb line, other portions of the chassis project outside this path which would strike poles, walls, fire hydrants, and so on, that extend above the curb line. This extreme point is usually the end of the bumper. In the trucks which have been compared, this point is 31 in. farther out on the conventional truck and 54 in. on the engine-under-the-seat truck. Therefore, the extreme turning circle is 79 ft. 2 in. and 59 ft. respectively, or 20 ft. 2 in. less for the "engine-under-the-seat" truck.

Although the discussion so far in this section has been aimed at developing the importance of total length and wheelbase reduction on mobility, it must be remembered at the same time that these dimensions can be reduced in the engine-under-the-seat truck to a value which cannot be approached in the conventional truck, except by restricting the body space to an unusable dimension. As an example we have models

in the lighter series of 84-in. wheelbase with a 62-in. C.A. (cab-to-axle) dimension and in the heavier series of 98-in. wheelbase with a 73-in. C.A. dimension.

The reduction in the length of the wheelbase also reduces the beam span of the frame. This reduction lowers the bending moment and, for the same section, reduces the stress deflection. Fig. 6 shows this change for vehicles of equal body capacity and with the same imposed loads.

Referring back to Figs. 3 and 4 we note that, for equal rear-axle loading of 18,000 lb., the "engine-under-the-seat" truck can be loaded to 26,000 lb. gross, whereas the conventional truck can only be loaded to 24,820 lb. The added chassis weight of 610 lb. still leaves a net payload difference of 570 lb.

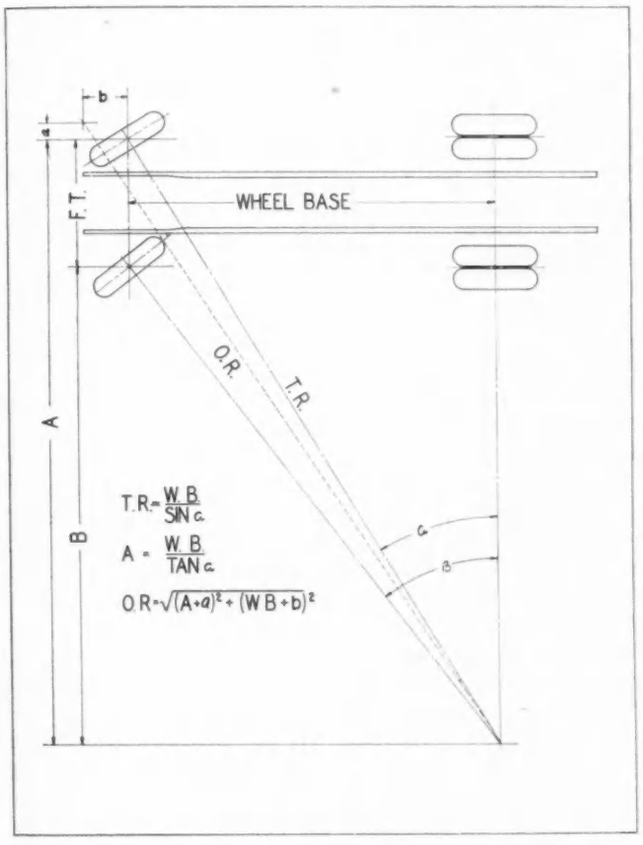


Fig. 5 – Factors Controlling the Diameter of the Turning Circle

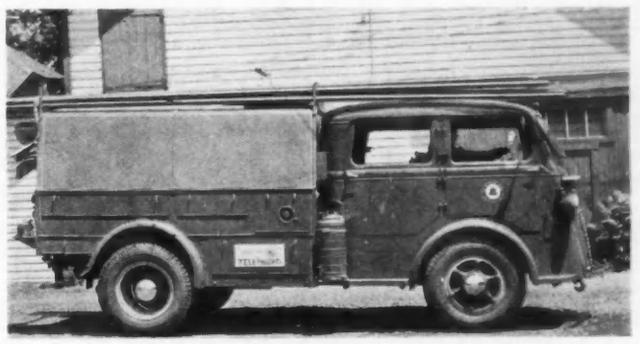


Fig. 7 – Public Utility Truck that Illustrates the Reduction in Length Made Possible by the Engine-under-Seat Construction

To obtain this same increase of payload on the conventional car it would be necessary to increase the wheelbase to 207 in. This change would increase the turning-circle diameter to 76 ft. 4 in., which is undesirable.

If the front axle is moved back, leaving the rear-axle stay in the same relation to the body, it is necessary to move it back 47 in. reducing the wheelbase to 153 in. This change is what we are doing by adopting the engine-under-the-seat construction. It is, of course, possible to move both the front and rear axles back relative to the body either keeping the 200-in. wheelbase or reducing it. No benefit is obtained by such an alternative unless the whole step is taken.

The illustration given for the conditions obtaining in Pennsylvania are not particularly favorable. In other states such as New Jersey, New York, Massachusetts, where 28,000 and 30,000 lb. are permitted on four-wheel trucks, a much more favorable case can be set up. To illustrate this point, let us assume a gross of 30,000 lb. permitted in Massachusetts. With the same relative wheelbase, C.A., and body dimensions used before, we find that the conventional truck carries 22,255 lb. on the rear axle, which is an overload of 500 lb. per tire for 10.50-22 tires. On the other hand the engine-under-the-seat truck carries 20,590 lb. on the rear axle, or 150 lb. more than

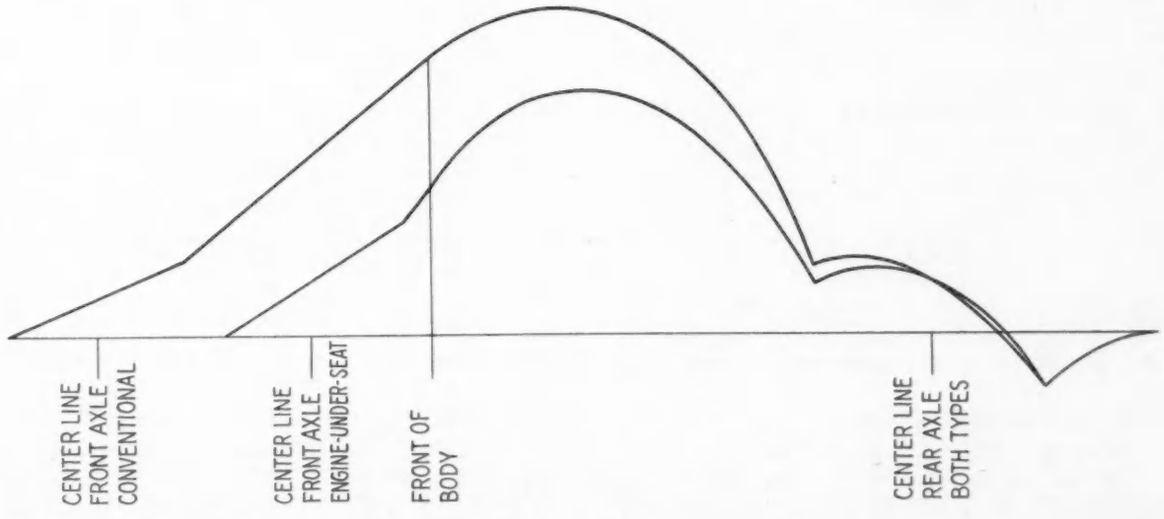


Fig. 6 – Relative Magnitude of Bending Moment for Conventional and Engine-under-Seat Trucks with the Same Body and Rear-Axle Load

the tire rating. In this case we must either reduce the payload on the conventional truck 2400 lb. to effect a reduction of 1665 lb. axle load, or increase the tires to 11.25-20, or change the wheelbase as discussed before. From lighter vehicles much the same results are obtained: assuming a maximum gross of 16,000 lb. with similar bodies we find that, on the conventional truck, the rear axle weight is 11,320 lb. and 10,410 lb. on the "C.O.E." truck. Consequently we must again either lower the weight in some way as already discussed, or increase the tire size.

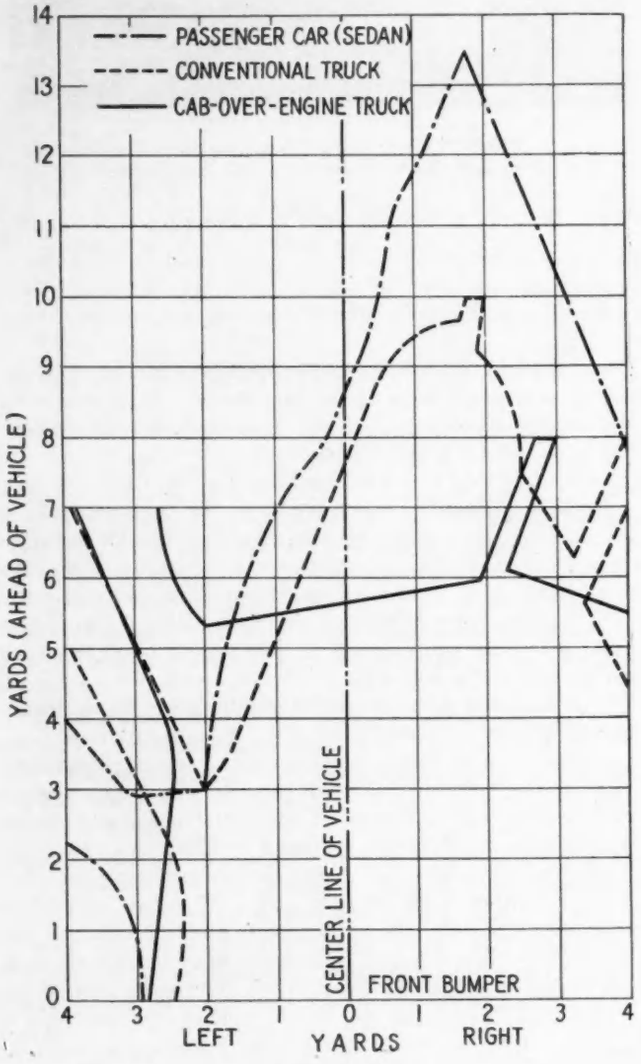


Fig. 8

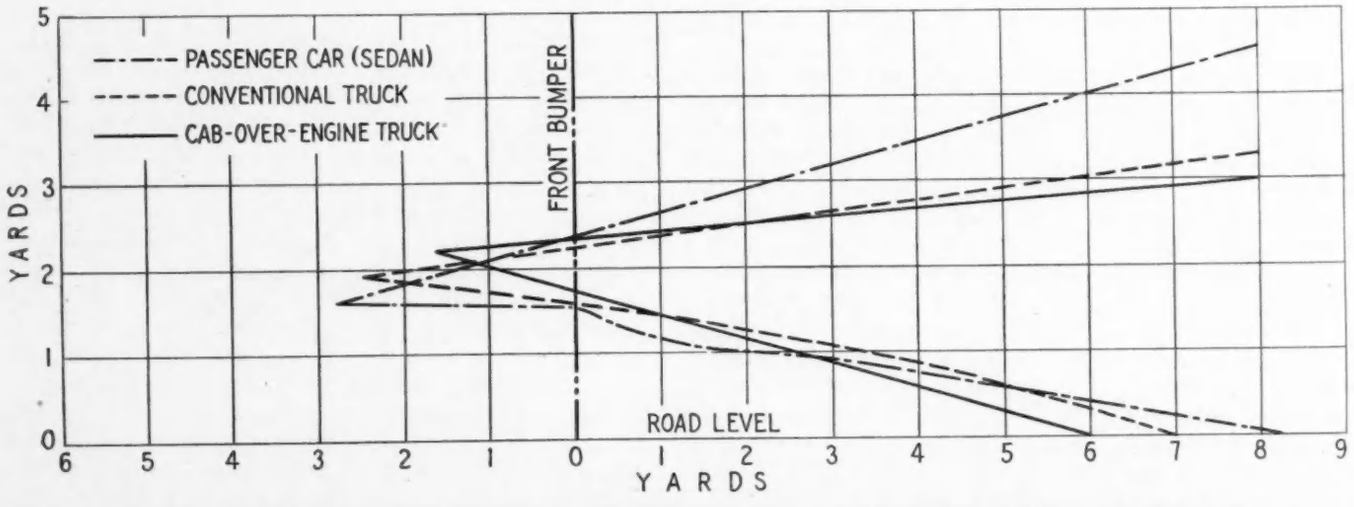
Fig. 7 illustrates quite graphically the advantages which can be obtained by the reduction in length made possible in the "engine-under-the-seat" truck. Here we have a unit for public utility line work equipped with a winch and a body of 109 in. length requiring a 67-in. C.A. dimension. In addition a cab is mounted on the chassis which provides seating for three additional men. To do this job requires an additional 35 in. for the length of the cab. The whole is carried readily on a wheelbase of 124 in. with an overall length of 218⅞ in. On a conventional truck a wheelbase of 178½ in. and a total length of 256¾ in. would be needed; an increase of 54½ in. in wheelbase and 37⅞ in. in overall length.

Note: In the discussion of turning circle the diagram and calculations have been simplified by assuming the tread and king pin centers equal (center-point steer). The error introduced by this assumption is not significant.

The driver who has been accustomed to driving a conventional truck naturally has a feeling of strangeness when he takes the wheel of one of these newer types. The absence of any structure ahead of him requires some readjustment of sensing the vehicle's position on the road and relation to other objects. Once this readjustment has been accomplished drivers almost as one prefer to drive the "engine-under-the-seat" truck. Where the driver gets the feel, the absence of the familiar front end, instead of being strange, is welcomed because the clearance needed is greatly reduced and the shorter wheelbase and turning circle afford easier handling. It will be noted that, although the shorter wheelbase provides a shorter turning circle, at the same time less wheel cut provides the same turning circle, and the combination of these two features makes a notable difference in handling.

The riding characteristics of this type we believe to be definitely better. Naturally the difference in weight distribution calls for changes in springing, principally increasing the stiffness of the front springs to provide for the greater weight which they must carry. The improvement in ride is due, however, principally to the fact that the driver is moved forward away from a position which is affected by the action of both the front and rear axles to one which is affected by the front axle only. In this position the motion is practically a bounce instead of a combination of bounce and pitch and the horizontal components of pitching are reduced to a minimum. This motion is much less tiring, particularly when the truck

(*Continued on page* 130)



Figs. 8 and 9 – Comparison of Visibility of Passenger Car, Conventional Truck, and Cab-over-Engine Truck

Maintenance of a Concentrated Fleet of Large Trucks

By Gavin W. Laurie
The Atlantic Refining Co.

MAINTENANCE economy is seldom a realization unless sufficient consideration has been given to the major factors entering into the cause for maintenance.

The expenditures necessary to correct normal wear conditions often represent a very small portion of the total maintenance cost. The operator is presented too frequently with the problem necessitating design corrections before the truck has operated many thousand miles. The responsibility for failures of this nature may rest with either the purchaser or the manufacturer.

The purchaser may, due to insufficient thought having been given to the contemplated assignment of the truck, misinform the manufacturer as to the load to be carried or the terrain to be traversed; or his appropriation for purchase may be insufficient for a suitable piece of equipment.

The manufacturer, on the other hand, may accept the order, knowing that he cannot furnish suitable equipment. Also, he often resorts to the use of obsolete parts, knowing that it will be necessary for the operator to make replacement at an early point in the operating life of the truck.

The maintenance of a fleet of large trucks is resolved into a reasonably simple procedure where the drivers are well trained, where the lubrication is systematic and thorough, and where the equipment is constructed in proportion to the delivery problem.

TO determine definitely with any degree of accuracy which of the major contributing factors are responsible for the high or low maintenance costs of a concentrated fleet of large trucks, it is necessary to have available detailed records of the cause of each item of expense, together with a detailed record of the corrective measures employed. It is apparent that all items which do not reveal a maximum return must be eliminated.

The major factors referred to in their apparent natural sequence are: records, planning, engineering, operation, maintenance, and replacement.

If we are to extract the maximum return from each item, the question of the "why" of the item must be answered first and the answer must come from past experiences or experimentation. If records are not available of past experiences, it becomes necessary to repeat experiments.

Fig. 1 illustrates a master maintenance card, and

Fig. 2 shows a specification record.

From the records we obtain the information that selection of type, size, and design of a motor vehicle is limited by the following factors:

(1) Municipal and/or state weight limitations.
(2) Municipal and/or state volume limitations.
(3) Topography of the operating territory.
(4) Type of goods to be transported – "solids or liquids."
(5) Size of deliveries to be made which may be full loads or part loads.

All of the preceding factors except the last are fixed factors. The last one is fixed or variable, dependent upon the policies established in the agreement between the vendor and vendee.

Where a definite understanding can be obtained on all the items mentioned previously, the choice or selection of equipment may be made quite readily. Where the factors are subject to change, change in type of vehicle becomes necessary.

It is all-important to direct the attention to low unit cost of delivery which is directly dependent upon the known factors.

The largest possible delivery in the shortest possible time produces, naturally, the lowest possible unit cost.

Various trip lengths, time of loading and unloading, average speed of vehicle employed, plus time spent on other miscellaneous activities pertinent to the delivery, must be studied and regular schedules or time tables instituted in an endeavor to establish uniform and smooth operation. Fig. 3 shows a time table and scheduling chart.

This type of planning points toward high efficiency. It demands that attention be given to all individual cases, the collection of which data permits accumulated detail to be used effectively.

Planning and maintenance records should be studied closely before any attempt is made to draw up specifications for a piece of equipment. The information contained in the planning records should enable the engineer to determine the size and type of equipment best suited to the operation. If the equipment is being purchased to replace a worn-out or ob-

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1938.]

EQUIP. NO.

MAINTENANCE RECORD

Engine Serial No. Date & Mileage	cyl. Bl. Ass.	Cylinder Head	Exhaust Manifold	Muffler & Tail Pipe	Timing Gears	Camshaft			Crank Shaft		Connecting Rods			Pistons				Valves						SCHEDULED REPLACEMENTS	
						Comp.	Bearings	Gears	Comp.	Main B.	Comp.	Bearings	Bushings	Comp.	Rings		Pins	Valves		Inserts	Springs			UNIT	Mileage or Date

Engine Accessories	Fuel System				Oil		Cooling System				Electical System											Windshield Wiper	Air Compressor
	Carburetor	Air Cleaner	Fuel Pump	Governor	Pump	Filter	Wat. Pump	Thermostat	Fan	General	Generator	Volt. Reg.	Battery	Start. Motor	Distributor	Condenser	Coil	Spark Plug Sets		Magneto	Speedome'r		

Transmission System	Clutch			Transmission		Propeller Shaft		Differential		Rear Axle Shafts	Rear Wheel Bearings	Rad. Rods & Torq. Arms	Front Axle			Front Wheel Bearings	Steering Gear Assembly						
	Assembly	Plates	Discs	Assembly	Gears	Assembly	Univ. Joints	Assembly	Gears				Assembly	Pins & Bush.	Tie Rod Ass.								

Chassis	Frame	Cross Members	Front Springs								Rear Springs							Trailer Hitch							
			Right		Left		Rebound	Bumper Bl.	Shackles	Pins	Right		Left		Helper	Shackles	Pins								

Brake System	Brake Linings					Brake Drums		Mechanical			Vacuum			Air					Hydraulic						
	Front	Foot	Rear	Foot	Hand	Front	Rear	Brake Cams	Brake Cab.	Slack Adj.	Power Cyl.	Ext. Valve	Check Valve	Governor	Brake Valve	Relay Valve	Em. Re. Val.	Quick Re. Val.	Brake Chem.	Master Cyl.	Oper. Cyl				

Operation Equipment	Tank	Tank Mounting	Tank Piping	Hydraulic Compartment Valve System					Mech. Comp. Valves		Power Take-Off	Pump	Meter Assembly				Hose Reel							
				Assembly	Operator		Distributor	Valves	Assembly	Valves			Assembly	Meter	Flow Reg.	Carrier								

General	Paint																								
	Date & No. Job	Make Pain																							

LEGEND O - ORIGINAL A - ADJUSTED S - REPAIRED R - REPLACED S. R. - SCHED. REPL. G - GROUND

Fig. 1 – Master Maintenance Card

8-37 6-46

Equip. No. _____ Purchased _____ Make _____ Model _____ Serial No. _____ Master Specification Card No. _____

LOCATION	DATE	LOCATION	DATE	LOCATION	DATE	LOCATION	DATE

Tank	Furnace Oil Delivery Hose
Number	Diameter
Make	Length
Construction	Couplings
Type	
Total Capacity	Furnace Oil Delivery Hose Reel
Make of Faucets	Make
	Type
Compartment Valves	Location
(a) Valves Make	
Type	
(b) Operating Unit Make	
Type	Dray Body
	Make
Gasoline Delivery Hose Tube	Type
Type	Capacity
Diameter	Inside Width
Length	Inside Length
Number	Height of Floor
Location	
Gasoline Delivery Hose	
Diameter	Pump and Tank Body
Length	Make
Couplings	Type
Number	Capacity
	Inside Width
Bucket Box	Inside Length
Type	Height of Floor
	Type Tool Box
Can Racks	
Type	MONTHLY MILEAGE
Number	193 193 193 194 194
	Jan.
Delivery Meter	Feb.
Make	Mar.
Type	Apr.
Size	May
Location	Jun.
	Jul.
Delivery Flow Regulator	Aug.
Make	Sep.
Type	Oct.
Size	Nov.
Location	Dec.
Furnace Oil Delivery Pump	
Make	
Type	
Capacity	
Size	
Location	

TESTS CONDUCTED ON THIS EQUIPMENT	DATE STARTED	DATE COMPLETED

SPECIAL ACCESSORIES INSTALLED ON THIS EQUIPMENT	DATE INSTALLED	DATE REMOVED

VARIATIONS FROM SPECIFICATIONS ON MASTER CARD	DATE	MILEAGE
INSPECTIONS		
A B C D E F G H I J K L M N		

Fig. 2 – Specification Record

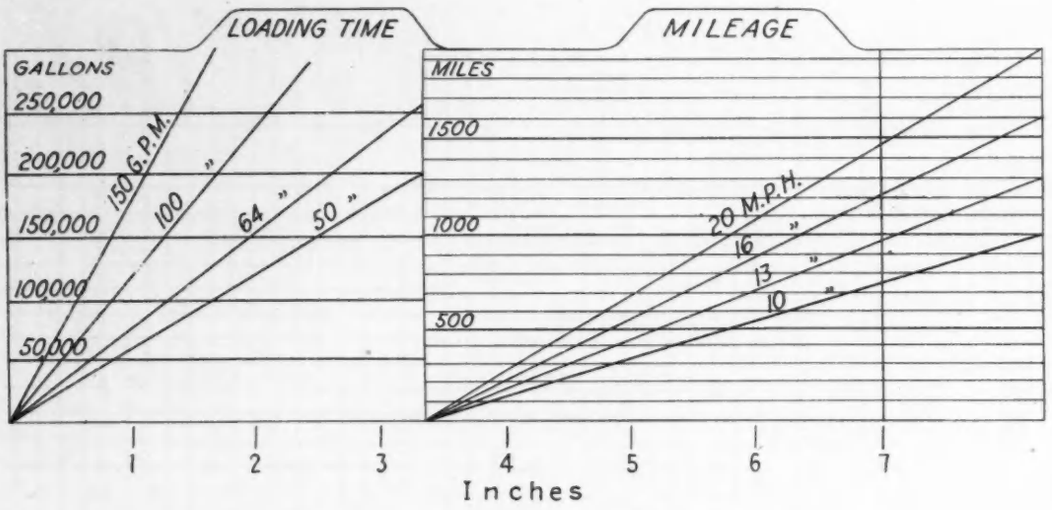


Fig. 3 – Time Table and Scheduling Chart

Efficiency of operation may be determined by employing the use of standard measuring devices based on averages of all factors consuming time. Two factors in truck operation are explained and exhibited at left.

The scale of the measuring device used is based on 20 hr. to 1 in. Average speeds according to the type of vehicle and territory traversed are shown; loading time in gallons per minute according to available head.

An example revealing the time consumed in loading 200,000 gal. at a rate of 50 gal. per min., in addition to running 1200 miles at an average speed of 16 m.p.h. is shown by measuring the number of inches on the full-scale chart between the beginning of the operation and the end, which is 7 in. or 140 hr. (The chart is shown in reduced scale.)

solete piece of equipment, the maintenance records should enable the engineer to correct items that have been a source of constant trouble to the type of equipment being replaced.

Having decided on the type and size of equipment for the particular job, the problem of the engineer is now to design the complete unit for low maintenance. Records should reveal to the engineers any necessity for consideration being given to change of mounting style to eliminate such items as broken frames and, if possible, the design should combine features to allow for accessibility of lubrication and maintenance. Such a design is shown in Fig. 4.

The engineer's responsibility is not confined solely to the mechanical structures for prevention of failures, but should extend also to such items of operating expense as washing, greasing, and painting of the equipment, tire changing, ease of loading and unloading, and safety features.

One example of engineering effort directed toward reducing maintenance is illustrated in the application of porcelain signs tack-welded and sealed to the sides and end of the truck tank as shown in Fig. 5.

The activating influence in this particular case originated from the study of the records revealing excessive expenditures necessary to maintain the appearance of the tank. Investigation of the former high repeat-painting costs for this item revealed that the disfiguring of the paint was caused by buckles and buttons of the driver's wearing apparel. These objects have no effect on the surface of the porcelain signs.

If properly instructed, the driver becomes an important factor in the economic maintenance of the vehicle. Where the records reveal excessive tire expense; abnormal brake lining; clutch, valves, and unit replacements for an individual piece of equipment, the cause often can be traced to inexperience or wilful disregard of the Driver's Operating Instructions. The driver's daily report on the operation of his



Fig. 4 – Truck Design That Allows Ample Accessibility for Lubrication and Maintenance

piece of equipment often warns the mechanic on the job sufficiently in advance of an impending failure and thus enables the mechanic to correct the condition before it becomes an item of major expense.

In dealing with the subject of maintenance, one phase of the problem includes the repairs and replacements of parts due to premature failure and normal operating life failure. As used in this paper, premature failure will mean failure of parts due to manufacturer's errors in design, selection of materials, or the use of faulty parts that would have been rejected had they been uncovered at the time of inspection prior to assembly. Such failures as these, brought to the attention of the manufacturer, in most instances, are admitted, and corrective measures are applied. However, the material cost is

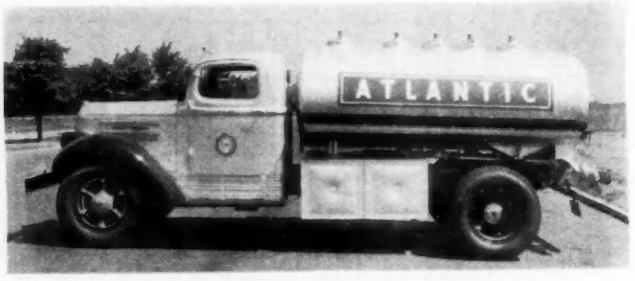


Fig. 5 – Truck Equipped with Porcelain Signs to Facilitate Maintenance

often an item of minor expense as compared to the loss sustained due to the equipment being out of service. A more serious aspect of such failures is the one involving a potential accident source where a weakness in steering, brakes, or driving train is directly responsible.

The foregoing discussion, as given, pertains to items uncovered by the operator and reported back to the manufacturer for correction. A closer servicing contact between the manufacturer and the operator should be encouraged, with the purpose of divulging to the operator failures uncovered by the manufacturer. In many instances, the manufacturer is cognizant of the existence of a weakness, and this condition is evidenced by the fact that corrective measures are immediately available.

With respect to the study of replacements due to normal operating life failure, the basis of scheduling for service is determined by operating data and records. Reference to these records, together with the results of relative test items, in

most cases, serves as a basis for setting up the preventive maintenance schedule.

A single example of this method may be found by referring to Figs. 6 and 7, a study of oil consumption and connecting-rod bearing failures on a rather large group of trucks. The oil-consumption curve of the original motors of this group, Fig. 6, increases rapidly until approximately 25,000 miles of operation when the slope of the curve flattens out. If the trucks are allowed to operate for 42,000 miles before rings are replaced, the rate of oil consumption increases rapidly with mileage as shown. With only such information as this in Fig. 6, the choice of mileage for ring replacements would be gained by a cost balancing of labor and materials involved versus savings in oil. However, Fig. 7 reveals that, at approximately 25,000 miles of operation, cracked rod bearings

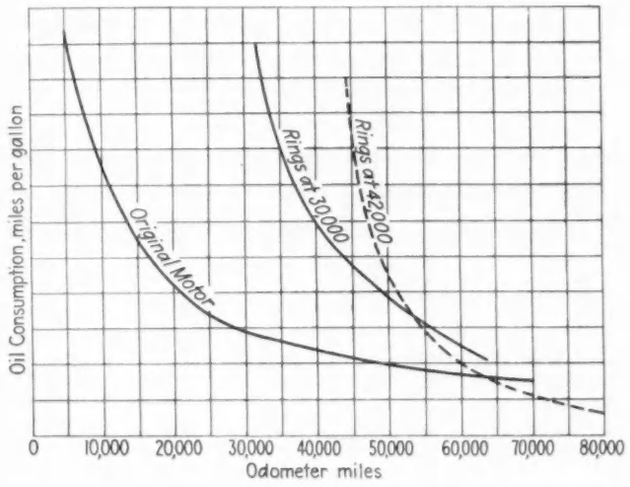


Fig. 6 – Oil-Consumption Curve with Original Motor and Various Ring Replacements

are encountered and the numbers increase from that point as shown. We now have an apparent operating limit, and the result has been a scheduling of this group of trucks for rings and inspection of rod bearings every 30,000 miles of operation. This is classed as a *G* inspection as shown in Fig. 8, schedule for truck Group No. 1, and was placed in the 30,000 mile interval instead of at 25,000 miles to conform to the other types of inspections shown on Fig. 8.

This information is included in the paper to demonstrate the method pursued to fix maintenance schedule points at the most economical and practical basis resulting from experimental facts and records.

Fig. 9 gives 17 examples of the 104 items covered by Schedule *A*, Fig. 8.

Fig. 10 is an explanation of the examples of schedule items given in Fig. 9.

We must not lose sight of the fact that improvements in design, materials, operation, and so on, that accompany the introduction of new models affect the maintenance scheduling of these units, and it often becomes necessary to re-schedule the new fleet. In many cases the improvements are such that the operating mileage schedule for maintenance can be extended, but the location of the actual mileage periods again is controlled by the type of study given previously. As an interesting point on this question, Fig. 11 shows the improvements in design for the same make and type of truck as reflected by the oil-consumption curves.

The mention of new models naturally brings up the question of replacements. In this respect, each replacement should be considered as a separate physical problem, and its determination reverts back to the first portion of the paper, namely, the factor of records.

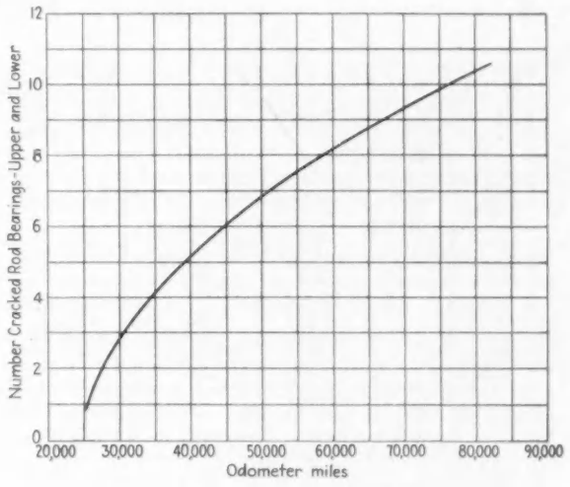


Fig. 7 – Connecting-Rod Failures Versus Miles

From the records are obtained such items as:

(1) Variable expense per mile of the old unit.
(2) Miles per year.
(3) Physical condition of the old unit.
(4) Miscellaneous factors, such as state regulations and so on.
(5) Variable expense per mile of the new unit (estimated).
(6) Savings, if any, in fixed charges due to increased speed, mobility, and other physical factors.
(7) Savings in expense per mile.
(8) Overhaul estimate.
(9) Trade-in value of the old car.
(10) Net purchase price.
(11) Net return on net investment.

The foregoing variables encountered in any specific case

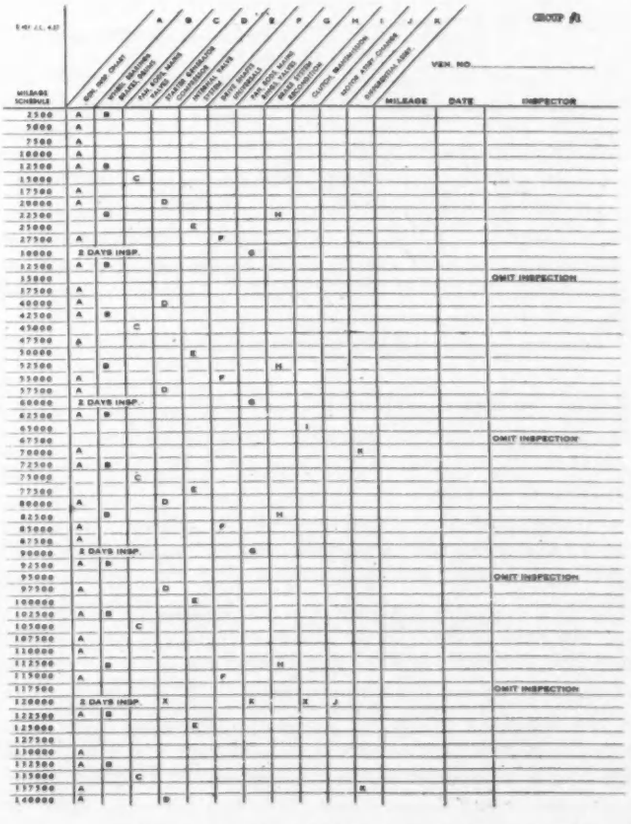


Fig. 8 – Mileage-Inspection Schedule for Group 1 Trucks

B-18 VEHICLE INSPECTION REPORT - CHART "A"

Veh.No.....Location..... Mileage .....Date

1 DRIVERS REPORT. On vehicle condition and operation.		
2 EXTERIOR CONDITION. Visual inspection. Tighten License Tags.		
3 PAINT CONDITION. Good Fair Poor		
4 START ENGINE. Note, starter action. Engine noises. Dash instruments.		
5 CHOKE. Operation and condition.		
6 DRIVING LIGHTS and Switches. Check bulbs, reflectors, lenses and wiring.		
7 SIGNAL LIGHTS and Switches, for operation. Check wiring, bulbs and lenses.		
8 HORN. Test operation. Button, wires and fastenings.		
9 SPARE BULBS & FUSES. Check and replace.		
10 FIRE EXTINGUISHER. Inspect and fill.		
11 CLUTCH AND BRAKE PEDALS. Test travel, clearance and free operation.		
12 EMERGENCY BRAKE LEVER. Test for holding, operation and condition.		
13 GEAR SHIFT LEVER, and connecting links for operation and condition.		
14 POWER TAKE-OFF LEVER, and connecting links for operation and condition.		
15 W.S. WIPERS. Test.		
16 R.V. MIRRORS. Tighten and adjust.		
17 GLASS IN CAB & WINDOW LIFTS, Inspect.		

Fig. 9 - 17 of the 104 Items Covered by Schedule *A*

certainly complicate the working out of a universally acceptable solution. It will be noted that no mention was made of depreciation, whereas depreciation is unquestionably an operating expense. Also, it can be seen readily that, regardless of the plant investment already made, if the saving in operating expense of the proposed new equipment is sufficiently large, the capital investment is warranted.

The forms submitted as exhibits were drawn up for a fleet comprised 90 per cent of tank trucks. The remaining 10 per cent of the fleet consists of dray-type trucks, all of which are maintained by the same schedule forms eliminating, of course, the instructions pertaining exclusively to tanks, cargo pumps, and power take-offs.

Figs. 1 and 2 are examples of the master maintenance record for each truck in operation. Fig. 1, it will be noted, gives the complete mechanical history of the equipment from which specific information can be obtained regarding the maintenance items. These items are summarized monthly, and any abnormal repairs or replacements can be detected at once.

On the reverse side of this card, represented by Fig. 2, equipment specifications are noted, and space also is provided for special accessories and test materials. In addition, there is a column which is used for recording the type, date, and mileage of the last inspection.

The origin of the information for this record is obtained from the mechanic's vehicle inspection report. Examples from an *"A"* inspection report are illustrated in Fig. 9, and the general explanatory notes for 17 of the 104 items of the *A* inspection are shown in Fig. 10.

Offhand, it appears that these 104 items of the *A* inspection would create the impression of a voluminous task, but a close study of the individual items involved reveals that the majority of them require little time to complete.

The control chart for the trucks of Group 1, Fig. 8, lists the other inspection items from *A*, the example given previously, to inspection *K*. The detailed items of these various inspections, together with their explanatory notes, are handled in the same manner, at the mileage periods given in Fig. 8, as the exhibits shown on Figs. 9 and 10.

Lubrication plays a very important part in the maintenance of a vehicle. It combines the proper application of inspection, lubrication, adjusting, and tightening. Its aim is simply the keeping or maintaining of a vehicle in a condition as close to the original as possible.

All moving parts of the truck must be reached for this work. If the man doing it has sufficient mechanical ability to permit him (while doing lubrication work) to observe those things which, if corrected, will prevent breakdowns and the corrections are made, preventive maintenance immediately

General

INSPECTION REPORT

Your inspection check sheet is to be with you at all times during your work and the notes made as the work progresses. Under no circumstances is an attempt to be made to fill out an inspection report after the work has been completed.

Instructions on Use of Chart "A"

Using Chart "A" it is best first to make the physical inspection, doing only minor adjustments or repairs at this time. After this inspection has been completed, the mechanic should do the major jobs which he has found were needed during the first half of the inspection. The mechanic should remain on the job as much as possible and any errands which are necessary should be done by the helper.

In explaining the defects found in the vehicle during inspection it is not necessary to write out the whole item – it can be designated by the number only. Under the number explain the situation in two parts: first, the condition found and the cause, if it is known, and second, the correction which was made or the plans that are made to correct the trouble.

No. 12. *Example* – Brake will not hold, ratchet worn – natural wear.

Installed new ratchet and pawl.

No. 17. *Example* – Broken glass in door caused by case of oil falling against door.

Issued requisition instructing driver to have glass installed tomorrow on first trip.

BEFORE THE MECHANIC STARTS WORK

The day of the inspection the driver is to put about 25 gal. of Rayolight or fuel oil in No. 1 compartment with all valves closed. This oil is to be used in the test of valves, manifolds, and pump by the mechanic. On trucks without pump, we should use White Flash.

Drain all fluid out of meter and syphon breaker also read item No. 80.

MATERIALS AND PARTS

Determine if the parts or material which are going to be needed for the job are on hand and are in proper order. Any parts or material which have been shipped in previously are to be opened and checked carefully to see that they are correct.

UNUSUAL DELAYS

It must be borne in mind from the beginning of an inspection that, where a case arises in which the truck is expected to be held out of service longer than a usual major inspection, the District Office is to be notified at once so that it may plan its deliveries in accordance. Where it is possible, let the District Office know approximately how much additional time will be required to get the truck back in service.

WARNING – CARBON-MONOXIDE GAS

If you are going to work on the vehicle inside of a building, make sure of proper ventilation while the engine is running, with doors or windows wide open and kept open for at least 5 min. after the motor is shut down.

Instructions

No. 1 – *Driver's Report* – The first thing, get a verbal report from the driver on the operation and condition of his unit. In the space provided note the remarks the driver has to make about his equipment. CHECK THE LUBRICATION REPORTS FOR ANY MENTION OF MECHANICAL DEFECTS. These lubrication tickets are to be attached to this report and forwarded to the office.

No. 2 – *Exterior Condition – Visual Inspection* – Make an exterior visual inspection of the unit, noting on your report any parts which have been damaged, broken, bent, or badly scuffed; paying particular attention to damaged fenders, headlights, bumpers, meter carriers, side can racks, running boards, cross-braces, and damage to the tread and sidewalls of the tires. Examine the fastenings of the license tags and tighten them if necessary.

No. 3 – *Paint Condition* – You will note on your inspection sheet whether the paint condition in general is good, fair, or poor. In event the condition of the paint is poor, you will specify what type of job is needed, No. 1, No. 2, or No. 3. After the mechanical work and tests are completed, the paint work on the truck should be touched up as far as it is practical to do so. This touch-up of the paint work is distinctly part of the inspection job. Note condition of gage markings on domes. Touch up if needed. Check condition of porcelain signs and see that they are sealed tightly around the edges.

No. 4 – *Start Engine* – Start the engine, watch the dash instruments, and observe their operation. Listen for noise in the engine and observe the engine's action at idling speed.

Observe the action of the starter when the motor is started up. Listen for any unusual noise in the starter operation, and note whether the starter switch is fastened securely.

Make a check for gas or air leaks in exhaust or intake manifolds.

As the motor warms up, listen for any unusual noises and investigate their cause. Anything of importance should be noted on your report.

No. 5 – *Choke* – Inspect the condition of the choke assembly and its fastening. Make sure that it opens and closes properly. Observe its operation.

No. 6 – *Driving Lights* – As the helper operates the light switches, the mechanic is to inspect the lights. On the driving lights make sure the brackets and lamps are fastened securely. Test the wires and connections for looseness and condition. Note whether the lights have been bent out of line. Examine the condition of the reflectors and lenses. Replace any defective bulbs.

No. 7 – *Signal Lights* – Tighten the brackets and fastenings. Inspect the wiring and connections for looseness and condition. Replace any burned-out bulbs or broken lenses.

Have the helper operate the tail-light and stop-lights.

Tighten the tail-light and stop-light brackets and lamps. See that they operate properly and that there are no broken lenses or defective bulbs.

While the lights were being tested, the condition of the light and dimmer switches should have been observed. Particular attention should be paid to the dimmer switch; for, because of its location, it is very likely to have been damaged by water or dirt. See that it is clean and in proper working order.

No. 8 – *Horn* – Press button to check for sound. See that the horn bracket and fastenings are secure. Examine the button wiring and connections for condition.

No. 9 – *Spare Bulbs and Fuses* – Inspect the spare bulbs and fuses and replace any which are lost or missing.

No. 10 – *Fire Extinguisher* – Remove the fire extinguisher from the bracket and test its operation. Refill it if necessary. Examine the condition of the fire extinguisher bracket and the fastenings.

No. 11 – *Clutch and Brake Pedals* – See that the clutch pedal works smoothly and has the proper clearance at the floorboard. Operate the brake pedal and test for travel and clearance. Inspect the cross-shaft and linkage for condition and wear. See that none of the moving parts bind or rub against the other parts of the truck.

No. 12 – *Emergency Brake Lever* – Test the emergency brake lever for holding and operation. Examine the condition of the ratchet and pawl. Assure yourself that there is no possible chance of the emergency brake lever slipping on the ratchet and releasing the brakes.

No. 13 – *Gear-Shift Lever* – Operate the gear-shift lever. See that it works smoothly and that there is no lost motion in the linkage.

No. 14 – *Power Take-off Lever* – Operate the power take-off lever and examine the linkage for operation and condition.

No. 15 – *Windshield Wiper* – A couple of drops of light oil is to be put in the intake port in each windshield-wiper housing. Examine the blade and arm for condition. Replace the blade if necessary. Test the windshield wiper for operation.

No. 16 – *Rear-View Mirror* – Examine the condition of the reflectors and the fastenings of the rear-view mirrors. See that it is possible to adjust them to enable the driver to see in the rear of the truck.

No. 17 – *Glass in the Cab and Window Lifts* – Check all the glass in the cab. Have any broken glass replaced. See that the glass is held securely in the frame so that it will not rattle or be damaged.

Operate the windshield adjustment arms and see that the fastenings are tight and that the windshield can be opened and closed properly. Raise and lower the windows and test the lifting mechanism.

Fig. 10 – Explanation of the Items of Fig. 9

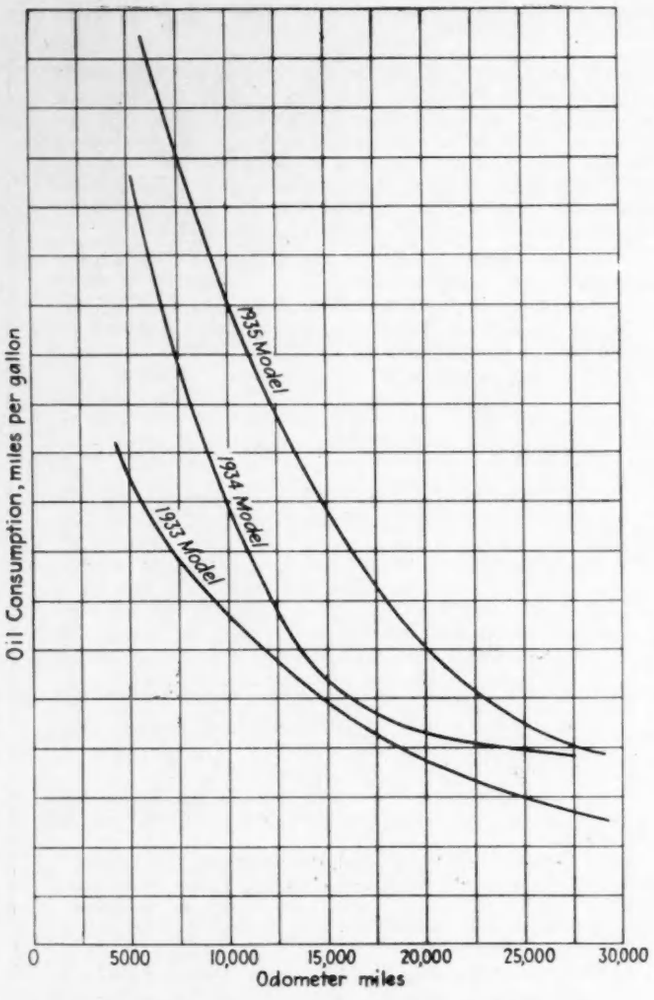


Fig. 11 – Improvement in Oil Consumption with Succeeding Models of the Same Make and Type of Truck

begins to be a reality. He may or may not personally make these corrections. This decision will depend upon the size of the fleet and the mechanical set-up.

Regardless of individual methods employed in lubrication schedules, its fundamentals must necessarily be regularity, uniformity, and thoroughness.

I have thus endeavored, within the space allotted, to answer the "why" of the factors that have a direct bearing on the economic "Maintenance of a Concentrated Fleet of Large Trucks." You have observed that the motive behind each of the governing items has been a collection of facts and figures. With such information at hand, the matter of guesswork, if any, is reduced to a minimum, and we feel that guesswork is not consistent with economic maintenance.

Modern Conditions Demand Modern Trucks

(*Continued from page* 122)

is empty under conditions where the rear springs are at minimum deflection. Admittedly this is an explanation in very general terms of what has been experienced and is not a complete and technical analysis.

The statement has been made that, with short wheelbases, riding and stability are impossible. It is true that, where the wheelbase is reduced to the limits controlled by other factors such as driveshaft length, and so on, it is difficult to get as satisfactory results as with relatively longer wheelbases, but we have been able to get acceptable results with wheelbases under 100 in. by careful attention to springs and steering layout.

Although the increased weight on the front wheels increases the effort needed to steer, this factor is compensated by the fact that the wheels do not need to be cut as much for a given turn and the greater weight on the front wheels makes steering more certain particularly on low-traction roads.

For the same reason and further because there is a smaller percentage of weight change from loaded to empty, a greater amount of brake effort can be applied to the front wheels without risk of encountering front-wheel locking. The weight transfer on the shorter wheelbase truck is, of course, proportionately greater as, in general, there is no difference in the height of the center of gravity, but this consideration need not create any difficulty that can not be met by a corresponding distribution of the braking effort. In this instance as well, the greater weight carried on the front axle helps.

Accessibility is a matter of comparison and, in some cases, the comparison is favorable and, in others, unfavorable but, in all cases, the various operations are accomplished by different means of access. For the ordinary routine daily service operations there is little difference. For preventive maintenance inspection of cylinders, valves, bearings, gears, and so on, there are also about the same time factors involved. Repairs requiring the removal of the powerplant will take more time, but the added cost for this operation in a well-built unit will certainly not detract materially from the economies provided by the other features. It is, of course, possible to provide removable cabs or cradle-mounted powerplants which will reduce this feature to a completely comparable basis, but experience has shown that such treatment is not required after operators have become familiar with the situation but, on the other hand, they prefer the simpler, more straightforward construction.

The area of clear vision which is available to the driver is a matter which has received considerable attention recently. As an ideal there should be nothing to interfere with his seeing in any direction. Obviously such an ideal is as remote from attainment as are most other ideals. Therefore, comparisons are the best basis for determining the relative degree to which different arrangements approach a practical standard. In Figs. 8 and 9 a comparison of a modern passenger car, a conventional truck, and an engine-under-the-seat truck is illustrated. It will be noted that there is little difference between the two trucks and the comparison between them and the passenger car is certainly not unfavorable to the trucks.

Appearance is an intangible matter that cannot be treated by the spoken or written word in a manner which is at all clear. Therefore, a few illustrations will be used to support the statement that the appearance of this type of truck is not only distinctive, but also pleasing, and that it couples attractive form and line with its other valuable features.

The engine-under-the-seat and C.O.E. truck is a logical development providing features and operating characteristics which are needed to meet modern operating conditions. The satisfactory development and application of these features result from the development and technical improvement in engines, steering apparatus, and tires. Increasing density of traffic and regulations require these features.

These are the basic assumptions upon which we established the development of this type of vehicle, and the acceptance which the type has received furnishes conclusive evidence that it is filling a real need in the transportation industry.

Synthetic-Resin Enamel Finishes

By J. L. McCloud
Ford Motor Co.

FIRST of all – what are synthetic-resin enamels? They are high-luster paints composed of pigment, solvent, and binder or non-volatile vehicle made of synthetic resin. The pigment or solid coloring matter determines the color. The solvent or thinner determines the successful application of the enamel. First, turpentine was used as a thinner and then petroleum spirits; but now it is made of alcohols, esters, synthetic hydrocarbons, and special cuts in the distillation of petroleum, which composition improves the solvent power.

In describing synthetic resins, we must first define a resin as a solid organic body, transparent or translucent, insoluble in water but soluble in organic liquids such as alcohol, esters or hydrocarbons. Formerly they were of vegetable origin, but recently organic chemists discovered how to synthetize them. The raw materials are not resinous at all. The most widely known synthetic resin, Bakelite, is made from phenol and formaldehyde.

Indeed, we may say that these enamels are composed of three parts new to their fields. We find that new pigments, both synthetic and natural, new oil resin vehicles, and new thinners – are all used to achieve the original beauty and the lasting quality of these synthetic-resin enamels.

Probably many of you know the history of automobile finishes. In this part of car-building, as in others, the horse carriage was copied or followed. So the first cars were primed with lead and linseed-oil paints, then glazed and surfaced with many coats of slow-drying, heavy-bodied filler paints, many of them containing lead. These paints were "sanded" or blocked down with stones to a smooth surface, and color-coated with natural resin enamels containing the pigment colors as required and, finally, they were "finish"-varnished with a "long" oil "short" resin coat. This finish was beautiful on the floor of the factory, but it took time measured in weeks to produce and, worst of all, its life also was measured in weeks.

The first improvement was in the use of nitrocellulose lacquers. The change here was to a finish applied in hours rather than in weeks and its life was measured in months rather than in weeks. Basically, the layers of finish applied were the same, except that the final clear or transparent coat was eliminated and the final coat of lacquer enamel was polished. Polishing was necessary because the film surface was dull and, incidentally, rough, as I shall describe in more detail later.

The latest change in the finish of cars is to synthetic-resin enamels. Improvement again is obtained in time of application, the saving being mostly due to the ability to apply at one time more substantial coats and the diminished need for polishing. But the main improvement has been in the increased durability of the film surface and the underneath layers. The time is still measured in hours to apply, but the life is to be measured in years, rather than in weeks or months, as was done formerly.

In this history I have indicated the essential reasons for the change, but let me repeat some details concerning the last change.

[This paper was presented at the National Production Meeting of the Society, Flint, Mich., Dec. 9, 1937.]

Lacquers owe their characteristic to nitrocellulose. But this material is brittle, hence it must be softened or plasticized.

This plastic material has very poor adhesion, so resins are incorporated. Now, this combination is not very durable in sunlight, checking badly, though it can be made quite stable under glass (as in the interior of a closed car). Fortunately, however, when it is pigmented to form a colored enamel, the pigment shields and protects the very vehicle that binds it together and to the surface that it covers. In a well-made colored lacquer, the film fails by "chalking" or gradual dulling of the surface, which is gradually washed or wiped away. Poorly made lacquers or those made with relatively transparent pigments may check as well.

In synthetic-resin vehicles we have a swing back to oil and resins alone, but this time it is an oil synthetically combined with a resin produced *in situ*. Such a clear varnish is itself very durable but, most of all, if the proper choice of oils and resin is made, the rate of dulling is less than one-fourth of that of the nitrocellulose lacquers that it replaced. Lacquers, to be sure, have been improved so that they far surpass those first used, but they have not been made to equal present synthetic-resin enamels.

The choice of oils in the vehicle of these enamels is a very important one. It may vary according to many contributing, and often conflicting, requirements. Soya-bean oil with the right proportion of china-wood oil has proved best for the main part of the car surface, which takes the largest gallonages. The use of this newest comer to our drying oils has captured the imagination and interest of many industrialists as well as farmers to whom it has become an advantageous crop. Finishes of this type are often referred to as soya-bean enamels.

Also important is the proper choice of a resin as well as the oil. Although I referred to Bakelite at the start as a typical synthetic resin, the proper choice for enamels at present is a glycerol-phthalate. The component glycerine you all know, and the phthalate part comes from coal-tar naphthalene as phthalic-acid anhydride. Neither component is in any sense resinous, but they are synthetized with the soya-bean and china-wood oil to a resin. These resins were made first commercially by General Electric, but it was duPont who so developed them that their use on automobiles and parts was feasible. The contribution of the Ford Motor Co. was in the development of application tools and technique. Others have followed suit, and we in the Ford Motor Co. make millions of gallons a year for our car finishes, aided by E. I. du Pont de Nemours & Co., Inc. Fig. 1 shows the charging floor in our Synthetic-Resin Plant at Highland Park. Here are combined the soya-bean and china-wood oils with glycerine and phthalic anhydride.

In the study of any finish, no information is as valuable as that concerning its durability in actual outside exposure. We have hundreds of panels continuously on exposure in Florida. Incidentally, we conduct exposure tests elsewhere, but we find that, almost universally, Florida exposure is worse than Texas with its hot dry valleys, or than the coasts of California, and far, far more severe than Michigan. It is at least equal to

exposure in the Canal Zone. In such exposures we consider that a finish should withstand two years with no more dulling than can be restored with a liquid body polish. The original dulling should not set in noticeably in less than six months. The backs (which are primed) of our panels (which panels are faced south and inclined at an angle of 45 deg.) should be free from rust in a year, and no more than a few small spots can appear in two years. Fig. 2 shows the manner that exposure panels are weathered.

From this discussion you will appreciate the standard of quality that we maintain, which accounts for the outstanding appearance of two-, three-, four-, and five-year old Ford cars on the road. Such a standard naturally requires time to evaluate new products, but eliminates those whose beauty is short-lived.

The surface durability or rate of dulling of an enamel varies with the color. Black in any vehicle is the most durable "color," and whites or tints of white are the first to dull. But, even in the field of blacks, there are differences, and we have found blacks that are outstanding in their very low rate of dulling and in their color. White pigments are no longer lead, but titanium-oxide for opacity and antimony-oxide for low rate of dulling. In the field of greens, many of these pigments can be made of yellow oxides of iron with white-and-black (which makes bluish grey) rather than chrome greens, with marked improvement in color retention or lack of fading. Yellows made synthetically as organic pigment are very permanent. Blues, when dark, are made of iron cyanides and, when in light tints, are made very permanent in organic lakes or color pigments. Reds in former finishes either bright or dark – as maroons – were so poor in durability as well as so low in opacity as to be shunned by the car buyer and car producer both. We now have, however, maroon pigments with a life equal to any color and with such opacity that the paint-shop foreman finds maroon as easy as any to apply.

In the application of any finish, smoothness or flow-out is the greatest bugaboo. When finishes were brushed on, the flow was slow but so, likewise, was the drying – hence a smooth job could be obtained by experts. With finishes like lacquers that dry so fast by mere evaporation, applications by brush were supplanted by the air spray gun, which breaks up a stream of finish with a jet of air which at the same time, carries it to the surface. Such a method naturally deposits the finish in droplets that must coalesce, otherwise a rough or "orange-peel"-like surface is produced. (Other factors of shrinkage make this condition even worse.) Generally it may be said that the faster a spray is applied and the finish dries, the more the orange-peel effect, but here again other factors may make the choice fall on a relatively fast-drying finish due to the thinner type.

The surface character influences the appearance of the orange-peel finish and, in dull lacquers, one is often fooled. We find that a properly applied synthetic-resin enamel flows out to a smooth finish with a satisfactory luster. For this reason, we do not sand off or even polish the surface of Ford cars, and hence we retain all the finish for the protection of the car body as well as for its ornamentation.

Our use of synthetic-resin enamels dates back to about 1928 when they were used first on wooden truck parts and on wire wheels. This use continued with only minor additions until, in 1932, we began to finish cars with it experimentally. The first jobs finished were trucks, panel jobs, and cabs. In the fall of 1932, we finished a few passenger cars and, beginning with the 1933 model, we finished standard cars. The middle of that year, the deluxe cars were added and, from thence on, all Ford cars have been finished in this way.

This year we are finishing Lincoln-Zephyr cars with synthetic-resin enamel. It is our belief that lacquer, as we have known it, will follow the old color varnish to the discard.

In addition to the advantages I have just cited, another one is important. This is the ability to finish fenders and sheet-metal parts in color in these enamels. In that way the equipment used in production for black-japanning these parts is usable.

The air in the spray room and ovens must be very clean. This condition is especially necessary for baked enamel that is not sanded or polished. In fact, it is our belief that the very ease with which lacquers could be repaired as compared with that of the old color varnish, turned out to be its principal sin. By that statement, we mean that operators became so used to expecting the cars to be repaired or "touched-up" that the incentive to perfect workmanship did not exist. If you consider that each time a car body is "touched-up" it must have been a reject, then it is usual to have 100 per cent rejects in a lacquer shop.

If any other factor that enabled us to succeed with synthetic-resin enamel is to be named, it is the will of the Ford management to produce car finishes right in the first place so that repairs are not necessary. To be sure, cars may be marred on the final line, and repairs thus made necessary. We can do this repairing, and with little more difficulty than with lacquer.

The actual application of the finish to car bodies is described on our sheets of "Operation and Specification Sheet for Painting." I shall not present all these details, but will note in order the important steps and the precautions to be taken.

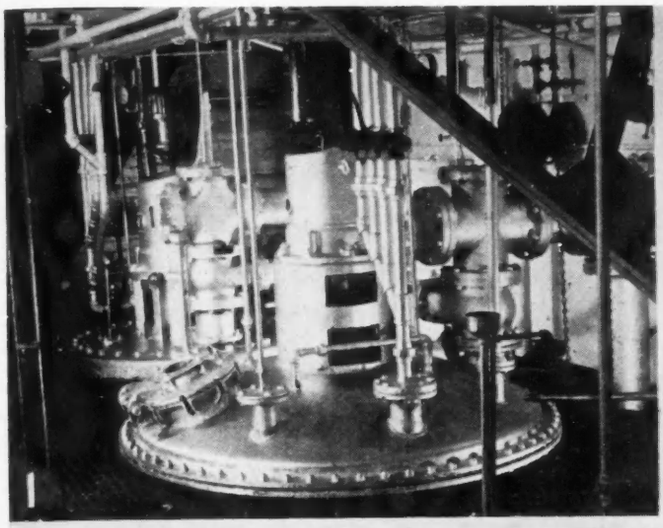
First, as to colors obtainable in these enamels, any that are possible in lacquer and many that are poor in lacquer are satisfactory here. Metallic or iridescent finishes are difficult in lacquer and in enamel, and are to be avoided. Naturally, the finishing of a car in one solid color is easier for the manufacturer. It also enhances the streamline bodies of the present model. However, we have finished many commercial cars in two- and three-color effects using enamels.

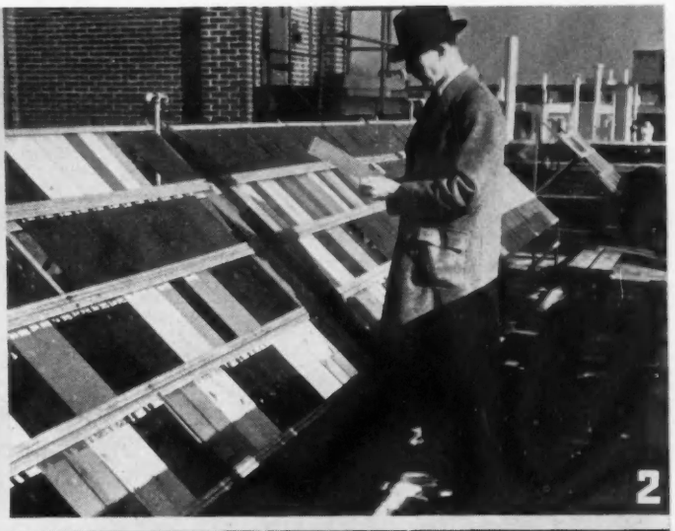
In preparing enamels for application, the thinning is done mostly with hydrocarbons rather than the mixtures containing the more expensive esters. To be sure, we have used butyl alcohol mixed 40-60 with a petroleum hydrocarbon and ethyl acetate mixed with petroleum hydrocarbon. There are now in use high-solvency hydrocarbons produced from petroleum that are economical and satisfactory.

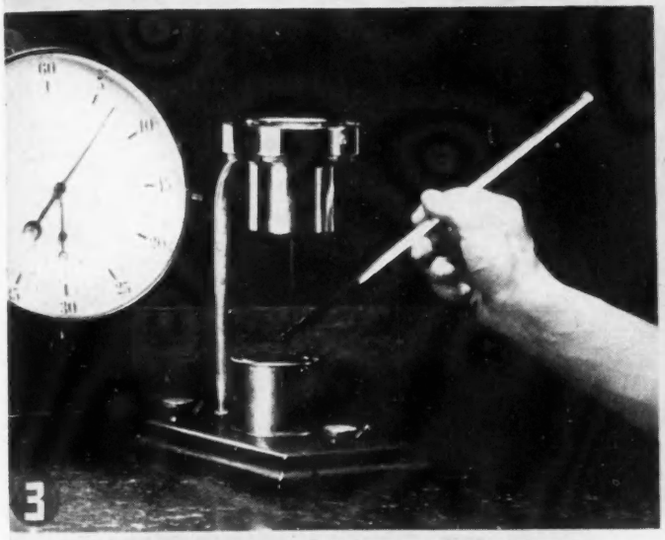
When we started the use of enamel, we drew up our operation for thinning based on viscosity. In fact, no record or note is made of the gravity. We use here the No. 4 Ford viscosity cup that has come to be widely used in the paint industry. The viscosity used for spray application is from 17 to 20 sec. at 80 deg. fahr. Some enamels are sprayed at viscosities even up to 30 sec., but not usually. Fig. 3 shows the Ford viscosity cup in use.

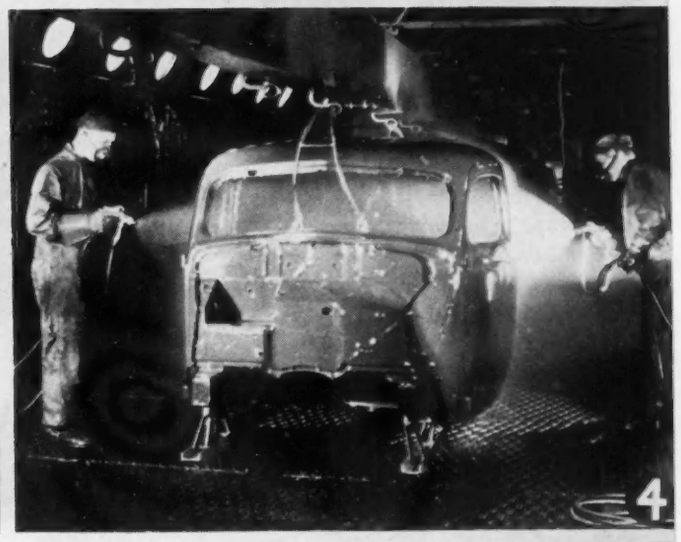
All enamels, after thinning, are strained through two layers of nainsook cloth, then they are transferred to the distributing system. We use a central storage head of about 50 gal. capacity continuously agitated 7 days a week, from which the enamels are piped to all locations in the building in which they are used. The enamel lines are all continuous, and the return is made to the storage head tank. An important point here is to see that the return-line pipe is always below the surface of the liquid in the tank. The pressure in the lines as they leave the tank is 50 to 90 lb. per sq. in., depending on the individual installation, and the return-line pressure at the tank is from 10 to 20 lb. per sq. in. This arrangement enables pressures at the spray booths to be held at from 15 to 20 lb. per sq. in. Another precaution taken is to strain the enamel through nainsook continuously. The nainsook strainer bag also must be kept under the surface level.

In addition to the precaution of cleaning the air supplied











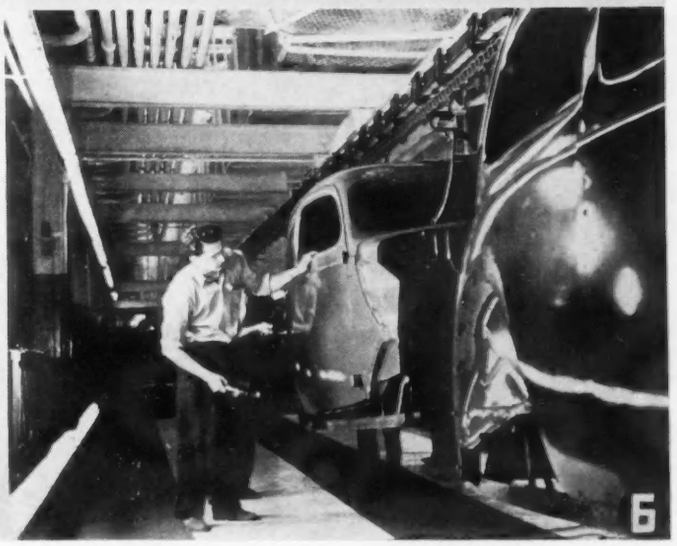


Fig. 1 – Charging Floor in Synthetic-Resin Enamel Plant

Fig. 2 – Method of Weathering Exposure Panels

Fig. 3 – Ford Viscosity Cup in Use

Fig. 4 – Spraying of Body

Fig. 5 – Applying Primer Surfacer

Fig. 6 – Ding-Out Line

to the spray booths, the walls and floors are covered with an oil, soap, and water mixture. The interiors of the ovens are coated with castor oil. To speak of air furnished to a spray room may sound unusual but, to maintain a clean room with no chance of infiltration of dust-laden air, we actually supply more fresh air than is exhausted through the stacks. This condition maintains a small positive pressure. Our exhaust stacks are constructed to draw the air through a water spray. In this way there is no accumulation of paint fumes on the outside of the building. The water used to wash this exhaust

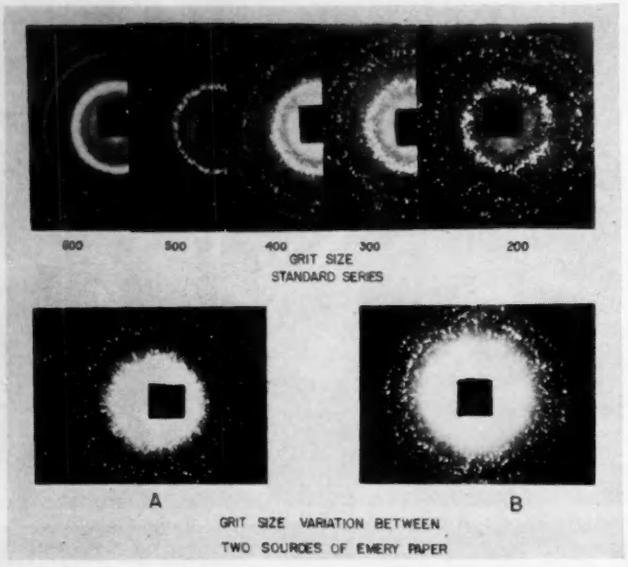


Fig. 7 – X-Ray Diffraction Analysis of Grit Size

is alkaline so that there is no plugging of lines or pumps. Fig. 4, illustrating actual spraying of a body, shows especially the excellent ventilation in a Rouge Plant spray room. The exhaust is through the floor grille.

The bodies in white are prepared for finishing by washing with oleum spirits, bumping out and metal-finishing very thoroughly. This preparation is especially necessary to keep the repairing of the bodies in the final high-luster coat to an absolute minimum, and since this high luster makes metal-surface defects more noticeable.

The bodies are then washed with a phosphoric-alcohol type of cleaner. For this operation, we have a circulating pump, and the brushes are supplied continuously with the acid mixture. In this way we can not only scrub the bodies, but they are flushed off at the same time.

After the acid scrub, the bodies are flushed well with water and steam and enter a dry-off oven. This oven must be maintained at a temperature above 225 deg. fahr., for about 20 min.

Bodies are then metal-glazed. This material is of the "oil" type thinned to a soft knifing consistency, and is used over weld areas and areas that are more heavily finished. It is then air-dried about 5 to 10 min. and sanded off.

The next operation is application of a solid coat of primer surfacer. A great deal of work has developed a primer that may be applied so as to finish over 0.002 in. in the dry state and not sag when wet. This primer is of the "oil" type. It is interesting to note that, for satisfactory water impermeability, oil-type primers are preferable to synthetic types for use under synthetic-resin enamels. The reverse is true in the case of lacquers. Fig. 5 shows this operation. In this case the exhaust is on the side.

On leaving the prime oven, the primer surface is sanded to a smooth finish. This operation may be done either dry or wet. Care must be taken to use a fine-grit paper. Inspection at this point may disclose some deep metal scratches or pits, in which case they are then glazed with a pyroxylin putty. If water-sanded, the body must be dried off through an oven and any glazed areas sanded smooth.

A solid first-coat enamel is then applied and baked for 1 hr. at 250 deg. fahr. or, alternately, for ½ hr. at 325 deg. fahr. In the latter case, a color that changes least in such high-temperature baking is the best choice.

The body then is "dinged-out" very thoroughly and further pyroxylin-glazed. Much stress should be made here to see that the bull's-eyes and dings and scratches are caught at this point. If they get by here, they will have to be entirely re-sanded in the final enamel and re-run. Fig. 6 shows the ding-out line, and here may be seen the lines of lights arranged to yield highlights on the body for ideal inspection.

The final sanding of the body follows next. This operation is, of course, wet-sanding, and a proper choice of paper must be made. When we first started this operation we used 320 paper. Then we had the sandpaper companies develop one that had no finer grit but was graded so that the coarser particles were eliminated. This is called a 360 paper and is used over the body, except in some areas that are not prominent where coarser paper may be used. In some areas of this sanding, electric or air sanders are used successfully. Fig. 7 shows how abrasive papers may be checked for size of grit by X-ray examinations. A small X-ray beam that is diffracted from the abrasive crystals without removing the grains from the paper base, forms these pictures. The size of the spots registered on a photographic film is compared quickly with a standard grain-size series.

Again some pyroxylin glazing may be done after washing of the body, and the water must, of course, be dried off. If any areas of bare metal occur here, they are re-primed, dried and re-sanded. This occurrence should be very rare if proper precautions are taken earlier. In fact, emphasis placed on actual care of the work in its proper place has been the reason for our success in this synthetic-enameling of cars.

Before the body is "tacked off" to enter the final enamel spray booth, it is washed off with a light cleaner's naphtha or white (that is, non-leaded) aviation gasoline.

The final enamel now follows. In this operation a great deal of the success depends on the technique of application. The trick is to apply the enamel in two operations so that the body has the right amount without any "dry spray" or sags. In fact these are the two rocks that must be avoided continuously. On the one hand, a good heavy coat levels better and the appearance is better, although too much enamel may lead to sags or crinkles. Also, if too much enamel is applied, drying difficulties are encountered and the bodies may be tacky. On the other hand, if too little enamel is applied, there may be dry spray areas, the finish is not smooth, and any scratches in the undercoat will show up; also, the body will be dull although it will bake out hard.

We have found many things to watch for and have changed compositions somewhat to correct these results. We have found, too, that conditions not heretofore considered objectionable, may be very vital. For instance, the heat distribution through the oven is very often neglected with a material having considerable baking latitude.

With synthetic-resin enamels, we have little baking latitude. We must see to it that the temperature through the ovens is actually 240 to 250 deg. fahr. This job is done by hanging recording thermometers on the conveyor at the top and bottom of the oven as well as at the center and by being sure that there is no more than that temperature difference. There may be also variation in temperature from the side to the center to watch for, although this variation is rare.

Some correction or aid in solving this problem of latitude may be obtained in the choice of thinner. In general, it can be said that a slow thinner will level better, but a fast high-solvency one may be the choice by enabling more enamel to be applied. Greater resistance to crinkle is attained by small additions of drying inhibitors and to a tendency to tackiness by small metallic drier additions. But here, too, the medicines must be used very judiciously and are best made in the paint manufacturing plant once the conditions to be encountered are realized fully. The success of this work is, in no small measure, attributable to close cooperation between paint manufacturing and application, divisions and companies.

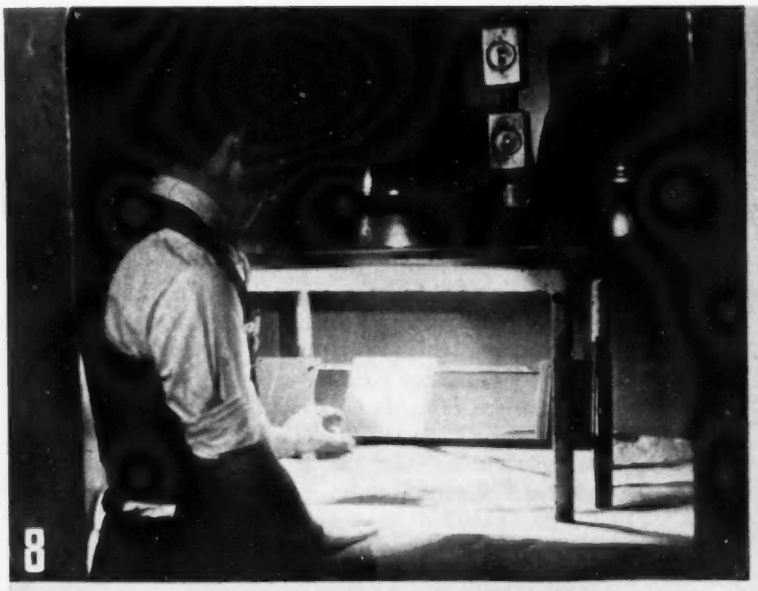
We have learned, for instance, that parts or bodies must not be chilled after application of the enamel and must be placed immediately in ovens to bake. We encounter craters or pop-ups of the enamel especially over solder-finished areas and, incidentally, they occur mostly in certain types of colors. These faults we can minimize but not entirely eliminate. They make re-runs necessary.

To digress a moment, the susceptibility of colors to temperature variation is an interesting problem that seems to be due in part to the method of dispersion of the pigment as well as to the composition of the enamel as a whole, but successful compositions are obtainable. Even satisfactory light blues made from the extremely durable organic blues are successful. Maroons of high purity of color or chroma may be made either high (light) or low (dark) in value. These maroons can be so good in hiding qualities as to be satisfactory without ground coats and to be durable for two years on Florida exposure panels. In fact, in this connection, a new conception of quality is possible with this color which has heretofore been considered a necessary evil.

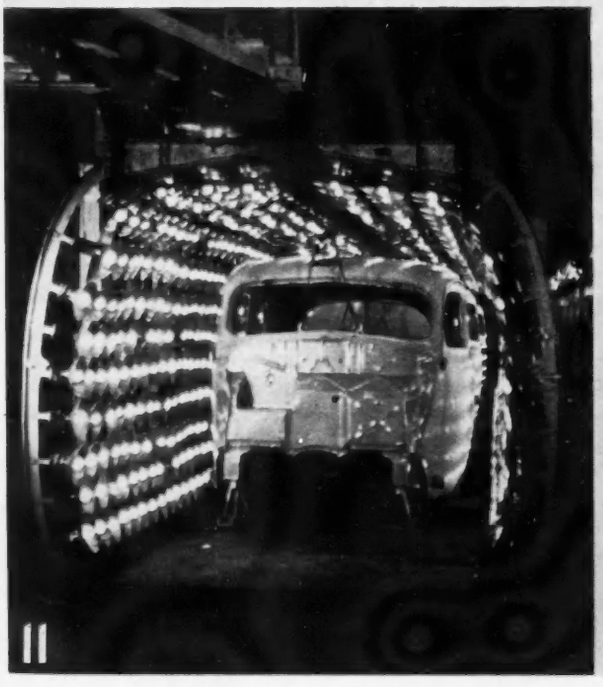
Returning to our body that is coming out of the final enamel oven, it has been baked at 250 deg. fahr. for 1 hr. and, after final inspection, it is turned over to the trim line. Along the trim line, we have again set up new standards of care so that mars and scratches that came to be tolerated with the more readily repairable lacquer just do not occur.

Stripes, if used, are of the pyroxylin type and are applied by tools that give substantial and durable qualities.

This description of the application of synthetic-resin enamel to the surfaces of bodies may serve as well for its use on other car parts. But there is one notable exception, namely, dip application. In fact, we consider the successful dipping of sheet-metal parts as outstanding as the enameling of bodies. It was formerly considered impossible to dip successfully colored enamels. For years, to be sure, sheet-metal parts have been finished in black japan. This material is sometimes re-







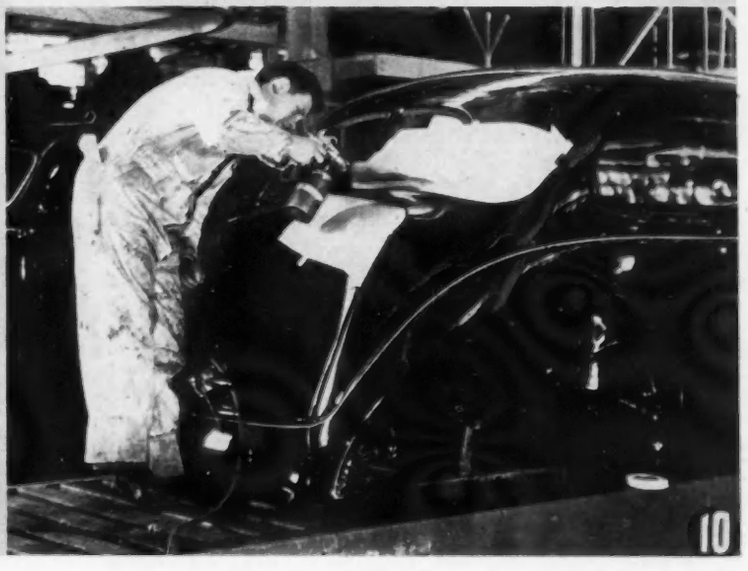


Fig. 8 – Color-Matching by Special Daylight-Intensity Lamps

Fig. 9 – Removing Enamel-Dipped Fender from Tank

Fig. 10 – Repair Spray on a Spot

Fig. 11 – Light-Bank Set-Up for Baking Enamels

ferred to as black enamel, but the finish coat owes its color not to pigment as such, but to the natural black bituminous resin, Gilsonite.

Small parts in colors and those without high-quality requirements have been made, but here we have succeeded in a high-luster coat free from objectionable silking or streaking of color. To the accomplishment of this achievement, several factors have contributed. The first is the quality of grind of the pigment in the resin vehicle and the proper choice of the pigments. This latter item is especially important in the case of greys and those that are greyed colors. Those of you who do not manufacture paints would be surprised at the amount of black and white additions necessary to make the softer, less vivid colors that we find to be more attractive as automotive finishes. The second factor in diminishing silking is grinding as a composite mill base all of the pigments to be used in any one color. In practice we blend or tint enamels to produce a certain shade and then, by breakdown calculation, determine the base formulation. A control used in this same connection is to test each batch of enamel by a dipped panel as well as a sprayed panel. These panels must match each other as well as match the standard color panel.

This dip-finishing of sheet-metal work, since these parts are of the same color as the sprayed bodies, necessitates a very close tolerance limit on the color. To make accurate color-matching possible, artificial-light sources of daylight "color" are necessary. When we started the color-matching, lights were of such low intensity of illumination that better types were found necessary. In this development we assisted, and we now have a light source of noon sunlight color as well as "horizon sunlight," that is, as searchingly intense as a bright sunny day outside. Fig. 8 shows these lamps in operation.

In order to dip fenders, a tank of about 400 gal. must be used. We finish fenders as well as bodies at 16 locations in this country alone and, where the demand for a color is low, some of the enamel stays in the tank for some time; so we have to stabilize the enamel or it would set up or form a jell structure in the tank. Inhibitors or anti-driers minimize this tendency and are used regularly. Naturally, we have to have a proper dry in the ovens later so a careful balance of added metallic drier and anti-drier must be made. See Fig. 9.

In the case of black, we have large demands, and here we continue the use of large automatic dip tanks which hold between 3500 and 4000 gal. These tanks are of the vee type and there is a continuous circulation of enamel from the bottom to the top. Strainer bags are in use continuously, and, as noted before, are kept below the level of the liquid in the strainer box. The tanks holding the color enamels also are equipped with circulation and straining. These tanks are located on tracks so that, with two or four lines of color ovens, there can be several colors in use.

Sheet-metal parts are all primed with an extremely fine-ground primer in an automatic tank and baked an hour at 350 deg. fahr. This primer is mossed off to remove any nibs and given one dip coat of enamel. The thinners used are either the butyl alcohol-oleum spirits referred to before, high-flash naphtha, or high-solvency petroleum naphthas, according to the color and exact oven conditions.

The viscosity is somewhat higher here than for spray application by about 2 to 4 sec. in any color. And this factor, combined with the rate of flash-off of the thinner, gives about the same fender thickness of enamel as with the bodies. The fender enamel is baked an hour at 250 deg. fahr.

Probably no subject is more interesting than service experience, but about which so little can be said. The appearance of the Ford cars so finished can be observed daily and compared. Color for color, they are outstanding.

Repairing of enamel finish was, at first, one of the greatest obstacles to its general use. In an industry where the units finished are bulky and must, of necessity, be delivered without mars, we had to evolve a repair system for enamel that would compare in time to that of lacquer. We accomplished this purpose by using these same enamels, in some cases, with additional metallic driers (usually cobalt). In drying a repair, heat is furnished by means of infra-red electric light. These are carbon-filament lamps of the same type as those used by physicians. Our assembly lines are all equipped with banks of these lights in various sizes, adjustable to any body location.

The area to be repaired is sanded to be sure of adhesion of the repair coat to the hard baked finish. This sanding is done with either 4-5 or 600 paper depending on how large and deep it must be done to correct the fault. Enamel, thinned in this case with a lacquer-type, fairly fast evaporating solvent, is sprayed on. The heat lamps are then arranged at about 12 to 20 in. away, and the baking is accomplished in from 5 to 20 min., depending on the size and color, and so on. After the repair is dry it is polished carefully, especially at the edges. Fig. 10 shows a repair spray on a spot.

Naturally, skill and practice are necessary to avoid color-float or overheating. Either of these troubles will cause an off-color spot. In repair work, wherever feasible, an entire panel is refinished, which job naturally does not require the skill necessary for spot repairs. Attempts to use lacquers, even of the present most improved type, as actual patch materials soon show the superiority in weathering of the enamel.

In service garage repair work the same general methods preferably are used. Often, however, the time necessary to dry such repairs which, of necessity, may have to be made from the metal up, precludes the use of anything but lacquers. Here the finishing with semi-synthetics is the practice.

The success of baking enamels with lights has led to their use for operations formerly done with steam-heated ovens, as is shown in Fig. 11. At the Rouge Plant we manufacture our own power at a low rate and have a very unusual prime drying "oven."

This oven consists of a series of double clam-shells carrying lights spaced so as to heat the bodies and bake the prime in 7 min. instead of the usual 1 hr. The bodies travel on an overhead conveyor. This conveyor carries them between two synchronized conveyors that carry the clam-shells of lights. The temporary assembly of lights and bodies travels together for the 7 min. and then the body emerges with the lights traveling back to surround another body.

Each bank or shell of lights is equipped with a time switch so that, if the conveyor is stopped, only 7 min. of baking occurs. If the lights should be left on too long, an extremely hard surface would be obtained, or the solder used over the weld area would sweat out.

Other applications of this method of heating are in the form of light tunnels to dry off water after the acid wash and wet-sanding. In this case switches are arranged so that a wave of light accompanies the body through the tunnel, which method obviously reduces the power load.

Since the radiant heat from the lights heats the metallic body which, in turn, heats the enamel, the heat loss in the room is remarkably low. (See Fig. 11.) In fact, there is no insulation provided about these lights. In the case of the primer "oven" a ventilation stack is provided, but the room temperatures are normal even in the hottest weather.

This method of baking finishes is, in fact, very economical and, in our Rouge Plant, is way under the cost of steam ovens.

We believe that the future will see more of these installations, as well as further reductions in the time required to bake enamels, and that the increasing use of enamels will bring the development of better synthetic thinners.

The Reduction of Piston-Ring and Cylinder Wear

By Macy O. Teetor
The Perfect Circle Co.

THE elimination of wear of piston-rings and cylinders can be the ultimate goal toward which to strive but, in reaching this Utopia if it can be reached, the most practical road seems to be by way of wear reduction.

Many factors indicate the necessity for a "wear-in" period. At some point in service wear-in ceases and "wear-out" starts. As wear-in takes place, performance only improves to a certain point and, from there on, piston-rings and cylinders can be considered as wearing out.

The rubbing action of a piston-ring on a cylinder wall breaks particles loose from the surfaces that act as an abrasive. This breakdown of the rubbing surfaces, regenerative because of the abrasive action of the resulting loose material, causes wear. The ease with which the surface of a material will break down and the physical characteristics of the loose particles so produced are indicated to a great extent by structure.

The structure of a material is therefore an indication of expected wear.

THE wear of metals has received so much attention for so many years that even a brief review of the published works would require too much time to combine with this presentation. The subject was reviewed very thoroughly, and much new available material was added to the existing fund of knowledge at the Symposium on Wear of Metals held in Philadelphia last summer by the A.S.T.M. Anyone deeply interested in this subject should not fail to read the papers presented at that Symposium.

"Wear-In"

The definition of "wear" that we choose to use in this presentation as most applicable to piston-ring and cylinder wear is by Louis Jordan[1]: "unintentional removal in service of the surface of a metal through the action of frictional forces." Even this definition is only partially applicable to piston-ring and cylinder wear because piston-rings and cylinder surfaces still must "wear-in." So-called wear-in is not necessarily intentional wear but a type of wear that has always existed and been considered a necessity. The perfect piston-ring and cylinder has not yet been made and, from what is known about the difficulties involved, it may be some time before this feat is accomplished. Initial wear of a not-too-destructive type is the great compensator for inability to eliminate some of the difficulties involved in producing the ideal piston-ring and cylinder combination. A certain amount of wear-in compensates for mechanical and thermal distortion and results in improved performance. As wear-in takes place, performance only improves to a certain point and, from there on, piston-rings and cylinders can be considered as wearing out. The problem is, therefore, not one of wear prevention, but one of wear reduction throughout the entire life of an engine.

Metal-to-Metal Contact

Metal-to-metal contact occurs between the bearing faces of piston-rings and cylinders because of the failure of lubrication. The almost universal system of cylinder lubrication makes it difficult, if not impossible, to have complete lubrication between piston-rings and cylinder walls. It may be possible to effect complete lubrication, but there are several problems to be solved before it is a possibility. Surface-finish contour variations must be less than an oil film of sufficient thickness before metal-to-metal contact can be avoided. Even when surfaces are smooth enough to not have projections through an oil film of satisfactory thickness, slight changes in shape due to distortion can create high pressures that will cause more metal-to-metal contact area than would be created with rougher surfaces. Piston design has improved but not to the point where it can be said that small high-pressure areas are not created between the skirt and the cylinder wall, thereby destroying the oil film. Constant minimum clearance of a piston in a cylinder under all operating conditions is still among the desirable and not wholly attained piston operating conditions. Any deviation from the ideal allows the piston to rock the rings off the cylinder wall to some extent which contributes to oil-film disturbance.[2] The cylinder-wall surface temperature in many engines is far from equalized or controlled below the disintegration point of the oil.

In addition to all the difficulties of maintaining an oil film on a cylinder wall, oil consumption is controlled conventionally by scraping the oil off rather than by attempting to control the supply. If excessive oil consumption can be considered

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[1] See *Mechanical Engineering*, Vol. 53, September, 1931, pp. 644-650: "The Wear of Metals," by Louis Jordan.

[2] See S.A.E. TRANSACTIONS, August, 1936, pp. 328-332: "Cylinder Temperature," by Macy O. Teetor.

as indicative of the presence of a thick film of oil on cylinder walls, a thick film is not even desirable from a wear consideration because excessive wear is often accompanied by excessive oil consumption. Much has been said about so-called oil film strength. If the strength of oil films can be increased further, metal-to-metal contact can be eliminated to some extent which, of course, would help to solve the problem. After looking over this brief list of suggested requirements as a basis for improvement in cylinder lubrication, it seemed more practical, at least for the present, to accept the undesirables and to try to reduce and control wear rather than attempt to prevent it.

Abrasive Action

Reduction of wear is possible because we have excessive wear in some instances and very little wear in others under what appear to be "identical operating conditions." Observations over a period of several years indicate that there is a direct relation between the structure of ferrous materials and wear. By structure is meant the physical assembly and arrangement of the elements and their combinations that compose the material. A microscopic examination of the surface of a mirror-finished cylinder reveals some interesting facts. When a sharp-pointed instrument is rubbed over such a surface, using about as much pressure as would be exerted when writing with a lead pencil, the surface is disturbed. Small particles loosen and very easily break out of the surface. Continued rubbing will loosen and remove enough particles so that a cavity can be seen with the naked eye. If a cavity can be so easily formed, it is not difficult to imagine the gentle removal of material from a large area of a cylinder surface.

When a piston-ring is producing the rubbing action, the removal of the surface is accelerated because the small particles that break out of the contacting surfaces either have to cut their way out from between the cylinder wall and ring face or crush to a size smaller than the surface irregularities. These particles actually do a little of each. As they find their way out from between the contacting surfaces, they leave trails in the form of scratches. The depth and width of a scratch depend upon the size of the particle, the pressure exerted upon it, its cutting ability, and the ease with which the cylinder and/or ring material cuts. These abrasive characteristics of the ring and cylinder materials differ with different structures. The type of structure is, therefore, an indication of the ease with which the material will come apart and of the size and physical characteristics of the particles that break loose from the surfaces.

Structure

Untiring efforts have been made to correlate hardness and analysis with rate of wear. So many upsets have been experienced in this attempted correlation that it is reasonable to assume that hardness and analysis indicate an expected rate of wear only as they indicate structure. Ferrous materials having the same hardness and analysis can have entirely different structures. The structure of cast iron is affected by the cooling rate of the casting. Iron from one ladle poured into sections of different dimensions will have different structures. At certain critical temperatures during the cooling period definite changes in structure take place. Alloys change these critical temperatures and, in so doing, affect the resultant structure. Heat-treatment can take advantage of these critical temperatures and rearrange the structure. Therefore, ferrous materials studied for wear characteristics should be classified as to structure in addition to analysis, hardness, and physical characteristics.

[3] See Transactions of the American Foundrymen's Association, December, 1937, pp. 449-458: "Ferrite – Its Occurrence and Control in Gray Cast Iron," by R. H. Bancroft and A. H. Dierker.

It is not necessary or desirable to use all the destructive forces found in the operating conditions of an engine in order to study the abrasive characteristics of different piston-ring and cylinder materials. The wear characteristics of ferrous materials can be studied by rubbing test pieces together and controlling the factors that seem to contribute most to the breakdown of the surface of the material. The results of our first attempt to study wear with a wear machine were very disappointing; however, several important things were learned. The machine was built to test pieces of cylinders and rings because cylinder and ring material structure could be duplicated most easily in cylinders and rings.

Comparative results were desired so the machine was designed to run 12 samples at once. The inside diameter of the rings was bored to the outside diameter of the cylinder. The segments of the rings were placed in holders and pressed with individual weight mechanism against the cylinder so that, when the cylinder was rotated, the inside of the rings would wear grooves on the outside of the cylinder. Wear was determined by the loss in weight of the ring sample and the depth of the groove worn in the cylinder. Oil was carried to the ring samples by the cylinder from a reservoir in the base of the machine. Results from this machine indicated the importance of abrasive action. Although the results were inconsistent and not reproducible, a tendency was noticed for like structures to wear alike. It was thought that the inconsistency of the results was caused by the abrasive action of the material worn from the samples. All of this material went into the oil and, of course, could affect the wear of all the samples. To test this idea one of the cast-iron samples was replaced with a steel sample and the wear of the remaining 11 cast-iron samples and the cylinder increased considerably. It was, therefore, decided that only one combination of ring and cylinder material should be tested at a time.

The type of motion influences the action of abrasives. An expert mechanic uses a certain motion to lap a surface. He follows a definite procedure to secure a definite result. The type of motion affects the flow of oil between the surfaces, and the oil carries the abrasive. Since the major motion of a piston-ring in a cylinder is reciprocal, it was decided that a reciprocating motion would be more applicable than a rotating motion. The experience gained from operating this machine indicated that a more careful analysis of the problem and a different type of machine would be required to arrive at any reasonable conclusions that could be correlated with engine operation. A new machine was, therefore, designed and constructed.

Standard Test Structures

The selection of a standard structure of iron for a test cylinder sample for the new machine was made after examining many cylinders. There are several distinct types of structures in one cylinder barrel of a multiple cylinder block because of the difference in cooling rate of the different sections. The top is not like the middle, and neither is it necessarily like the bottom. The valve side, if there is one, is not like the water-jacket side, and both can be different from a siamese section. Each type probably has its own particular wear characteristics. One type wears excessively and is present in most cylinder barrels – a structure containing large particles of free primary ferrite. It appears in clusters and is commonly called snowball ferrite.[3] Ferrite being structurally soft pure iron, essentially ductile, is embedded loosely in a background of even weaker graphite from which it may be dislodged readily. Since it is probably these ferritic areas that furnish the large quantities of abrasive, this type of structure was selected as a standard and reproduced in test

cylinders. Normal wear attracts very little attention. Exceptionally little wear, because it is unusual, sometimes attracts attention and excessive wear, usually resulting in performance difficulties, always attracts attention. A careful examination of the structures of many rings classified as to wear indicated that one particular type of structure usually wore excessively. A structure of this type was selected as a standard for the wear-machine samples.

Abrasive action causes surface disturbance which can be measured in terms of surface temperature. It is reasonable to assume that surface temperatures of small areas on piston-ring faces under some operating conditions can reach temperatures in excess of 1200 deg. fahr. because microscopic examinations of scuffed faces of rings show areas of iron-carbide that were not there previous to running. A temperature of at least 1200 deg. fahr. is required to form or flow iron-carbide. Pressure undoubtedly contributes to the temperature rise because cross-sectional photomicrographs show a structural migration or flow indicating a slipping of the surface. Since the ring surface temperature probably is higher and subject to quicker change than the cylinder surface, a thermocouple connected to a temperature recorder was located in the center of the ring sample, the ends of the wires coming within a few thousandths of an inch of the rubbing surface.

Reciprocating the ring sample at high speed with a thermocouple in it appeared too difficult so the cylinder sample was reciprocated and the ring sample held stationary. The ring sample was mounted in a steel block with sufficient edge clearance so that it could assume a position and movement similar to a ring in a ring groove. Pressure was applied by a weight and beam mechanism calibrated in pounds per square inch for the contact area of the ring sample. After several experimental runs, a consistent condition of boundary lubrication similar to what could be expected in an engine was effected by the use of an atomizer. Clean oil was used and not recirculated. There is always a quantity of loose material on the surfaces of newly machined rings and cylinders. Particles are broken loose at the first movement of the rings over the cylinder-wall surface. These particles are abrasive and start the scratching process which is the beginning of wear.

Surface Temperature

This same phenomena happened on the wear machine and were indicated on the temperature recorder by waves in the temperature curve as the intensity of the surface disturbance increased and decreased. If the machine was operated at low speed (600 r.p.m.) and light load (600 lb. per sq. in.), the raw surfaces would masticate the loose particles and the temperature curve would settle down to a straight line. If, however, the load or speed or both were increased sufficiently before this critical stage was passed, more violent scuffing would start and result in a score. A standard run-in period was adopted during which the samples were "run-in" until the temperature line ran straight for 15 min. The surface temperature of the ring sample as shown by the temperature recorder was a good indicator of surface disturbance. A sudden increase in indicated temperature always denoted an increase in surface disturbance, but a decrease in temperature could indicate either less or more disturbance depending upon its location and extent. The analysis of temperature decreases could be made by watching the cylinder sample with a stroboscope. A scratch starting directly under the thermocouple would naturally raise the temperature, but one over at the side away from the thermocouple would lower the temperature because the abrasive particles torn from the faces of the materials would separate the surfaces at this point and the oil film having an opportunity to thicken and circulate under the thermocouple would have a cooling effect and drop the indicated temperature.

Scuffing

The operation of the machine under run-in conditions, light load, and slow speed indicated a polishing of the two surfaces. If the speed or load or both were increased, a point in relation to speed and load was reached when a surface disturbance would start and be indicated by the temperature recorder. After it once started, it usually would continue and increase in intensity until the entire surfaces were scored, provided the load and speed were maintained. If, at the first indication of a scratch, the load or speed was reduced sufficiently, the scratch would polish out and the temperature would return to normal. At this point in our observations interest was lost in studying the rate of wear under scuffing conditions because it was inconsistent and regenerative.

When an area in a cylinder or on a ring scuffs in an engine under operating conditions, it has many opportunities to polish out. A change in load or speed, an influx of lubricant produced by a shift of the piston or by increased vacuum, a slight rotation of the rings, or many other changes in operating conditions can prevent the scuffed area from spreading at a rapid rate, or operating conditions can aggravate the situation which could result in more scuffing. This analysis indicated that the rate of wear after scuffing started was affected primarily by operating conditions. The wear machine indicated that, as long as the speed and load were held below the critical point where scuffing started, the wear was negligible and the surfaces attained a high polish. This action can be considered *a decreasing rate of wear*. A decreasing rate of wear can exist until some operating condition induces scuffing, and then an increasing rate of wear can prevail. The wear in an engine that we see and try to analyze is, therefore, the result of a very complex combination of wearing at increasing and decreasing rates. Any practical analysis of this condition has been and probably always will be difficult. The important thing to know seemed to be the critical point at which the wear first changed from a decreasing to an increasing rate – from polishing to scuffing.

After samples had passed the "run-in" period, either the speed was held constant and the load increased or the load held constant and the speed increased to determine the critical scuffing point. The increases were made in equal time intervals, and the test was continued until the temperature indicated that the samples had started to scuff. Following this procedure the critical scuffing points of the standard cylinder material running against the standard ring material were determined in combinations of speed and load. These critical scuffing points, plotted in terms of speed and load, produced the curve *S* of Fig. 1. The performance of the standard materials on the wear machine could not be correlated exactly with engine experience because of the many engine operation variables not duplicated in the wear machine, but a relation, usable as a basis, was very apparent. The standard cylinder and ring materials were run many times in a single-cylinder engine having the same stroke and cylinder diameter as the wear machine, consistently reached the critical scuffing point between 3750 and 4000 r.p.m. full load. With this basis from which to work, materials showing improvement on the wear machine were checked in the single-cylinder engine.

The testing of many different structures on the wear machine indicated that the actual ability of a structure to break down into an abrasive was the characteristic that made it possible somewhat successfully to operate piston-rings in cylinders in an engine under conditions of boundary lubrica-

tion. Actual resistance to breakdown as was encountered with steel structures brought the critical scuffing point into the low-speed, low-load range. Ring structures that resisted abrasion usually showed indications of a smearing or displacement of the surface which was indicated readily by an abrupt change in surface temperature. Continued running of a steel ring sample would wear the standard iron cylinder rapidly, the material having the least resistance to abrasion. This experience checked unusually well in the single-cylinder engine because most of these tenacious ring structures would scuff and wear the cylinder at speeds as low as 2000 r.p.m. road load.

The structure that seemed most desirable was one that would not completely resist abrasion but, in abrading, would break down into extremely small particles, thereby causing so little surface disturbance that a polishing action – wear at a decreasing rate – would take place. A ring structure of this type running in combination with the standard cylinder material produced curve *M* of Fig. 1. Rings of this material were run in the single-cylinder engine using standard cylinder material and did not scuff at maximum speed – 4400 r.p.m. full load. A test cylinder sample having the same type of structure as the ring material used in developing curve *M* was then run on the wear machine against the *M* ring material. The series of tests run with these materials produced curve *MM* of Fig. 1. Rings and cylinder sleeves having these same structures were run in a Diesel engine that had previously scuffed with standard ring and cylinder material. The new materials did not show indications of scuffing but tended to polish as wear took place. The change in the piston-ring and cylinder-sleeve material changed the wear from an increasing rate to a decreasing rate.

Ferrox

A material having the *best wearing* characteristics, may not have the physical characteristics necessary to make a good piston-ring. Future research may reveal such a material but, in order to approach the ideal, a piston-ring material having the required physical characteristics was given a treatment which changed the structure of the surface for a depth of approximately 0.0005 in. The structure of the surface layer so produced was extremely fine and granular but firmly rooted to the original remaining structure. Tests on the wear machine indicated that this surface structure was ideal from a wear standpoint because it was not too tenacious to resist abrasion and, in abrading, broke down into a very fine polishing medium which polished the cylinder sample. The ring sample also acquired a high polish. The material broken loose from the face of the ring immediately started to polish the cylinder sample. As the test continued and "wear-in" took place, the degree of polish increased. As a material polishes, the rate of wear decreases and, of course, it can decrease to a practical stop. This surface has been given the trade name of Ferrox.

The scuffing curve *F* (Fig. 1) could not be developed from the wear-machine results using the standard procedure because scuffing did not occur within the speed and load limits of the machine. Points were reached in the operation of the machine using different combinations of speed and load when the oil would vaporize because of the high surface temperature generated. Curve *F* was therefore developed from the vaporization points of the oil. Even after the oil vaporized and the samples ran dry, a breakdown of the surface did not occur for several minutes. Rings with Ferrox surface have been run in many different types of engines. Ferrox rings that were run in a two-stroke Diesel engine under conditions of high pressure and temperature, polished and showed no signs of scuffing. Standard and *M* ring material had previously scuffed in this engine to the place where wear was a serious problem.

Cylinder samples for the wear machine were cut from a

(*Continued on page* 156)

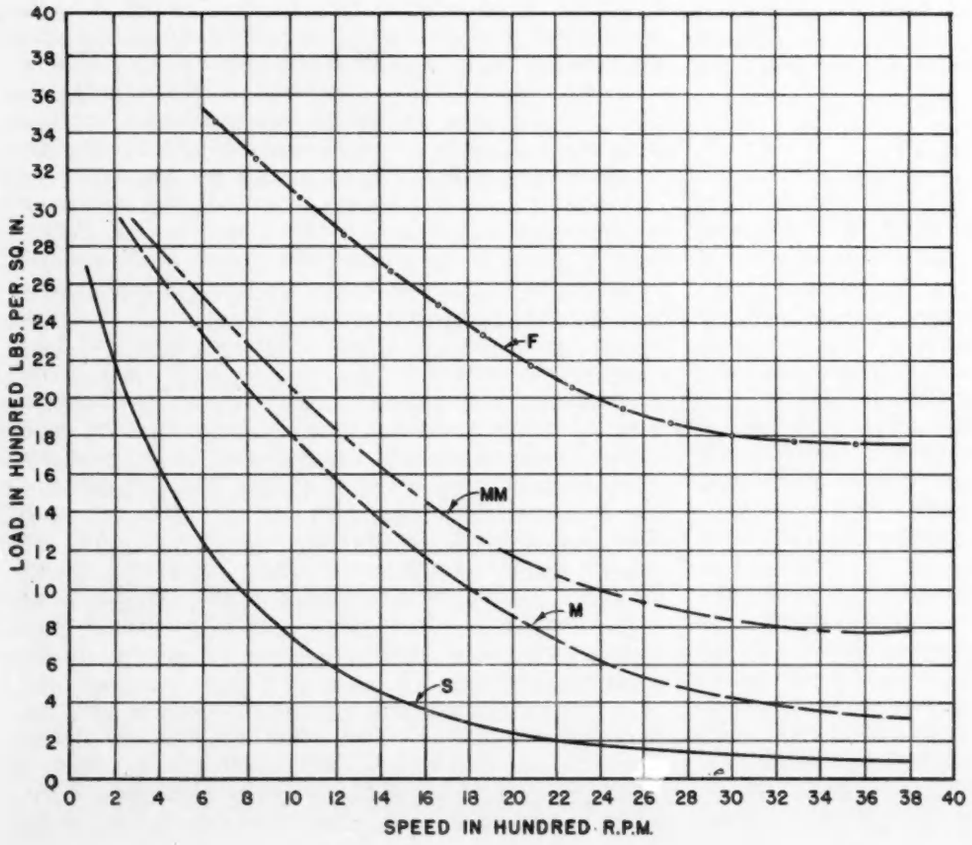


Fig. 1 – Critical Scuffing Curves for Various Ring and Cylinder Materials

Carburetion, Manifolding and Fuel Antiknock Value

By Earl Bartholomew, Harold Chalk, and Benjamin Brewster
Ethyl Gasoline Corp.

INCREASED performance at high speed, which is an important characteristic of cars built during the last few years, has required the use of intake manifolds of considerably greater cross-sectional area which, together with certain other changes in engine design, cause unequal distribution of the less volatile portions of the fuel not vaporized in the manifold. Consequently, the fuel which enters the cylinders varies in composition. If other conditions are equal, the relative knocking tendency of the cylinders is determined by the distribution of antiknock value through the fractions of the fuel and by the air-fuel ratios of the mixtures in the cylinders. Ordinarily the leanest cylinder knocks the most.

Fuels rated in the laboratory by conventional methods as having equal antiknock value may differ considerably in road performance because of differences in volatility and distribution of antiknock value through their fractions. The problem is complicated further by differences in the volatility of the commonly used secondary reference fuels.

Recently developed instrumentation permits the determination of the knocking tendency, air-fuel ratio, and spark advance for the individual cylinders of an automobile engine on the road. A modified fuel system and intake manifold for the C.F.R. engine make possible the duplication and measurement in the laboratory of the effects caused by imperfect fuel distribution within the car engine.

DURING the past six years the development of laboratory testing methods to give knock ratings that correlate with the ratings obtained in cars on the road has been a major problem of the oil refiner and car manufacturer. After the cooperative road-testing program at Uniontown, Pa., in 1932 a laboratory procedure was developed which gave ratings agreeing remarkably well with the road ratings. The ratings obtained on the road in the second Uniontown program in 1934, did not agree quite so well with ratings obtained by the Motor Method.

Since that time road ratings have tended to deviate more and more from the consistent pattern which characterized the earlier ratings. This deviation applies to average road ratings and particularly to ratings in individual cars. Fuels which have approximately the same Motor and Research Method ratings may have road ratings as much as 8 octane numbers higher or lower than their laboratory ratings. Fuels which have a fairly large difference between their Research and Motor Method ratings may, on the road, show no appreciation over their Motor Method ratings, and others having the same difference between their Research and Motor Method ratings in the laboratory may, under the same conditions of car operation, have road ratings considerably higher than their Research-Method ratings.

As an example of such performance we may consider the ratings of two specially blended fuels whose road and laboratory ratings are tabulated below:

Table 1 – Road and Laboratory Knock Ratings of Two Specially Blended Fuels

Fuel	Research-Method Rating	Motor-Method Rating	Average of Winter Road Ratings in Eleven Cars	Winter Road Rating in Car No. 155
S- 6	72.9	75.6	71.4	67.6
S-11	71.7	73.8	74.8	80.0

The average road rating of fuel S-6 is substantially *lower* than either its Motor- or Research-Method rating, and the average road rating of fuel S-11 is considerably *higher* than either its Motor- or Research-Method rating. Although the Research- and Motor-Method ratings of the two fuels differ by 1.2 and 1.8 octane numbers respectively in one direction, their ratings in Car No. 155 differ by 12.4 octane numbers in the opposite direction.

Car No. 155 is highly satisfactory for knock-testing purposes, being relatively free of noises other than that due to knock and having more than average sensitivity to changes in fuel octane number. The influence of certain engine variables such as speed, spark advance, and temperature on the relative antiknock value of different types of fuels is recognized generally, and may be demonstrated readily in the laboratory on single-cylinder engines such as the A.S.T.M.-C.F.R. engine, but no variations of the Motor and Research Methods which have been tried, involving only changes in speed, temperature, and spark advance, have served to bring the laboratory ratings of these fuels into agreement with the road ratings.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1938.]

Changes in Engine Design and Fuels

A study of such data obtained in numerous laboratories during the past three years led to the suspicion that some factor or combination of factors as yet neglected in the standard laboratory knock-testing procedure, in some cases, may have roughly as great an influence on road ratings as have engine speed and temperature. Since the serious disagreement of road ratings and laboratory ratings made by the generally established procedures, or simple variations thereof, is of relatively recent origin, the changes in engine design which may account for such deviations are of immediate interest. The following changes seem to warrant attention:

(1) Increased cross-sectional area of intake manifolds in conjunction with the general adoption of down-draft carburetors having larger tube diameters, resulting in considerably lower intake-manifold velocities in the low-speed range of the engine, to which knocking now is confined generally.

(2) The automatic choke.

(3) Decreased mixture heating by the exhaust gases and the use of automatic heat-control valves whose operation is controlled largely by the temperature of the air surrounding the bi-metallic element.

(4) The elimination of thermostatically controlled radiator shutters.

[1] See S.A.E. TRANSACTIONS, April, 1937, pp. 144-148: "Relative Knocking Characteristics of Motor Fuels in Service," by John M. Campbell, Wheeler G. Lovell, and T. A. Boyd.

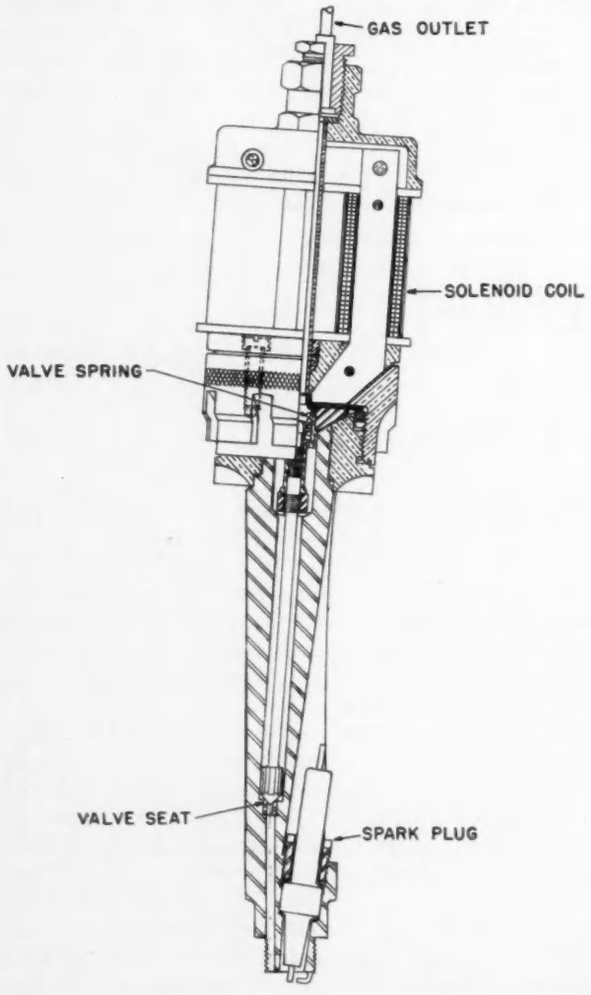


FIG. 1 VALVE FOR SAMPLING MIXTURE IN THE CYLINDERS OF A MULTICYLINDER ENGINE.

In addition to these engine-design factors there is the possible effect of recent developments in gasoline production such as polymerization, reforming, and new cracking methods. It is the purpose of this paper to review some of the recent work in the laboratories of the Ethyl Gasoline Corp. which has had for its purpose:

(1) The explanation for at least some of the previously described unusual performance of fuels in cars on the road.

(2) The development of improved instrumentation for the analysis of events in the intake manifold and cylinder.

(3) The formulation of principles to govern the selection of fuels to meet best any given set of manifold and carburetion conditions.

The paper is designed to present fundamental principles that apply to a complex problem, hence, in the interest of simplification, the discussion is confined in so far as possible to typical conditions to be met.

Campbell, Lovell and Boyd[1] pointed to the probability that mixture ratio and fuel volatility affect the relative antiknock value of fuels and cited tests in cars on the road which appeared to confirm the hypothesis. The work in the laboratories of the Ethyl Gasoline Corp. indicates that fuel volatility, variation of fuel antiknock value with distillation temperature, carburetor air-fuel ratio, design of the intake manifold and associated parts, variation of other conditions which affect fuel distribution within the engine, and the reference fuels used, all have an important bearing on the relative knock ratings of fuels.

Fuel Distribution Within the Engine

Fuel of substantially uniform composition is distributed to all the cylinders of a multicylinder engine in approximately equal proportions with the admixed air if the available heat, time of contact of air with fuel, and the agitation of the mixture are sufficient to vaporize the fuel completely and mix it homogeneously with the air, or if the manifold velocity is high enough to carry the liquid fuel in the air stream by entrainment. High volumetric efficiency of automobile engines has been obtained only with reasonably low intake-manifold velocity. The high brake mean effective pressures at high engine speed in recently built cars require the use of intake manifolds of large cross-sectional area and a minimum of mixture heating. Both of these design features are diametrically opposed to good fuel distribution at low car speeds where knocking has become most prevalent. It is impossible to combine in a single conventional unsupercharged manifold the most satisfactory characteristics for both high- and low-speed operation.

The determination of the air-fuel ratios of the mixtures entering the individual cylinders of a multicylinder engine on the road has always presented a problem. Samples of exhaust gases withdrawn from the exhaust manifold as close as possible to the exhaust valves are invariably contaminated by products of combustion from other cylinders. The substitution for the conventional manifold of a separate exhaust stack for each cylinder destroys the standard hot-spot arrangement and considerably increases the temperature under the hood. The most desirable approach is the sampling of the mixture within each cylinder. This procedure has been made possible by the combination spark-plug and gas sampling valve shown in Fig. 1. The small solenoid valve between the cylinder and the gas sampling tube has only a small movement and remains open for a very short period. Its operation is controlled by an auxiliary breaker mechanism mounted above the conventional distributor and driven by the distributor shaft. At the instant the points of the auxiliary breaker close, a condenser in the circuit discharges a relatively heavy current through the solenoid coil and causes the valve to open momentarily.

At all other times the valve is held in the closed position by the pressure of a coil spring.

The complete apparatus is shown diagrammatically in Fig. 2. When a sample is to be obtained, the vacuum pump, which is driven by a motor which draws its current from a 6-220 volt motor-generator set, evacuates the gas sampling tube to an absolute pressure of about 5 mm. mercury. The car is accelerated under the conditions prevailing in the road knock test and the valve is allowed to operate only within the 5 m.p.h. speed range of maximum knock intensity. From 6 to 8 repetitions of the acceleration are required to fill the gas sampling tube to atmospheric pressure. The procedure is then repeated for the other cylinders.

Table 2 – Air-Fuel Ratios of the Mixtures in the Cylinders of an Automobile Engine at 10 to 15 M.P.H. Atmospheric Temperature, 56 Deg. Fahr., Fuel of Winter Volatility

Cylinder No.							
1	2	3	4	5	6	7	8
14.59	16.01	13.29	14.13	12.63	11.56	15.27	12.37

Average air-fuel ratio, all cylinders	13.6
Average for cylinders 1, 2, 7 and 8	14.4
Average for cylinders 3, 4, 5 and 6	12.8

Table 2 shows the air-fuel ratios for the individual cylinders of a certain car during acceleration at 10 to 15 m.p.h. at an atmospheric temperature of 56 deg. fahr., as determined by the apparatus shown in Fig. 2. The mean air-fuel ratio computed from these data is shown, also the computed means for cylinder groups 1, 2, 7 and 8 and 3, 4, 5 and 6, each of which is fed by a single barrel of a dual down-draft carburetor.

For the same conditions of operation, the mean air-fuel ratio is normally richer than that shown in Table 2. There is, however, always a tendency for a portion of the heavy ends of the acceleration charge to lag behind the air-vapor mixture for a short period after the sudden opening of the throttle, with the result that the mean cylinder air-fuel ratio is momentarily leaner than after the lagging liquid fuel starts to flow into the cylinders. This factor may, in part, account also for the difference between the mean air-fuel ratios for the two groups of cylinders, inasmuch as the manifold for cylinders 1, 2, 7 and 8 is longer than that for the remaining cylinders and has more surface to be wet by the acceleration charge. It should be noted that the rear cylinder of each group receives the richest mixture, presumably because of the forward motion of the car which tends to cause liquid fuel to accumulate in the rear of the manifolds.

The air-fuel ratio of the mixtures in cylinders 1, 2, 7 and 8 varies from 12.37 to 16.01. Often the variation is greater. The mechanism by which fuel and air are delivered to the cylinders in such a wide range of proportions appears to be as follows:

Air and fuel are metered by the carburetor and enter the manifold. The magnitude of the fraction of the fuel in the vapor state as the mixture leaves the manifold is a function of the heat in the fuel and air supplied to the carburetor, the heat added by the manifold, the volatility and latent heat of the fuel, and the efficiency of mixing in the manifold. Although the available heat may be adequate to ensure complete vaporization under equilibrium conditions, the time of contact and degree of mixing in the manifold may not be sufficient to bring about this condition.

The air and the lighter fractions of the fuel, vaporized in the carburetor and manifold, are distributed substantially equally to the cylinders. A part of the unvaporized, less volatile portion is distributed fairly uniformly by entrainment with the air, but the quantity of heavy ends so distributed

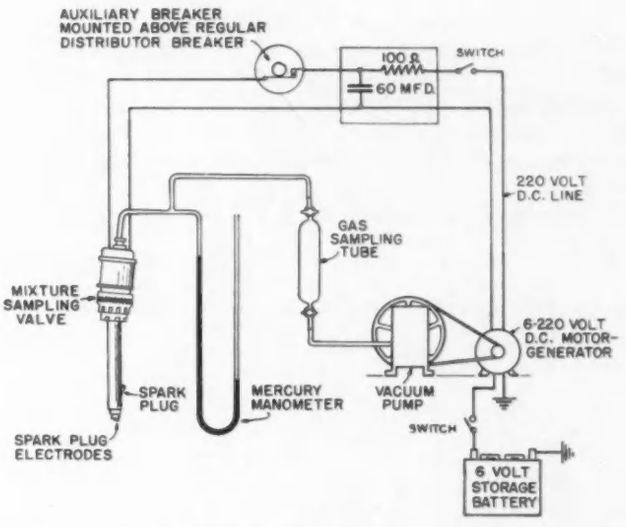


FIG. 2 DIAGRAMMATIC SKETCH OF MIXTURE SAMPLING VALVE AND ASSOCIATED EQUIPMENT.

becomes less as engine speed decreases and appears to be small at low engine speed.

The remainder of the heavy liquid portion is distributed unequally to the cylinders, the quantity entering any given cylinder depending on the shape, cross-sectional area and inclination of the manifold, engine speed, surging of the mixture within the manifold, and other factors. Thus, in the three typical cases, one or more cylinders may receive substantially only air and the lighter ends which leave the manifold in the vapor state; others may receive air, vaporized light ends and liquid heavy ends in approximately the same air-fuel ratio as that of the mixture leaving the carburetor, and still others may receive air, vaporized light ends and liquid heavy ends in the same proportions as in the second case plus the liquid heavy ends which, under conditions of perfect distribution, would have entered the first-mentioned cylinder or group of cylinders. Other cylinders may receive liquid heavy ends in any amounts between the two limits just described, and it would be possible for one or more cylinders to receive even more liquid heavy ends than the richest cylinder cited. The three typical distribution conditions are summarized in Table 3 in which the fuel entering the carburetor is assumed to contain L lb. of light ends, which leave the manifold in the vapor state, for each H lb. of heavy ends that leave the manifold in the liquid state.

Table 3 – Three Typical Fuel Distribution Conditions in the Multicylinder Engine

Cylinder Type	Air Received by Cylinder, lb.	Vaporized Light Ends Received by Cylinder, lb.	Liquid Heavy Ends Received by Cylinder, lb.	Cylinder Air-fuel Ratio	Carburetor Air-fuel Ratio	Cylinder Mixture Strength Relative to Carburetor
A	M	L	O	$\frac{M}{L}$	$\frac{M}{L+H}$	Leaner
B	M	L	H	$\frac{M}{L+H}$	$\frac{M}{L+H}$	Equal
C	M	L	2 H	$\frac{M}{L+2H}$	$\frac{M}{L+H}$	Richer

$$\text{Mean air-fuel ratio} = \text{carburetor air-fuel ratio} = \frac{3}{\frac{L}{M} + \frac{L+H}{M} + \frac{L+2H}{M}} = \frac{M}{L+H}$$

Assume, for example, in the case of cylinder group 1, 2, 7, and 8 of the engine in Table 2, that the average cylinder air-fuel ratio was equal to the carburetor air-fuel ratio, that no liquid fuel was distributed by entrainment, and that the leanest cylinder received only vaporized fuel, in an amount equal to that distributed to each of the remaining cylinders, along with varying amounts of liquid fuel. The fraction of the fuel entering cylinders 1, 2, 7, and 8 which left the manifold in the vapor state was $100 \times \frac{14.4}{16.01} = 90.0$ per cent. If M in Table 3 is taken as 1 lb. $(L + H)$ for cylinders 1, 2, 7, and 8 was $\frac{1}{14.4} = 0.0695$ lb. L was $\frac{1}{16.01} = 0.0625$ lb. H was $0.0695 - 0.0625 = 0.0070$ lb. $\frac{M}{L+2H} = 13.08$, which is the air-fuel ratio that would have existed in one or more of the cylinders if they had received air and fuel in the same proportions as cylinder C of Table 3. Actually no cylinder in this group received such a mixture, but cylinder 8 received a mixture having an air-fuel ratio of 12.37, or 0.0809 lb. fuel per lb. of air, consisting of L lb. of vaporized light ends and 2.63 H lb. of liquid heavy ends.

Effect of Variable Fuel Distribution on Knocking

The non-uniform distribution of fuel to the cylinders as just described gives rise to two conditions which are of vital importance in the fuel-knocking problem, namely:

(1) The air-fuel ratios of the mixtures in the cylinders may vary widely and cause a substantial variation in knocking tendency among the cylinders.

(2) The composition of the fuel in the cylinders may be quite different and cause a variation in knocking tendency in addition to that introduced by the differences in air-fuel ratios.

Because of these conditions, independent of all others, there is ample cause for a marked variation in the knocking tendency of the fuel in the individual cylinders of an engine operated under conditions of imperfect distribution. Whether the cylinders of Type *A*, *B* or *C* (Table 3) have the greatest knocking tendency depends on the following three factors:

(1) The carburetor air-fuel ratio.

(2) The degree of vaporization and hence the magnitude of the variation in cylinder air-fuel ratio.

(3) The relative concentration of antiknock value in the vaporized light ends and the unvaporized heavy ends of the fuel.

Adaptation of the Single-Cylinder Engine

The laboratory knock-testing engine, operated in the conventional manner, permits no prediction of the behavior of a fuel under variations of the three conditions just enumerated. Under such a procedure it is useful only for the evaluation of the knocking tendency of a fuel under a range of cylinder conditions when the composition and air-fuel ratio of the mixture entering the cylinder are the same as for the mixture leaving the carburetor.

A fundamental study on a multicylinder engine of fuel distribution and its effect on knocking is surrounded by many difficulties. The richness of the mixture which enters the several cylinders can be determined most conveniently by exhaust-gas analysis as described previously. The knocking tendency of the fuel in the individual cylinders is not measurable by any instrumentation developed to date but can be evaluated by audibility methods to be described later. The maintenance of constant conditions of operation is difficult because of uncontrollable variations in ignition timing and in temperatures. It is, therefore, highly desirable to adapt the C.F.R. engine to the study of this problem, inasmuch as the above-mentioned variations in operating conditions have been minimized.

Fig. 3 shows four intake manifolds for the C.F.R. engine which have been designed to permit a study, under controlled conditions, of the effects on knocking caused by variations of fuel distribution in the multicylinder engine. In the arrangement shown in Fig. 3*a* (upper left), a short spacer bolted directly to the engine cylinder is fitted with a thermometer for the indication of mixture temperature. A standard C.F.R. manifold, with its mixture heater, is bolted to the spacer, the flange normally bolted to the carburetor being attached to the spacer. A specially designed water-jacketed manifold section is provided with two flanges, one of which is connected to the standard C.F.R. manifold and the other to the standard C.F.R. carburetor body.

After the fuel-air mixture leaves the carburetor, it is forced to flow down through the jacketed section of manifold and up through the standard C.F.R. manifold before entering the cylinder. Unvaporized fuel which is not carried into the cylinder by entrainment in the air stream drains from the jacketed section of the manifold into a graduated burette. The degree of fuel vaporization is controlled by the temperature of the water that enters the manifold jacket. Water is heated or cooled externally and mixed with tap water to produce the temperature required to effect the desired degree of vaporization. Any final mixture temperature higher than that existing in the jacketed manifold may be obtained by the use of the standard C.F.R. mixture heater.

In order that the rate of carburetor fuel consumption may be measured, the standard C.F.R. integral float-bowl and fuel-container assemblies are replaced by separate float-bowl and fuel-container assemblies from the Series 30 engine. Three-way cocks and graduated glass bulbs permit the volumetric measurement of fuel entering the carburetor. An air-conditioning unit, as described in the June, 1937, issue of the S.A.E. JOURNAL, page 250,[2] maintains carburetor air at constant temperature and humidity and provides for air measurement by means of a round-edge orifice similar to that described in the 1936 Transactions of the S.A.E., p. 97.[3]

The arrangement shown in Fig. 3*b* (upper right) has been found to effect a more complete separation of liquid fuel from the mixture of air and vaporized fuel and, thus, to simulate more closely the functioning of the multicylinder engine manifold. Essentially it is the set-up shown in Fig. 3*a* with the regular C.F.R. manifold and special jacketed section reversed in position. Variation in the position of the piston in the surge cylinder attached to the liquid fuel drain at the bottom of the manifold permits, with little or no application of heat to the manifold, a wide range of adjustment of the fraction of the fuel drained from the manifold in the liquid state. It is impossible, however, to obtain with this manifold a final mixture temperature higher than that of the liquid in the jacket. This restriction prevents operation with cylinder conditions having a severity much in excess of those for the Research Method.

The manifold shown in Fig. 3*c* (lower left) permits the use of any desired final mixture temperature higher than approximately 200 deg. fahr., effects as good separation of vaporized and liquid fuel as the manifold shown in Fig. 3*b*, and requires no surge chamber. The separating chamber has a cross-sectional area over four times as great as that of the remainder of the manifold and consequently reduces the velocity of flow to a point that ensures good separation of liquid fuel from the stream of air and vaporized fuel. Heat may be applied to the mixture by conventional immersion

[2] See S.A.E. TRANSACTIONS, June, 1937, p. 250: Earl Bartholomew's discussion of "Influence of Humidity on Knock Ratings," by J. R. MacGregor.

[3] See S.A.E. TRANSACTIONS, March, 1936, p. 97: Earl Bartholomew's discussion of "Relation of Exhaust-Gas Composition to Air-Fuel Ratio," by B. A. D'Alleva and W. G. Lovell.

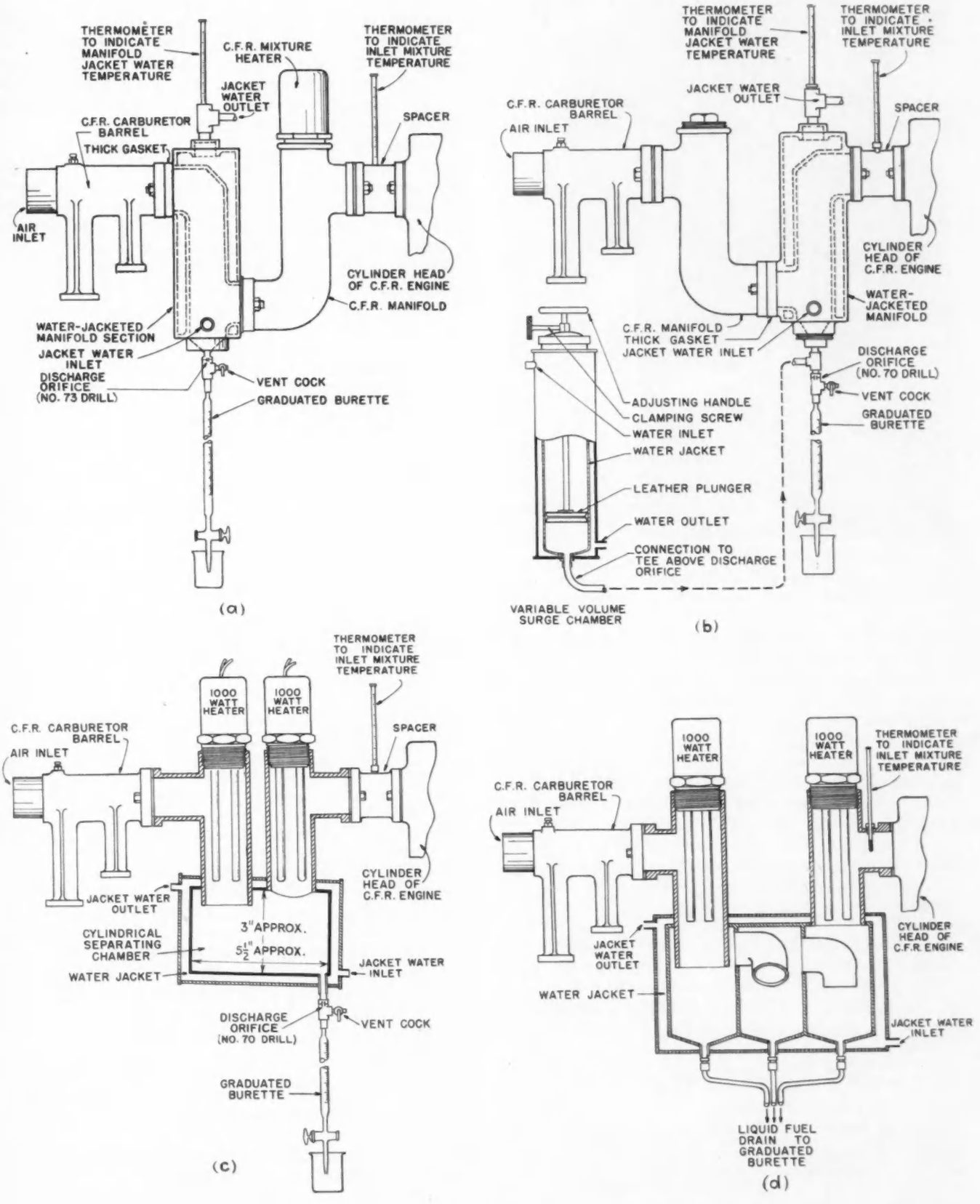


FIG. 3 DIAGRAMMATIC SKETCHES OF INTAKE MANIFOLDS FOR THE C.F.R. ENGINE WHICH SIMULATE THE EFFECTS OF IMPERFECT MIXTURE DISTRIBUTION IN THE MULTICYLINDER ENGINE.

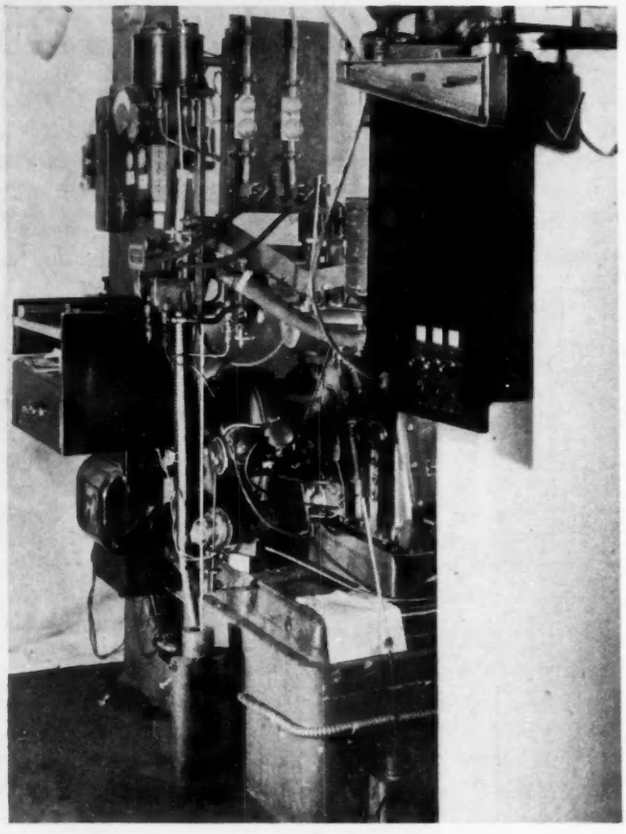


Fig. 4 – Modified Intake Manifold and Fuel System Installed on the C.F.R. Engine for Studies of the Effects of Imperfect Fuel Distribution in the Automobile Engine

heaters before or after it enters the separating chamber and the chamber itself may be heated or cooled by the circulation of water through the surrounding jacket. Fig. 4 shows the complete modified fuel system and the manifold of Fig. 3*c* installed on a C.F.R. engine.

The manifold of Fig. 3*d* (lower right) is of the same essential design as that of Fig. 3*c* but is provided with three separating chambers instead of one. Experience to date with this design is insufficient to indicate whether it possesses any significant advantage over that shown in Fig. 3*c*.

Since provision is made for the measurement of air and fuel delivered to the carburetor and the unvaporized fuel bled from the manifold, it is possible to compute the air-fuel ratios of the mixtures leaving the carburetor and of those entering the cylinder. When unvaporized fuel is withdrawn from the manifold, the mixture entering the cylinder is leaner than that leaving the carburetor and the distribution condition is that of Cylinder *A* in Table 3. When no unvaporized fuel is bled from the manifold, the cylinder and carburetor air-fuel ratios are equal and the distribution condition is that of Cylinder *B* in Table 3.

The distribution condition of Cylinder *C* in Table 3 is duplicated on the single-cylinder engine by running first under the conditions of Cylinder *A* and collecting *H* lb. of unvaporized heavy ends. It would be difficult to design an arrangement for the modified C.F.R. manifold which would feed the additional liquid heavy ends to the normal charge in the desired ratio. It is easy to accomplish the same result by adding the excess heavy ends for Cylinder *C* directly to the original fuel in the ratio of *H* lb. of heavy ends for each $(L + H)$ lb. of original fuel and feeding this mixture to the carburetor. The carburetor air-fuel ratio is adjusted to $\frac{L+H}{L+2H} \times$ (air-fuel ratio for Cylinder *B*) $= \frac{M}{L+2H}$, and the manifold heating is adjusted for complete vaporization. It should be noted that, during the determination of the data for the curve of Cylinder *C*, the carburetor air-fuel ratio for the single-cylinder engine is not the same as that for the multicylinder engine whose performance is simulated.

The ratio of *L* to *H* may be varied over a wide range by adjustment of manifold heating. The liquid heavy ends also may be added to the original fuel in other proportions to duplicate other mixture ratios found to exist in the cylinders of an automobile engine.

Fig. 5 shows, for 1937 C.F.R. winter fuel WB-2 and secondary reference fuels A-4 and C-10 the relation of the degree of vaporization in the manifold to the temperature of the water leaving the jacket of the manifold of Fig. 3*a*. The distillation curves for the three fuels are shown in Fig. 6.

The degree of vaporization in the modified single cylinder engine manifold varies with fuel volatility as in the manifold of the multicylinder engine. Fig. 7 shows for one manifold condition for the 1937 C.F.R. preferred winter fuels, and for 3 blends of the approved secondary reference fuels, the relation between the per cent of fuel vaporized in the manifold and the per cent over at 310 deg. fahr. in the A.S.T.M. distillation, whereas Fig. 8 shows, for the same fuels and manifold condition, the relation between the per cent of the fuel vaporized in the manifold and the area under the distillation curve above 80 deg. fahr. Figs. 7 and 8 show a fairly consistent relation between the variables plotted but, of course, the degree of vaporization depends also on the latent heat of vaporization of the fuel components. The degree of vaporization in the manifold might correlate even more satisfactorily with data from an air distillation.

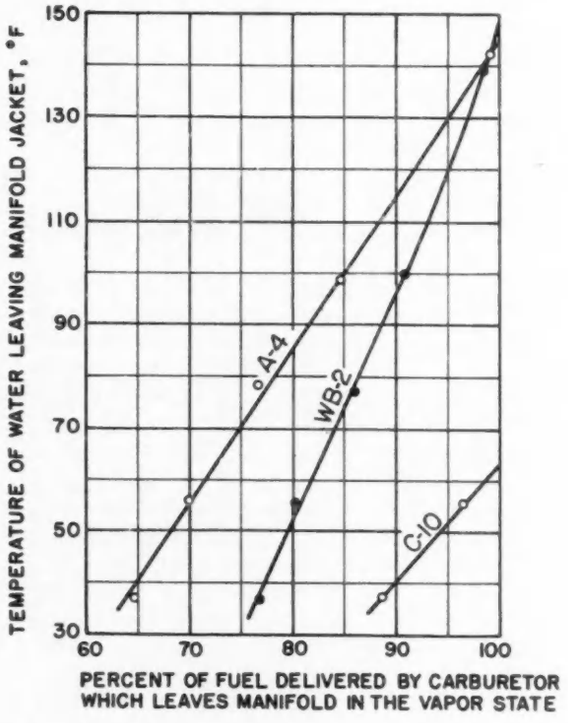


FIG. 5. EFFECT OF INTAKE MANIFOLD JACKET TEMPERATURE ON FUEL VAPORIZATION

C.F.R. ENGINE, SPECIAL MANIFOLD AND CARBURETOR (FIG. 3a)
FIXED CARBURETOR SETTING
MIXTURE TEMP. ENTERING CYLINDER, 300° F.
SPARK ADVANCE, 21°
CYLINDER JACKET TEMP., 212° F.
ENGINE SPEED, 900 R.P.M.

Knock Ratings with the Bled Manifold

After the carburetor float level and manifold jacket temperature have been adjusted to give the desired carburetor and cylinder air-fuel ratios respectively, and the final mixture temperature has been brought to the desired figure, the general procedure for making knock ratings with the modified manifold is the same as for the Motor Method. The compression ratio is adjusted for standard knock intensity and the bouncing-pin indicator is used to measure knocking tendency.

If the primary reference fuels are used, the fuel octane numbers obtained are likely to differ considerably, under conditions of partial fuel vaporization, from those obtained by the use of the approved secondary reference fuels. Within the range of manifold conditions of practical interest, the vaporization of heptane-octane mixtures is substantially complete, hence octane numbers obtained by the use of the primary reference fuels have the same meaning as those obtained by

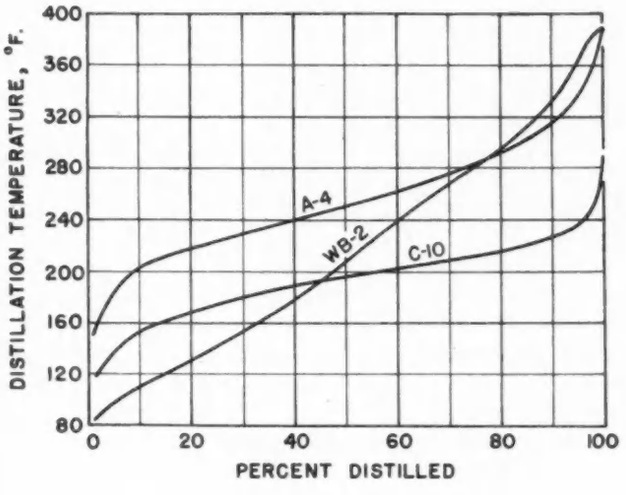


FIG. 6 DISTILLATION CURVES OF FUELS WHOSE VAPORIZATION IN THE MANIFOLD IS ILLUSTRATED IN FIG. 5.

the Motor or Research Methods or by any other method which employs the conventional manifold. On the other hand, the volatility of the C-10 and A-5 reference fuels is quite different; therefore, under any condition of partial vaporization, a larger fraction of the C-10 component is vaporized in the manifold. As the concentration of A-5 in the blend is increased, the per cent of the blend leaving the manifold in the vapor state is decreased, and the inherent antiknock value of the reference-fuel mixture which enters the cylinder decreases less rapidly than does the per cent of C-10 in the blend. The result is that, under these conditions, a given increment of antiknock value matched against blends of the secondary reference fuels is interpreted as representing a larger difference in octane numbers than when matched directly against the primary reference fuels. The same condition presumably exists during road knock tests and raises the question of the possible relation between it and the occasionally observed relative insensitivity of cars to changes in the per cent of C-10 in the blends of the secondary reference fuels.

When a measure of the difference in knocking tendency of fuels under conditions of imperfect distribution is desired in units bearing a definite relation to critical compression ratio, it is customary to make the comparisons directly against heptane-octane mixtures. When the problem is to correlate laboratory ratings with those obtained on the road with the secondary reference fuels, the latter reference fuels are used

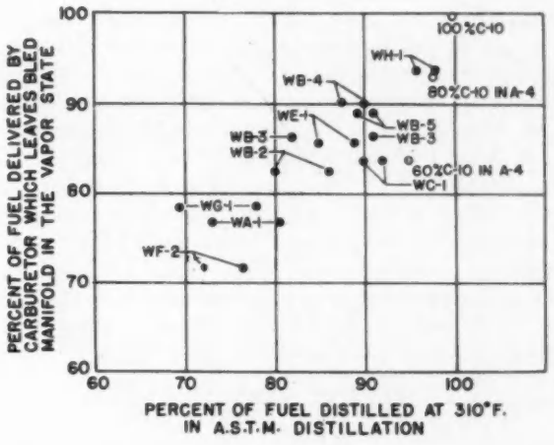


FIG. 7 RELATION BETWEEN THE PERCENT OF FUEL DELIVERED BY THE CARBURETOR WHICH LEAVES THE BLED MANIFOLD IN THE VAPOR STATE, AND THE PERCENT OVER AT 310 °F. IN THE A.S.T.M. DISTILLATION.

C.F.R. ENGINE, SPECIAL MANIFOLD AND CARBURETOR (FIG. 3a)
MAXIMUM KNOCK AIR-FUEL RATIO
MANIFOLD JACKET TEMP., 68°F.
CARBURETOR AIR TEMP., 80°F.
MIXTURE TEMP. ENTERING CYLINDER, 85°F.
SPARK ADVANCE, MOTOR METHOD
CYLINDER JACKET TEMP., 212°F.
ENGINE SPEED, 900 R.P.M.

in the laboratory. If the conditions of engine operation on the road are duplicated in the laboratory, the fractions of the reference-fuel blends delivered by the carburetor that leave the manifold in the vapor state are the same in both cases. If the octane numbers in terms of the primary reference fuels are to be determined at one time for one or more fuels at several degrees of imperfect vaporization, it is customary to determine the compression ratio for each that produces a knockmeter reading of 50 and to read octane numbers from a curve of octane number versus compression ratio for constant knock intensity, made at the conclusion of the tests.

When the degree of vaporization is low, as for example 50 to 60 per cent, the unsteady state following a change from one fuel to another is of quite long duration. In any case

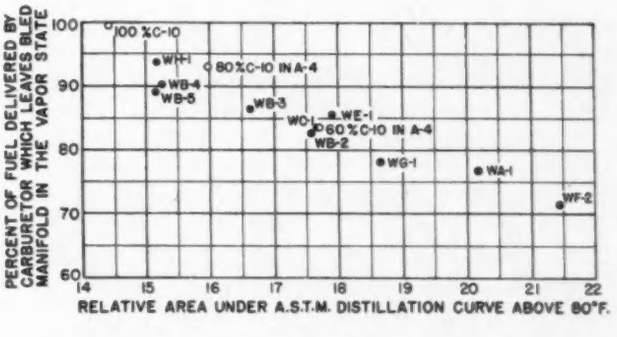


FIG. 8 RELATION BETWEEN THE PERCENT OF FUEL DELIVERED BY THE CARBURETOR WHICH LEAVES THE BLED MANIFOLD IN THE VAPOR STATE, AND THE RELATIVE AREA UNDER THE A.S.T.M. DISTILLATION CURVE

C.F.R. ENGINE, SPECIAL MANIFOLD AND CARBURETOR (FIG. 3a)
MAXIMUM KNOCK AIR-FUEL RATIO
MANIFOLD JACKET TEMP., 68°F.
CARBURETOR AIR TEMP., 80°F.
MIXTURE TEMP. ENTERING CYLINDER, 85°F.
SPARK ADVANCE, MOTOR METHOD
CYLINDER JACKET TEMP., 212°F.
ENGINE SPEED, 900 R.P.M.

where vaporization is not perfect, the unsteady state persists for a longer period than under the Motor Method conditions. There appears to be no way at constant speed of the laboratory engine to shorten the period of transient conditions. In tests on the road the car engine is usually operated at high speed for a brief period, following a change from one fuel to another, during which the manifold is washed clean of the previous fuel more quickly. A multi-speed, power-absorption generator and larger samples of fuel should permit a similar procedure in the laboratory.

Even under the most favorable conditions in the laboratory, ratings made with the bled manifold require more time and a higher degree of skill than those made by the conventional

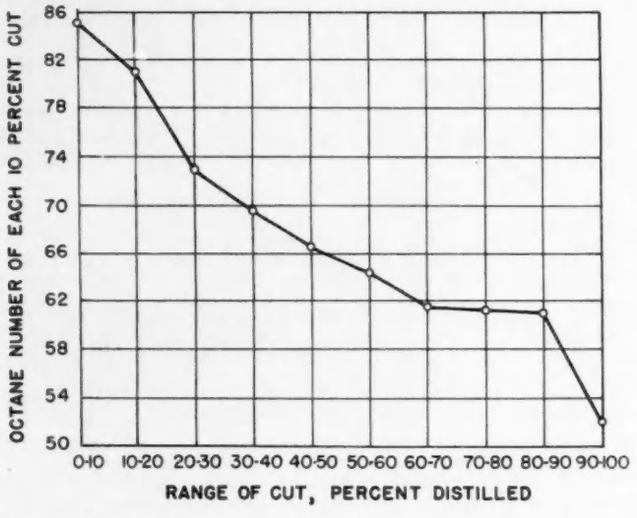


FIG. 9 ANTIKNOCK VALUE OF TEN EQUAL CUTS OF A TYPICAL COMMERCIAL BASE GASOLINE SEPARATED BY SIMPLE DISTILLATION.

C.F.R. ENGINE, STANDARD MANIFOLD
NO MIXTURE HEATER
MAXIMUM KNOCK AIR-FUEL RATIO
MIXTURE TEMP. ENTERING CYLINDER, 140°F. (OBTAINED BY HEATING INTAKE AIR)
SPARK ADVANCE, 18°
CYLINDER JACKET TEMP., 212°F.
ENGINE SPEED, 600 R.P.M.

methods. It is believed that reproducibility of ratings is about the same. Complete studies of fuels under a wide range of carburetor and manifold conditions, carried out by the procedure just described, should constitute a valuable tool for both the fuel refiner and automotive engineer.

The result of tests made on a single-cylinder engine equipped with the bled manifold might be used as a basis for the correction of ratings obtained by the conventional procedure. It would be preferable to obtain final ratings, by a single engine test, which would be corrected for the distribution effects in the car engine and in agreement with the average ratings obtained in cars on the road. Possibly a set of engine conditions, including a modification of the intake manifold, could be found to produce such ratings.

Distribution through the Fuel Distillation Range

It has been stated previously that the relative knocking tendency of the lean, normal, and rich cylinders (denoted by *A, B,* and *C* respectively in Table 3) depends on the carburetor air-fuel ratio, the degree of vaporization in the carburetor and manifold, and the relative concentration of antiknock value in the vaporized light ends and unvaporized heavy ends of the fuel. The distribution of antiknock value through the distillation range may be measured by separating the fuel into a number of cuts by distillation and determining the octane number of each by any of the conventional knock-testing methods. Fig. 9 shows such data for a typical commercial lead-free gasoline base stock so treated. The most volatile fractions have the highest antiknock value, and the heavier cuts have progressively lower octane number.

Such a procedure is time-consuming and not required for the determination of the distribution of antiknock value within the practical limits of interest. A more convenient method is to effect the distillation and measure antiknock value simultaneously in the engine itself. Fig. 10 shows the octane numbers of a number of cuts of the 1937 C.F.R. preferred winter road-test fuels and of five other fuels. The cuts were made by varying the degree of vaporization in the manifold and, in each case, they included the most volatile fractions of the fuel. All ratings were made at the air-fuel ratio for maximum knock. The curves are stopped at approximately the 50 per cent vaporization point for the least volatile fuels, inasmuch as this point appears to represent roughly the extreme condition to be met on the road.

Fuels may be separated roughly into three broad classes in accordance with the distribution of antiknock value along their distillation curves, as follows:

(1) Those whose cuts by simple distillation from the 50 to 100 per cent points have approximately the same antiknock value as the first 50 per cent cut as, for example, fuel WA-1 in Fig. 10*a*.

(2) Those whose cuts by simple distillation from the 50 to 100 per cent points have constantly increasing antiknock value, illustrated by specially blended fuel S-6 in Fig. 10*a*.

(3) Those whose cuts between the 50 and 100 per cent points have constantly decreasing antiknock value, such as fuel WG-1 of Fig. 10*b*.

Among commercial lead-free fuels, those of the third class are by far the most numerous; however, the degree of concentration of antiknock value in the light ends, although variable, never approaches that of specially blended fuel S-11 referred to herein (Fig. 10*b*). Commercial lead-free fuels of the first class are encountered occasionally and fuels of the third class may be changed, by the addition of lead tetraethyl, to have the characteristics of those of the first class. Those in the second class are not available commercially but may be made for research purposes by careful blending of selected components.

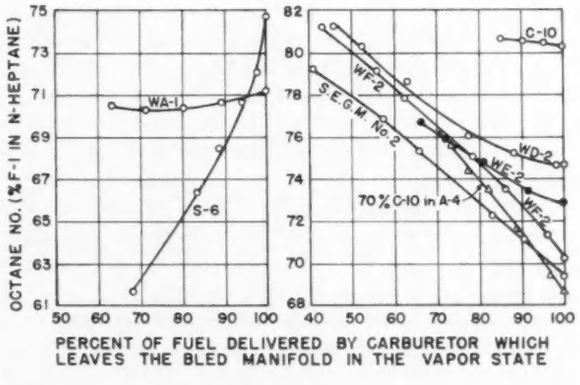


FIG. 10a. VARIATION OF THE ANTIKNOCK VALUE OF THE VAPORIZED PORTION OF 4 OF THE 1937 C.F.R. WINTER FUELS AND 4 OTHER FUELS WITH THE DEGREE OF VAPORIZATION.

C.F.R. ENGINE, SPECIAL MANIFOLD (FIG. 3c).
CYLINDER AIR-FUEL RATIO FOR MAXIMUM KNOCK.
TEMP. MIXTURE ENTERING CYLINDER, 220°F.
SPARK ADVANCE, 18°
ENGINE SPEED, 600 R.P.M.

Variation of Cylinder Air-Fuel Ratio

It has been pointed out in Table 3 that imperfect distribution causes some cylinders of the multicylinder engine to receive mixtures richer and leaner respectively than that leaving the carburetor, whereas others may receive a mixture having substantially the same composition and air-fuel ratio as that leaving the carburetor. Inasmuch as the knocking tendency of the fuel in the cylinder is influenced markedly by air-fuel ratio as well as by the composition of the fuel, it is interesting to investigate the combined effect of these two variables resulting from imperfect distribution of the fuel in the multicylinder engine. It is possible to predict for any given fuel the relative knocking tendency of the several cylinders by the construction of any one of a number of different charts after a series of tests of the fuel made on a C.F.R. engine equipped with one of the manifolds shown in Fig. 3. The type of chart which is most satisfactory depends on the information desired.

Inasmuch as the most frequent problem is the prediction of the relative knocking tendency of a group of fuels in a single car under a given set of conditions, the most valuable type of chart is usually one constructed on the basis of constant carburetor air-fuel ratio. The chart of Fig. 11 is constructed at a carburetor air-fuel ratio of 11.21 for fuel S.E.G.M. No. 2, which is a typical commercial, regular grade, lead-free base gasoline comprising a mixture of straight-run and cracked stocks. The scale of ordinates is critical compression ratio, defined as the compression ratio required for a knockmeter reading of 50. The scale of abscissae is the per cent of the fuel delivered by the carburetor which leaves the manifold in the vapor state.

Data for the construction of the curve for Cylinder *A* are obtained by operating the engine at the desired speed, spark advance, and final mixture temperature and, by adjustment of manifold temperatures, varying the per cent of the fuel delivered by the carburetor which is bled from the manifold

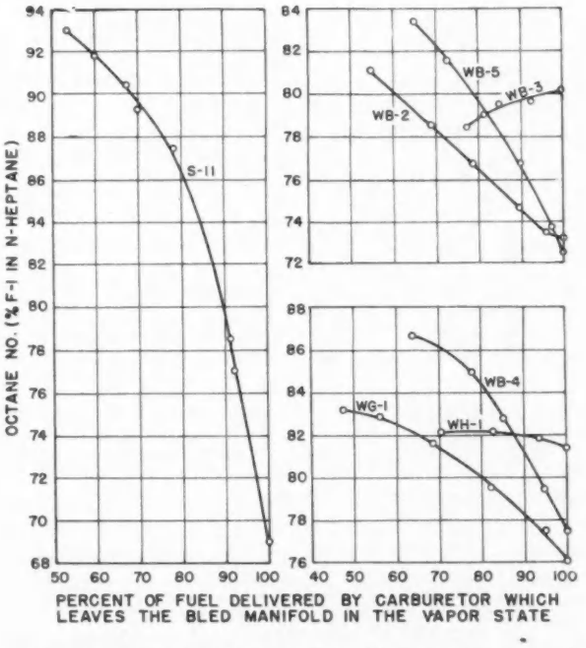


FIG. 10b VARIATION OF THE ANTIKNOCK VALUE OF THE VAPORIZED PORTION OF 6 OF THE 1937 C.F.R. WINTER FUELS AND 1 OTHER FUEL WITH THE DEGREE OF VAPORIZATION.

C.F.R. ENGINE, SPECIAL MANIFOLD (FIG. 3c).
CYLINDER AIR-FUEL RATIO FOR MAXIMUM KNOCK.
TEMP. MIXTURE ENTERING CYLINDER, 220 °F.
SPARK ADVANCE, 18°
ENGINE SPEED, 600 R.P.M.

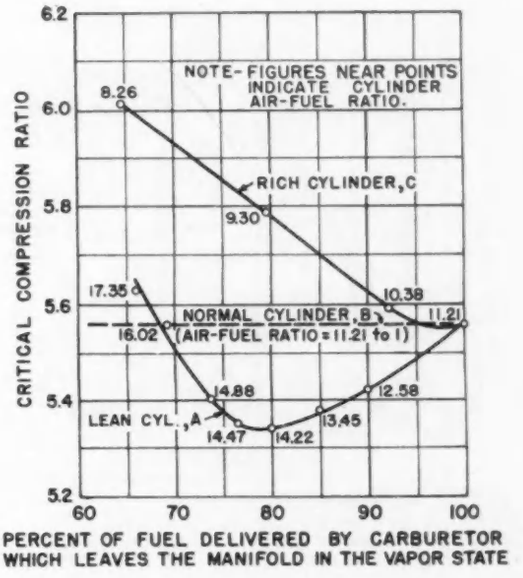


FIG. 11 EFFECT OF UNEQUAL DISTRIBUTION OF THE FUEL IN THE MANIFOLD ON THE CRITICAL COMPRESSION RATIOS OF THE INDIVIDUAL CYLINDERS OF AN ENGINE.

TYPICAL COMMERCIAL LEAD-FREE BASE STOCK.
AIR-FUEL RATIO AT THE CARBURETOR, 11.21

in the liquid state. The compression ratio for a knockmeter reading of 50 is determined for each rate of fuel bleeding and the curve is drawn through the points so determined. If the air and fuel passing through the carburetor and the liquid fuel bled from the manifold are measured, it is possible to compute the air-fuel ratio of the mixture which enters the cylinder. This information is shown beside some of the data points in Fig. 11.

The air-fuel ratio and composition of the mixture in Cylinder *A* under the condition of complete fuel vaporization would apply to the normal Cylinder *B* of the multicylinder engine regardless of the degree of fuel vaporization in the manifold. The curve for Cylinder *B* therefore is a straight line, parallel to the horizontal axis, and intersecting the curve for Cylinder *A* at the 100 per cent vaporization point.

It will be recalled that Cylinder *C* of Table 3 receives a normal charge plus the liquid heavy ends which, under conditions of perfect fuel distribution, would have entered Cylinder *A*. Data for the construction of the curve for Cylinder *C* are determined in the manner previously described. When vaporization is complete in the manifold, *H* becomes zero and the curve for Cylinder *C* has a point in common with the curves for Cylinders *A* and *B*.

Fig. 10*a* shows that the lighter fractions of fuel S.E.G.M. No. 2 have the higher antiknock value, nevertheless, Fig. 11 indicates that, under conditions of imperfect fuel distribution at a carburetor air-fuel ratio of 11.21, Cylinder *A* which receives the fuel of highest inherent antiknock value has the greatest knocking tendency of the three cylinders when 69 to 100 per cent of the fuel leaves the manifold in the vapor state. Furthermore, Cylinder *C*, which receives fuel having the lowest inherent antiknock value, has the least tendency to knock. As the degree of vaporization decreases, the air-fuel ratio of Cylinder *A* decreases from 11.21 toward the maximum knock air-fuel ratio, and this change in air-fuel ratio has a greater influence on the knocking tendency of the fuel in the cylinder than the increase in the inherent antiknock value of the fuel fraction itself. Similarly, the richening of the mixture in Cylinder *C* under the same conditions has a

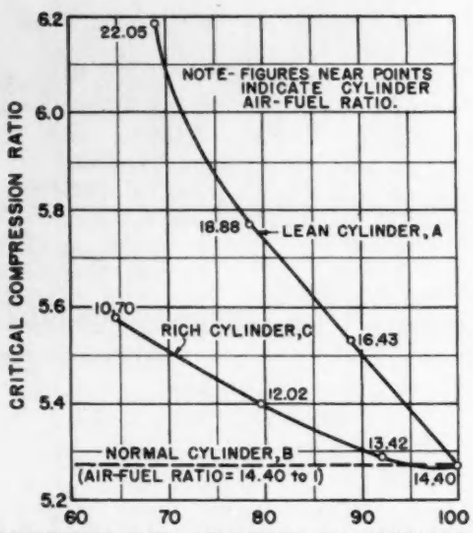


FIG. 12 EFFECT OF UNEQUAL DISTRIBUTION OF THE FUEL IN THE MANIFOLD ON THE CRITICAL COMPRESSION RATIOS OF THE INDIVIDUAL CYLINDERS OF AN ENGINE.

TYPICAL COMMERCIAL LEAD-FREE BASE STOCK. AIR-FUEL RATIO AT THE CARBURETOR, 14.40 (MAXIMUM KNOCK)

greater effect on the knocking tendency of the fuel in this cylinder than the decrease in the inherent antiknock value of the fuel.

The base fuel and conditions illustrated in Fig. 11 are those of commercial interest. As a matter of academic interest and to provide a more complete background for the fuel-distribution problem, Figs. 12 to 16 have been prepared. Fig. 12 is constructed for fuel S.E.G.M. No. 2 illustrated in Fig. 11 but at the carburetor air-fuel ratio for maximum knock. A full-throttle setting so lean is not used in cars because of the resultant loss of power at all speeds and uneven operation at low speeds. The carburetor conditions illustrated in Figs. 13 and 15 are in the normal range but the fuels for these figures, as well as for Figs. 14 and 16, are specially blended extreme types which are not representative of commercial gasolines.

Fig. 12 is similar to Fig. 11 in all respects except that the carburetor air-fuel ratio is the maximum knock ratio for fuel S.E.G.M. No. 2. At this carburetor setting the air-fuel ratio for Cylinder *A* becomes leaner than that for maximum knock as the degree of fuel vaporization is decreased. Simultaneously the inherent antiknock value of the fuel in the cylinder increases. The result is a marked rate of increase in critical compression ratio as the degree of vaporization decreases.

Under the same conditions, the air-fuel ratio of the mixture in Cylinder *C* becomes richer than that for maximum knock and for this reason the fuel tends to knock less. At the same time, however, the inherent antiknock value of the fuel decreases. The result of these two opposed tendencies is that, as the degree of vaporization decreases, the knocking tendency of Cylinder *C* at first remains close to that of normal Cylinder *B*, but becomes less than that for Cylinder *B* as the degree of vaporization is decreased still further.

The differential in antiknock value between the light and heavy ends of fuel S.E.G.M. No. 2 is within the normal range for commercial fuels. Fuel S-11 was specially blended to have light and heavy ends of extremely high and low antiknock value respectively. This abnormal distribution of antiknock value through the fuel fractions reverses the relative knocking tendency of the rich and lean cylinders. Fig. 13 shows for this fuel the relative knocking tendency of the three typical cylinders, *A*, *B*, and *C*, when the carburetor air-fuel ratio is 11.46. The difference between the knocking tendency of the light and heavy fractions of this fuel is so great that, as the air-fuel ratio of Cylinder *A* moves from 11.46 toward the maximum knock air-fuel ratio, the increase in inherent antiknock value of the fuel in the cylinder almost exactly balances the effect of the change in cylinder air-fuel ratio. At cylinder air-fuel ratios leaner than for maximum knock, both influences tend to decrease knocking and the curve for Cylinder *A* rises rapidly.

As the degree of vaporization is decreased, the knock-suppressing effect of the richer mixture in Cylinder *C* is not sufficient to compensate for the depressing effect of the additional heavy ends of very low antiknock value. Rich Cylinder *C*, therefore, has the greatest knocking tendency, relative to that of Cylinders *A* and *B*, when vaporization is between 71 and 100 per cent.

When the carburetor air-fuel ratio is that for maximum knock (Fig. 14), the rich Cylinder *C* still has the greatest knocking tendency, relative to Cylinders *A* and *B*, between vaporization rates of 72 and 100 per cent. Fig. 14 shows, however, that its knocking tendency does not greatly exceed that of the normal Cylinder *B*.

Fig. 15, similar to Figs. 11 to 14, is constructed at a carburetor air-fuel ratio of 11.03 for fuel S-6 which was specially blended to have light and heavy ends of extremely low and high antiknock value respectively. As the rate of fuel vaporization in the manifold is decreased, the inherent antiknock value of the fuel in lean Cylinder *A* is decreased, and the air-fuel ratio of the mixture moves nearer to that for maximum knock, consequently the knocking tendency increases rapidly. Under the same conditions, as the mixture in rich Cylinder *C* moves farther away from the maximum knock point, the inherent antiknock value of the fuel in the cylinder increases. The two effects combine to cause the knocking tendency of the fuel in Cylinder *C* to diminish rapidly. It should be noted

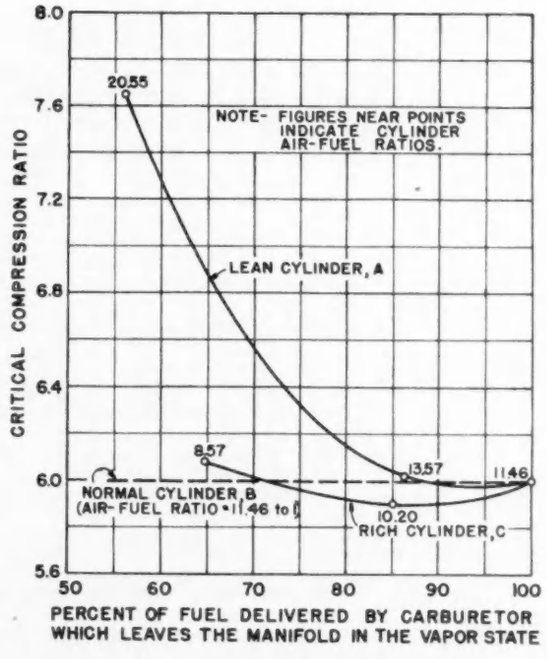


FIG. 13 EFFECT OF UNEQUAL DISTRIBUTION OF THE FUEL IN THE MANIFOLD ON THE CRITICAL COMPRESSION RATIOS OF THE INDIVIDUAL CYLINDERS OF AN ENGINE.

FUEL S-11, SUMMER VOLATILITY, EXTREME CONCENTRATION OF ANTIKNOCK IN LIGHT ENDS. AIR-FUEL RATIO AT THE CARBURETOR, 11.46

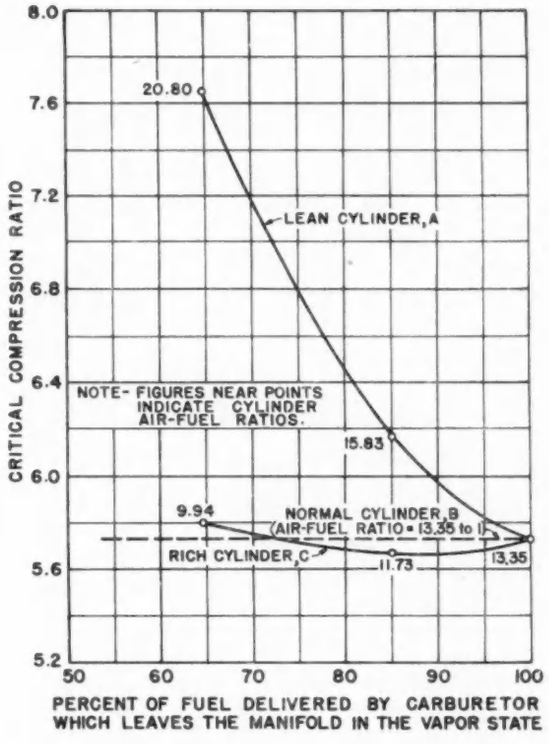


FIG. 14 EFFECT OF UNEQUAL DISTRIBUTION OF THE FUEL IN THE MANIFOLD ON THE CRITICAL COMPRESSION RATIOS OF THE INDIVIDUAL CYLINDERS OF AN ENGINE.

FUEL S-11, SUMMER VOLATILITY, EXTREME CONCENTRATION OF ANTIKNOCK IN LIGHT ENDS. AIR-FUEL RATIO AT THE CARBURETOR, 13.35 (MAXIMUM KNOCK)

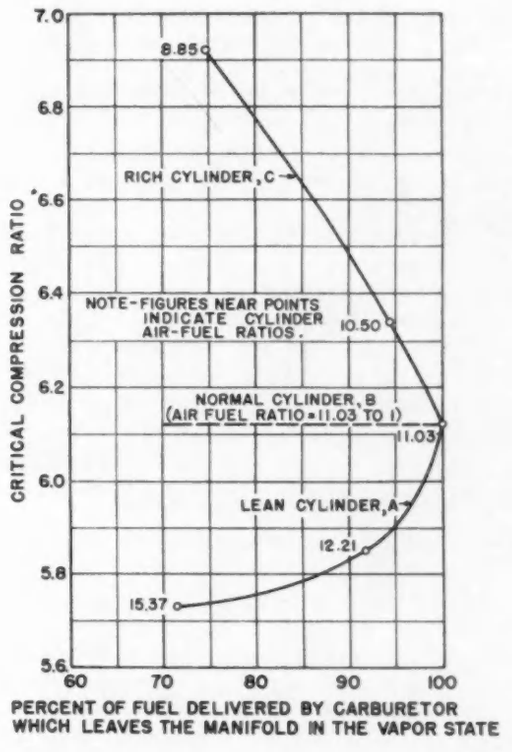


FIG. 15 EFFECT OF UNEQUAL DISTRIBUTION OF THE FUEL IN THE MANIFOLD ON THE CRITICAL COMPRESSION RATIOS OF THE INDIVIDUAL CYLINDERS OF AN ENGINE.

FUEL S-6, WINTER VOLATILITY, EXTREME CONCENTRATION OF ANTIKNOCK IN HEAVY ENDS. AIR-FUEL RATIO AT THE CARBURETOR, 11.03

that at 75 per cent vaporization the difference between the critical compression ratios for Cylinders *A* and *C* is 1.19 ratios, equivalent roughly to 20 octane numbers.

Fig. 16 shows that, when the carburetor air-fuel ratio for fuel S-6 is that for maximum knock, the lean Cylinder *A* has a slightly greater tendency to knock than the normal Cylinder *B* when the rate of vaporization is between 90 and 100 per cent, but that Cylinder *B* knocks most at lower rates of vaporization. The effect of carburetor air-fuel ratio is quite pronounced for this fuel.

Fuels of commercial interest have a distribution of antiknock value along their distillation curves which lies between but does not approach that for fuels S-6 and S-11. A study of the charts for these two fuels, together with those for fuel S.E.G.M. No. 2, should permit a prediction of the behavior of fuels having various degrees of concentration of antiknock value in the light and heavy ends.

Some mention should perhaps be made of a hypothetical fuel, all of whose fractions have equal inherent antiknock value. The knocking tendency of the fuel in any cylinder would depend only on the air-fuel ratio with respect to that for maximum knock. At the carburetor air-fuel ratio for maximum knock, the normal Cylinder *B* would have the greatest knocking tendency, regardless of the degree of vaporization. At a normal carburetor air-fuel ratio, which would be richer than that for maximum knock, the knocking tendency of Cylinder *C* would decrease with a decrease in the rate of vaporization, whereas Cylinder *A* would knock increasingly more as its air-fuel ratio approached that for maximum knock.

Effect of Atmospheric Temperature

In the case of an engine whose choke and mixture-heating valves are not adjusted automatically or manually as the atmospheric temperature changes, the effect of a decrease in atmospheric temperature is two-fold, as follows:

(1) To lean the carburetor mixture.

(2) To decrease the vaporization of the fuel in the carburetor and intake manifold and to widen the differential between the air-fuel ratios of the richest and leanest cylinders.

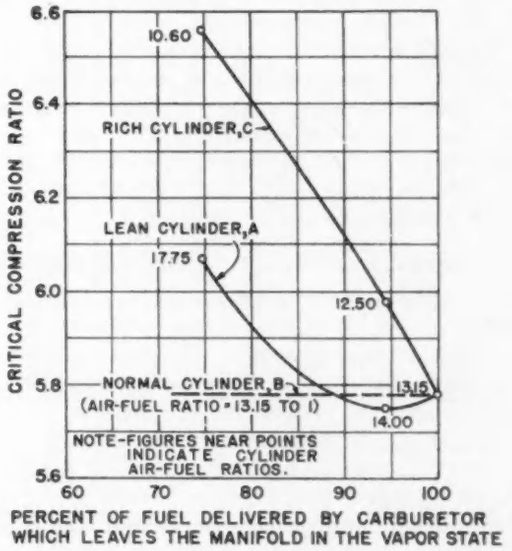


FIG. 16 EFFECT OF UNEQUAL DISTRIBUTION OF THE FUEL IN THE MANIFOLD ON THE CRITICAL COMPRESSION RATIOS OF THE INDIVIDUAL CYLINDERS OF AN ENGINE.

FUEL S-6, WINTER VOLATILITY, EXTREME CONCENTRATION OF ANTIKNOCK IN HEAVY ENDS. AIR-FUEL RATIO AT THE CARBURETOR, 13.15 (MAXIMUM KNOCK)

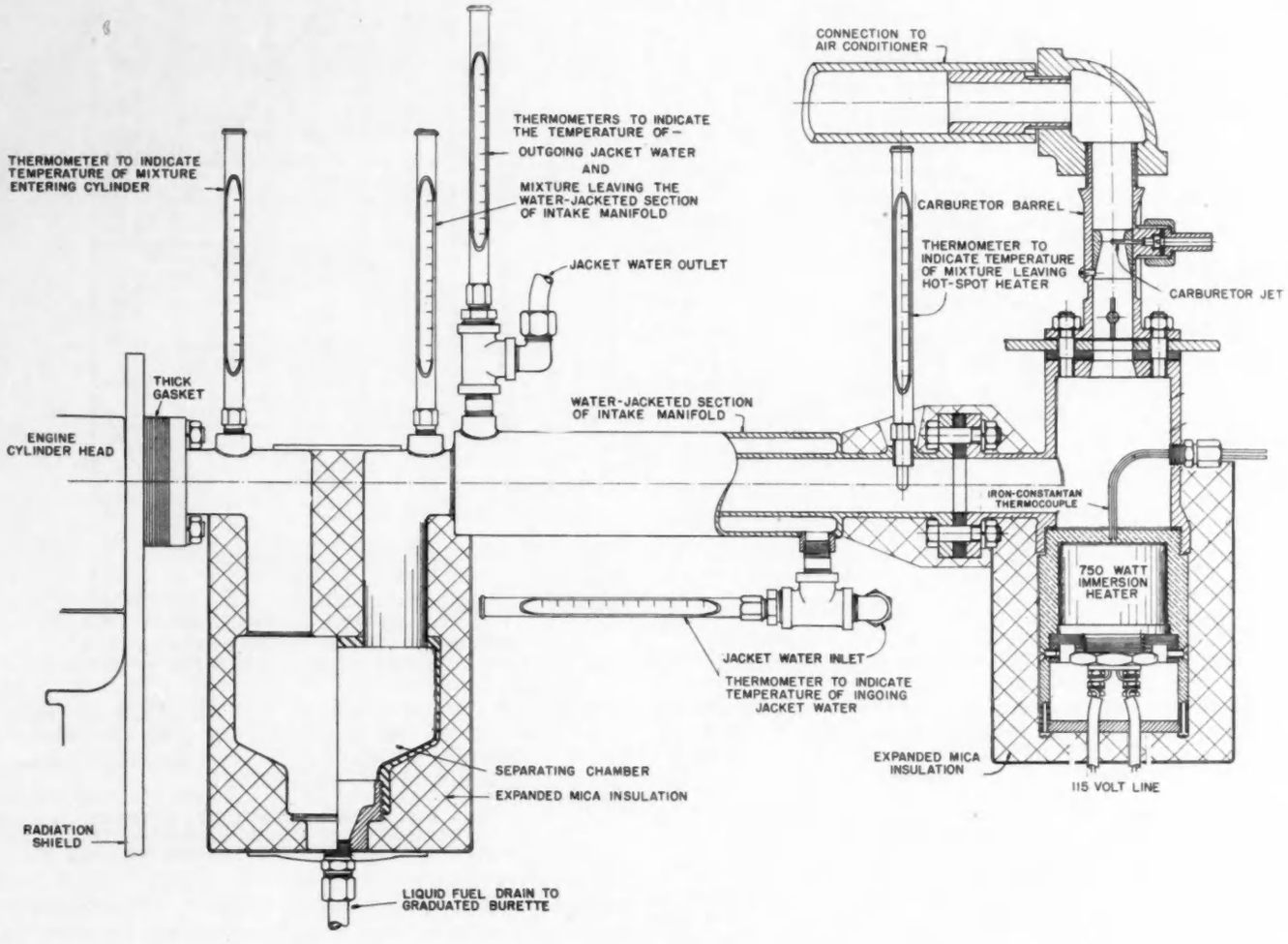


FIG. 17 MODIFIED SINGLE CYLINDER ENGINE MANIFOLD AND ASSOCIATED EQUIPMENT FOR THE STUDY OF THE RELATIVE EFFICIENCY OF AIR AND MIXTURE HEATING IN THE VAPORIZATION OF FUEL

Obviously a reduction in atmospheric temperature, in the absence of adjustment of choke or mixture heat-control valve, may result in marked changes in the relative ratings of fuels which bear no relation whatever to the sensitivity of the fuels as determined by the differences between their Research and Motor Method ratings. This point is confirmed by a study of the data from recent road knock tests.

Adjustment of Choke and Mixture Heat Control

Some form of choke, either manual or automatic, always is installed on the car engine. There is usually also a manual or automatic heat-control valve. During the warm-up period in the winter the effect of the choke is to decrease the air-fuel ratio considerably. Under these conditions the degree of vaporization is low, and the relative knocking tendency of the several cylinders of the engine may be predicted by a study of Figs. 11, 13, and 15.

After the warm-up period the choke is usually not an important factor in the knocking problem, but this condition is not true of the mixture heat-control valve, especially if automatically actuated. As applied to one well-known car, the valve bypasses the maximum heat to the intake manifold when the temperature of the air surrounding the manifold is low, but applies progressively less heat as the temperature under the hood rises until, at an air temperature of 75 deg. fahr. near the thermostat, the valve is closed entirely. The temperature of the air at the thermostat is about 60 deg. fahr. higher than that of the air outside the hood. If an outside air temperature of 100 deg. fahr. and a corresponding temperature under the hood of 160 deg. fahr. are required for complete vaporization, a substantial portion of the fuel leaves the manifold in the liquid state when the mixture heat-control valve closes. Vaporization may actually be less when the valve closes than at some lower outside air temperature.

The relative efficiency of heat in vaporizing the fuel, when applied to the carburetor air and to the mixture through a hot-spot, immediately becomes of much interest. Fig. 17 shows diagrammatically a modified single-cylinder engine intake manifold, together with associated equipment, designed for a study of these effects. The manifold has a long horizontal section and a U-bend, the latter being provided with an opening through which liquid fuel flows to a graduated burette. After flowing downward through the vertical carburetor, the fuel and air mixture is forced to turn a corner. Directly under the carburetor is the electrically heated hot-spot against which liquid fuel impinges as it turns the corner.

A section of the manifold is surrounded by a jacket through which water of varying temperature may be circulated for heating of the mixture. Air for the carburetor is supplied by an air-conditioner which is capable of delivering air having a range of temperature and humidity.

Fig. 18 shows for one commercial gasoline the relation

between temperature of the mixture entering the cylinder and the degree of fuel vaporization when the heat is supplied alternately to the carburetor air and to the hot-spot. Between the limits of no heat applied to the hot-spot and complete vaporization of the fuel by either method of heating, the mixture at any temperature leaving the manifold contains a greater fraction of the fuel delivered by the carburetor when the heating is done by the hot-spot. For example, at a final mixture temperature of 85 deg. fahr., 88 per cent of the fuel passes into the cylinder with the air when the carburetor air is heated, but the quantity increases to 94 per cent when the heat is applied by the hot-spot. Similarly, complete vaporization of the fuel is accomplished at a lower mixture temperature in the case of hot-spot heating.

Variations in Ignition Timing Between Cylinders

The relative knocking tendency of the cylinders of a multicylinder engine varies with carburetion and fuel distribution, as illustrated in Figs. 11 to 16, provided that spark advance and cylinder cooling are uniform. Because of the longer cylinder jackets and better direction of flow of cooling water in recently built car engines, it is believed that variations in knocking among the cylinders due to differences in cooling have, in general, been minimized. Variations in spark advance appear to create a greater problem. For example, the leanest cylinder may have the greatest knocking tendency because of partial fuel vaporization and imperfect distribution, as illustrated in Fig. 11. If, however, the spark advance in this cylinder during knocking is retarded relative to that in the other cylinders, the most pronounced knock may occur in some other cylinder. In any study of the effects of fuel distribution it is desirable, therefore, to be able to measure the ignition advance in each cylinder simultaneously.

Fig. 19 shows a recently developed instrument for this purpose. A drum weighing 20 lb. and designed to hold a strip of 35-mm. cinema film on its outside surface is driven at a constant speed of 75 r.p.m. by a 6-volt electric motor. A neon tube is enclosed in a small light-tight box which is quite close to the drum. A narrow slot in the box, parallel to the axis of the film drum, permits the flash of the neon tube to appear as a line on the film, perpendicular to its length. The tube lights as a result of the voltage induced by the breaking of the primary circuit of the ignition system and by the motion of two flywheel studs, 180 deg. apart, past a magnetic pickup mounted on the flywheel housing. The pickup is similar in design to that employed by the Chrysler Corp. in its instrument for the measurement of mean spark advance in the cylinders. The induced voltage is amplified by the triode section of a duplex diode-triode vacuum tube (Type 6R-7) and the amplified output is coupled to the grid of a power tube (Type 6A-3). The neon tube is connected in parallel with the load resistance of the power tube. It is lighted when the voltage across the load resistance of the power tube rises to approximately 90 volts and is extinguished when the voltage applied to the grid of the power tube decreases due to termination of the signal impulse.

The oscillatory impulse from the primary of the ignition coil due to the opening of the breaker points is rectified and amplified in the Type 6R-7 tube and applied to the grid of the power tube, producing a flash of the neon tube and a mark on the film for the firing of each cylinder. A portion of the high-voltage impulse to No. 1 spark-plug is amplified and applied to the grid of the power tube, resulting in a flash of longer duration which identifies the film mark for No. 1 cylinder. One revolution of the film drum covers a number of complete engine cycles. A ratchet-type relay limits the exposure to one revolution of the film drum.

Ignition timing of the individual cylinders is read from a curve of angular displacement of the crankshaft versus time, obtained from the flywheel reference points.

Knocking Tendency of Individual Cylinders

The knocking tendency of the individual cylinders of a car engine may be determined by the aid of the auxiliary breaker used with the gas sampling valve as shown in Fig. 2. When the knocking of No. 1 cylinder is to be measured, the regular distributor is retarded and the auxiliary breaker is set to produce a timing in No. 1 cylinder which is the same as that given by the regular distributor when adjusted properly. After the knocking tendency of No. 1 cylinder has been measured, the auxiliary breaker is used to produce the correct ignition timing in Cylinder No. 2, and so on for the remainder of the cylinders. Since the auxiliary breaker is mounted on the same shaft as the regular breaker, it duplicates the advance characteristics of the regular breaker.

Reference Fuels

The scale of ordinates for Figs. 11 to 16 is critical compression ratio. For some purposes it may be desirable to substitute an octane-number scale, but many such scales are possible, depending on the choice of reference fuels and the conditions under which they are tested. For example, either the primary or secondary reference fuels might be used. They might be run always at the maximum knock air-fuel ratio or at the same carburetor setting as for the fuel under test. A blend of the secondary reference fuels equal to a given blend of the primary reference fuels at the maximum knock air-fuel ratio may be far from equal at another air-fuel ratio.

The reference fuels may be run under conditions of complete vaporization or under the manifold conditions employed for the fuels under test. In the usual tests of commercial

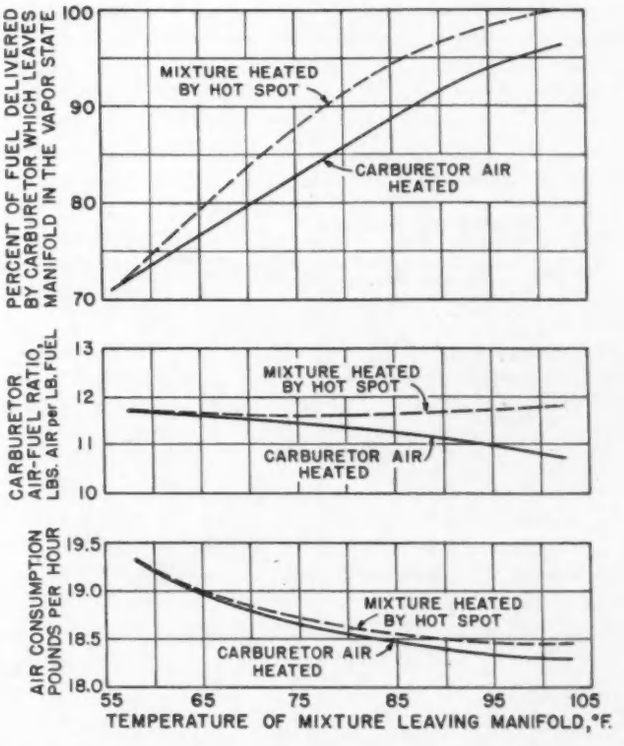


FIG. 18. EFFECT OF HOT SPOT MIXTURE HEATING AND CARBURETOR AIR HEATING ON THE VAPORIZATION OF FUEL IN THE MANIFOLD.

SERIES 308 ENGINE WITH SPECIAL MANIFOLD AND EQUIPMENT OF FIG. 17. FIXED CARBURETOR SETTING ENGINE SPEED, 900 R.P.M.

gasolines, only a small fraction of the primary reference fuels fails to vaporize, even when the rate of vaporization of the test fuels is low. Under the same conditions, a much larger fraction of the secondary reference fuel blends remains in the liquid state and the unvaporized portion is composed largely of fuel A-5. The equivalent octane numbers of single points on the curves of Figs. 11 to 16 may vary as much as 15 octane numbers, depending on the choice and method of testing of the reference fuels. Curves of octane number versus per cent of vaporization may have shapes entirely different from the corresponding curves of critical compression ratio versus per cent of vaporization.

There is every reason to believe that the complication introduced into the interpretation of laboratory octane numbers obtained under conditions of partial fuel vaporization enters also into the determination of road octane numbers. Regular-grade gasolines require the use of the C-10 and A-5 reference fuels. As the per cent of the A-5 component in the blend is increased, the fraction of the blend which is vaporized in the manifold should decrease, causing the leanest cylinder to become leaner and the richest cylinder richer. The cylinder which knocks most on the test fuel may not knock most on the reference-fuel blend, and the cylinder which knocks most on a blend containing a high per cent of C-10 may not knock most when the per cent of C-10 is reduced.

If the volatility of the two secondary reference fuels were interchanged, the knocking tendency of a given percentage blend under any condition of imperfect distribution obviously might be markedly different. An increase in the concentration of A-5 in the blend would then *increase* the vaporization in the manifold with a resultant *decrease* in the inherent anti-knock value of the vaporized portion. Such effects introduced into the road knock-testing procedure by the difference in volatility of the reference fuels appear in the road ratings of the test fuels as apparent appreciation or depreciation of these fuels with respect to the laboratory ratings.

Once the observation is made that road octane numbers determined under conditions of imperfect manifold vaporization and fuel distribution are influenced by the choice of secondary reference fuels, a consideration of desirable characteristics of secondary reference fuels assumes real importance. In the light of the previous discussion it appears that, from the standpoint of minimizing the complications introduced into road knock testing by the secondary reference fuels, such fuels ideally should have the following characteristics:

(1) The distillation curve of the fuel of high antiknock value should match exactly that of the fuel of low antiknock value in order to provide constant vaporization in the manifold as the proportions of the two fuels in the blend are changed.

(2) The volatility of the reference fuels should approximate that of the average commercial gasoline, thereby ensuring, in so far as possible, the same cylinder air-fuel ratios for the test and reference fuels.

(3) All fractions of each fuel should have equal antiknock value in order to prevent a change in the inherent antiknock value of the vaporized portion of the blend in the manifold as the degree of vaporization is changed. Inasmuch as commercial gasolines have light ends whose antiknock value is higher or lower than or equal to that of the heavy ends, the distribution of antiknock value in the reference-fuel blends would then approximate that of the average of fuels met in service.

These specifications constitute a large order which will probably never be filled. It should, however, be possible to produce secondary reference fuels which, for use under conditions of partial vaporization, are an improvement over the present A-5 and C-10 fuels. The latter fuels are entirely satisfactory for use in the laboratory in conjunction with the standard C.F.R. engine manifold and were almost equally suitable for road-test purposes until the fuel-distribution prob-

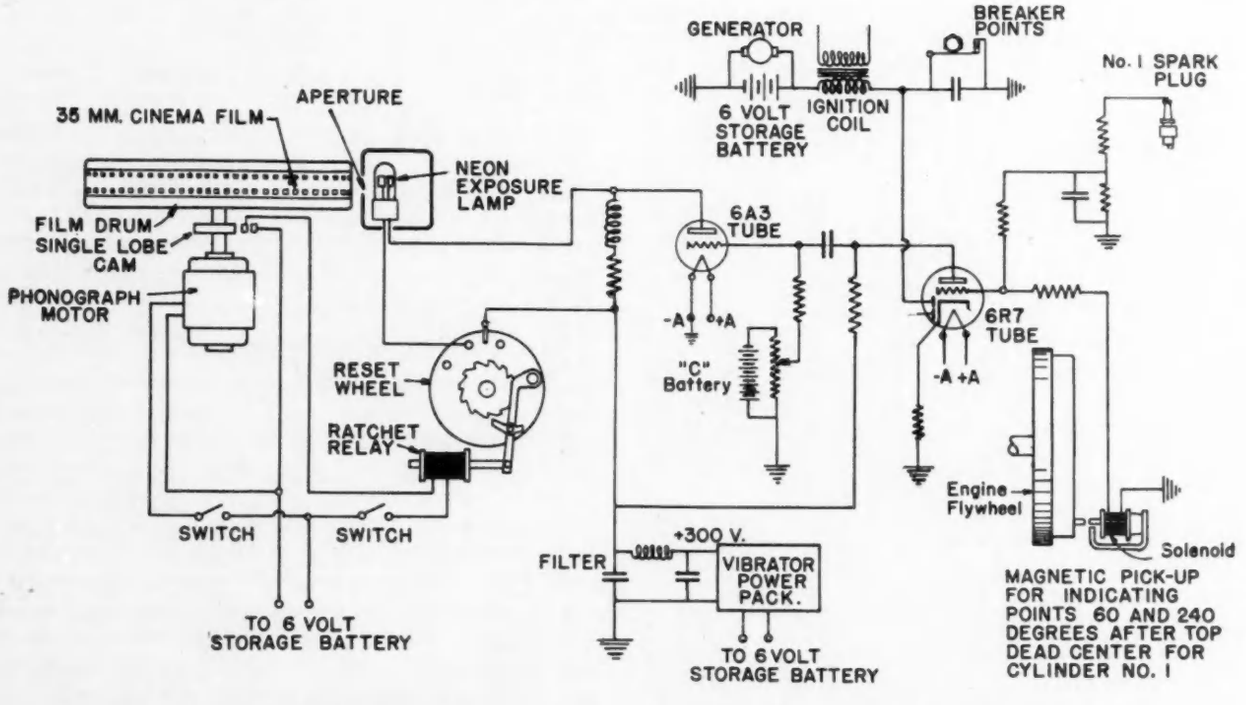


FIG. 19 DIAGRAMMATIC SKETCH OF AN INSTRUMENT FOR RECORDING ON CINEMA FILM THE IGNITION TIMING IN THE INDIVIDUAL CYLINDERS OF A MULTICYLINDER ENGINE.

lem attained its present importance. It should be noted that the primary reference fuels substantially fulfill the first and third of the above-described specifications.

Correlation of Road and Laboratory Ratings

The correlation of laboratory with road knock ratings creates two distinct and basically different problems, namely:

(1) Reproducing on the laboratory engine the conditions of operation of a single car engine in a single test.

(2) Determining a laboratory procedure which gives ratings that agree in so far as possible with the averages of ratings obtained in a group of cars under one or more conditions of test.

The first problem is by far the simpler. If every variable of engine operation such as mixture and cylinder temperatures, spark advance, speed, fuel composition, and mixture ratio, applicable to the knocking cylinders of the multicylinder engine is duplicated on the laboratory engine, the ratings of all types of fuels should be duplicated provided, of course, that the same reference fuels are used in both cases.

If partial fuel vaporization in the intake manifold of the car engine gives rise to differential distribution of unvaporized liquid fuel which, in turn, causes a cylinder richer or leaner than the normal cylinder to have the greatest knocking tendency, the knock ratings in the car engine of a wide variety of fuels can not be duplicated on the laboratory engine unless provision is made for the simulation of the mixture distribution of the multicylinder engine by some means such as that described herein. A fortuitous compensation for the effects of mixture distribution may be made for a limited number of fuels by changes in other engine variables, but such an arrangement is almost certain to fail when applied to a large group of fuels.

A modification of the conventional laboratory knock-testing procedure by the substitution of a manifold designed to simulate the action of the intake manifold of the car under consideration should go a long way toward explaining the anomalous behavior of fuels observed from time to time in a single car.

Before an attempt is made to duplicate on the laboratory engine the ratings obtained on a given car, it is believed advisable to construct a chart similar to Fig. 20 in which air-fuel ratio, spark advance, and cylinder knock intensity are plotted against cylinder number. Such a chart not only indicates the cylinder of greatest knocking tendency and its air-fuel ratio, but furnishes the data required for the computation of average air-fuel ratio and the degree of vaporization. The information on ignition timing may serve to explain the greatest knocking tendency of one cylinder when, under conditions of uniform ignition timing, the laboratory engine indicates that a cylinder having a different mixture condition should knock most.

The second problem – that of finding a set of operating conditions for the laboratory engine which gives ratings that agree with the average of ratings in a group of cars – is quite different. The average road ratings reflect the results of the simultaneous operation of a host of variables combined differently in each car. Considering alone those variables which arise through variations in carburetion and fuel distribution in the engine, it is obvious that a fuel might be rated in one car under conditions of substantially perfect distribution and at an air-fuel ratio near that for maximum knock whereas, in another car, distribution might be equally good but the air-fuel ratio in all cylinders might be considerably richer than that for maximum knock. In a third car with imperfect fuel distribution the rating might be determined by the behavior of the portion of the fuel leaving the manifold in the vapor

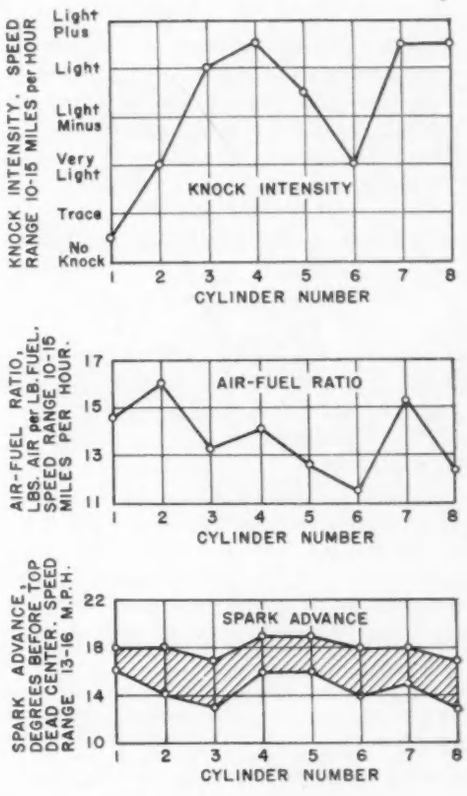


FIG. 20. VARIATION OF AIR-FUEL RATIO, SPARK ADVANCE, AND KNOCK INTENSITY AMONG THE CYLINDERS OF AN AUTOMOBILE ENGINE IN THE SPEED RANGE OF MAXIMUM KNOCKING.

state and burned at the air-fuel ratio for maximum knock. The list of possibilities is far from exhausted when it becomes apparent that no single set of carburetor and manifold conditions for the laboratory engine theoretically should give ratings equal to the average of ratings obtained on several cars having a wide variation in carburetion and manifold conditions.

Notwithstanding this situation, definite and consistent effects of non-uniform fuel distribution are now apparent in any fuel knock-rating data obtained on a group of cars. One of the most impressive demonstrations of the importance of this variable was furnished by a series of road tests in which the 12 test fuels were specially blended to have an extreme concentration of antiknock value in different portions of the distillation range. Fuels S-6 and S-11, previously mentioned, were two of the test fuels.

After the effects of imperfect fuel distribution within the engine have been found to exist, the problem of most immediate interest is the development of some means for the correction of variations in the road ratings of fuels, rated as equal by present laboratory methods, due to this variable. The British have for some time used a correction factor for benzol blends. For the first 30 per cent of the total aromatics in any gasoline, 1 octane number is added to the conventional laboratory rating and, for each additional 15 per cent of aromatics, another octane number is added. A correction factor applicable to all types of fuels and capable of direct determination on the engine would be desirable.

It is to be hoped that automotive engineers will be able to find a not-too-expensive means for minimizing variations in fuel distribution at low engine speeds. In the meantime the oil refiner will undoubtedly make an effort to correct, by some change in the laboratory knock-testing procedure, present

inequalities in the road performance of fuels of equal octane number arising because of the mixture-distribution problem.

The addition of concentrated knock suppressors to fuels may change the distribution of antiknock value through the fuel fractions. The choice of antiknocks to meet the wide range of conditions resulting from variations in carburetion and manifolding is a complex problem in itself. It is hoped that this subject may be discussed in a later paper.

Discussion

Stresses Need for Further Study

– L. E. Hebl

Shell Petroleum Corp.

WE must recognize that, in our study of detonation, none of us has followed the scientific method more closely than has Mr. Bartholomew, whose evaluation and classification of the variables involved has been relentless. He was probably the first to begin classifying primary and secondary variables when in 1929 he said: "In fact the variation in ratings with changes in speed, spark advance, compression ratio, throttle opening, and so on can, in almost every case, be traced to a difference in temperature conditions."[a] Now again he pioneers in demonstrating that not only those variables influencing the manner in which the mixture burns within the cylinder are significant, but also account must be taken of the variables that affect the composition of the mixture that enters the cylinder.

Mr. Bartholomew's paper comes at a very opportune time when the automobile and oil industries are disturbed by the realization that the present test method does not rate fuels in the correct order of their performance in multicylinder engines. Although the Cooperative Committee has already carried out some tests which could probably be used as a basis for modifying the test method, it would be very unwise to proceed with the adoption of a modified test until Mr. Bartholomew has had time to examine the manifold phenomena more thoroughly and to learn whether a practical correction for these phenomena can be applied to the laboratory test.

Briefly, it now appears that the manifold phenomena tend to lower the ratings in multicylinder engines and, in order to obtain agreement, the single-cylinder laboratory engine has been operated at excessively high temperatures so that ratings in it also are reduced to match the average road values. However, the high-temperature affects cracked gasolines mainly, whose ratings are reduced too much in the laboratory, whereas ratings on straight-run gasolines, which are insensitive to temperature, are not reduced enough. The resulting deviations are thus both positive and negative so the average appears to be correct; however, if the errors on individual fuels are to be corrected, it is obviously necessary to reduce the temperature of the laboratory engine to a value in line with multicylinder engines and to correct for the lower multicylinder ratings by correcting the same factors that cause those ratings to be low. Mr. Bartholomew has demonstrated a possible source of those errors and it seems logical that a full understanding of them is necessary before further action should be taken.

Caution Urged in Interpreting Results

– A. J. Blackwood

Standard Oil Development Co.

WE wish to throw out a word of caution regarding one phase of the discussion and that is "to watch every step when interpreting volatility and octane number of the different fuel fractions as related to road behavior."

It would appear strange that, at this relatively late date, fuel volatility and octane-number distribution should assume any great importance not hitherto apparent, particularly in view of the reasonably good correlations existing between laboratory fuel ratings and road performance. Fuels which in the past have not given the expected road performance as predicted from laboratory ratings have not been, as a rule, fuels having unusual volatility or unusual octane-number distribution throughout the boiling range.

Some work conducted at our laboratories along somewhat similar lines to the multicylinder work described in the paper has indicated that the qualitative distribution factor may not be anywhere near as important as we had previously anticipated. For example, individual exhaust-gas analyses were made on an eight-cylinder engine when burning blends of reference fuels C-10 and A-4. For a range covering the 50-50 blend up through straight C-10 fuel, the air-fuel ratio of any given individual cylinder remained constant although the various cylinders differed widely in themselves. In other words, if say No. 1 cylinder received a 14:1 mixture ratio and No. 7, a 11:1 mixture ratio of the 50-50 blend, they received the same mixture ratios 14:1 and 11:1 with the straight C-10. Keep in mind that these two reference fuels differ widely in volatility and antiknock value.

Now, coincident with the determination of mixture ratio, values for the apparent octane-number requirement of the individual cylinders were determined. These data on the 8 individual cylinders, when plotted against the corresponding air-fuel ratio, gave a smooth curve very closely similar to what would have been obtained if the octane-number requirement of a single-cylinder engine were determined for different mixture ratios. These data were all obtained at full throttle at the speed of maximum knock.

This result would indicate that there had been no appreciable deviations due to qualitative distribution but rather that, if the vaporized fuel is distributed unevenly to the various cylinders, the liquid particles are distributed over the same general pattern.

Therefore, we urge that haste be made slowly when attempting to tie fuel volatility in with road antiknock behavior.

[a] See *Oil and Gas Journal*, Oct. 17, 1929, p. 32: "Antiknock Testing and Performance," by Earl Bartholomew.

The Reduction of Piston-Ring and Cylinder Wear

(*Continued from page* 140)

nitrided-steel aircraft cylinder. The structure of this material was similar to the *M* ring and *M* cylinder material, except that it was considerably harder. Standard ring material was run on one of these samples and scuffed at 2700 lb. per sq. in., 600 r.p.m. Material from the ring sample cold-welded onto the cylinder surface. When metal-to-metal contact occurred between the two structures, the cylinder material would not abrade. Particles were, therefore, torn out of the ring material which was a very much weaker structure and so coarse that it would not abrade in fine particles. The particles that cold-welded onto the cylinder surface and the particles torn loose that did not cold-weld to the cylinder surface acting as an abrasive wore the ring sample at a very fast and increasing rate.

The Ferrox ring sample that was run on a nitrided steel cylinder sample did not scuff nor cold-weld to the cylinder up to the capacity of the machine. The surface of the Ferrox ring sample abraded very slightly, producing a polishing compound that polished the cylinder sample and, at the same time, polished itself. The surface temperature of the cylinder was so high that slightly distorted areas upon which the pressure from the ring sample would naturally concentrate showed indigo color. Even though the surface temperature was extremely high and not lubricated, scuffing did not occur. The surface of the Ferrox ring sample looked like it had not been run except for narrow polished streaks around the ring face on the tops of the tool marks. Even with the maximum pressure of the machine, 4000 lb. per sq. in., concentrated on the top of the tool marks, scuffing did not occur because the first movement of the machine started a polishing action between the two materials – wear at a reducing rate. The standard ring material on the nitrided-steel cylinder sample wore at such a fast rate that the surface of the ring sample was removed in a very few minutes after scuffing started to the extent that the thermocouple hole showed through the face of the ring.

Wear of materials at an increasing rate decreases the effectiveness of lubrication and decreases their load-carrying capacity. Wear at a decreasing rate improves the effectiveness of lubrication and increases the load-carrying capacity of the surfaces.

Truck Maintenance Problems

By J. L. S. Snead, Jr.

Equipment Engineer, Consolidated Freight Lines, Inc.

TRUCK maintenance problems differ in many respects from those encountered in bus maintenance. In a truck, high speeds, exterior beauty, and quiet operation are relatively unimportant. Rigidly enforced laws govern length, height, width, weight, and distribution of load. The character of the work makes a scheduled hour for maintenance impossible. Most service work is performed while the equipment is being loaded or unloaded.

In the maintenance of a fleet of heavy-duty trucks used in intercity transportation the first problem is to set up standard maintenance operations and periods at which they are to be performed. They should be set up conservatively and changed as experience warrants. On the basis of our experience we now grease all road equipment at 1000 miles; give it a thorough inspection service and tightening at 2000 miles; adjust the valves at 2000 miles; change oil (on trucks equipped with filters) at 3000 miles; repack wheel bearings at 15,000 miles; change transmission and differential lubricants at 20,000 miles; and check valves, pistons, rings, sleeves and bearings at 45,000 miles. These maintenance operations are performed when the mileage is reached regardless of apparent condition. All other work is performed as required. We have a complete record of all maintenance work done and the mileage run by all units or parts, such as brake lining, and so on.

Many Diesel Trucks

Most of our road equipment is powered with Cummins Diesel motors. We have in service 115 motors which have run a total of over 25,000,000 miles. At present we are adding over 800,000 miles per month to our experience. Ten motors have already run in excess of 400,000 miles each. Many more have run well over 300,000 miles.

As a result of our experience with these motors we have adopted standard practices covering their maintenance and operation. There is no question but that Diesel motors are far more sensitive than the gasoline motors used in the past. One fact that we have established beyond any possible doubt – the Cummins motor must be driven within the range of 1500-1800 r.p.m. for really successful operation, and water temperature should be kept between 160 and 180 deg. fahr. Lugging results in incomplete combustion, excessive cylinder wear, bearing failures, stuck rings, excessive deposits of sludge and carbon plugged injectors, and of course, in loss of power due to them. Our experience has proved that a truck which has been driven several hundred miles at motor speeds below 1500 r.p.m. will carbon up, miss, and perform very poorly. The same truck can then be taken out and driven several hundred miles at motor speeds between 1600-1800 r.p.m. (wherever possible) with amazing results. The motor will clean up and, after a few hundred miles performance, will return to normal.

One more rule must be observed in driving these motors. The motor must never be used as a brake at speeds in excess of 1600 r.p.m. We have found this rule to be very important.

The fuel pump setting is vital. We have found that a motor receiving more than 32 cc. (according to Cummins standard fuel-pump test) will show rapid cylinder wear, dilution, and excessive deposits of sludge and carbon. The power is greater with more fuel, but the results on maintenance cost outweigh the advantages of the additional power. A truck so geared that its motor must operate for long periods at speeds lower than 1500 r.p.m. should not be given over 30 cc. Even then it will show greater cylinder wear than a truck so geared that its motor must be turned at higher speeds.

Timing and adjustment must be exactly right for proper performance. Even a slight error in setting will result in a lifeless or ragged motor. The injector timing is very important. So is the cup wiper clearance. Both should be checked at each 45,000 mile service and set to factory specifications. Valve and injector settings must be accurate. The water temperature should always be at least 180 deg. fahr. when the valves are adjusted. Injectors must be kept clean and leaking seats or check valves repaired. The importance of regular cleaning and inspection of all fuel lines and fuel strainers cannot be overemphasized. Air leaks in the fuel system are responsible for more road failures than any other cause. Elimination of back pressure in the exhaust line is another big factor in improving performance and reducing maintenance cost. We use a manifold of our own design and a 4-in. exhaust pipe. Our mufflers are designed so that they have a larger capacity than the pipe. We use a short individual intake manifold of our own design in each of the three blocks. This design gives each cylinder an equal amount of air and prevents starving any one cylinder. The air cleaners, however, give considerable trouble because the pulsation loosens the filter and permits it to be pulled into the motor.

Extent of Operations

Our operations extend from San Francisco, Portland, and Seattle east to Minneapolis, and, as you will understand readily, cold weather is a real problem in that territory. In Nevada, Idaho, and eastern Oregon, eastern Washington, Montana and North Dakota, and Minnesota we give daily service at temperatures as low as 60 and 70 deg. fahr. below zero. 30 to 40 deg. fahr. below zero is commonplace in the areas we serve. At such temperatures 27 plus Diesel oil will not pass through a filter at all. Straight 27 plus will not even flow through a ⅜-in. copper tubing. The moisture content will gather in the slightest dip in the fuel lines, form ice and stop the flow of fuel entirely. Lying on your back under a truck thawing fuel lines with a rag dipped in fuel oil burning on the end of a piece of welding rod is a most unpleasant task. The most you can possibly say for it is that it is better than freezing to death.

To combat this condition extreme care must be taken in designing and maintaining the fuel tanks, lines, and filters. The tanks are designed so that the exhaust passes through a

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tube built lengthwise in the lower part of the tank. The tanks on equipment operating in the coldest territory are insulated heavily to retain the heat. Fuel lines are taken from the top of the tank rather than from the bottom. The line is never brought below ½-in. from the bottom of the tank. Each tank has two large sumps for collecting any moisture in the fuel. All fuel lines are of ½-in. tubing and are insulated and, even in some cases, heated by flexible exhaust lines. No dip, however slight, is permitted in a fuel line. The full length from fuel tank to filters drains back into the tank. Filters are mounted on the left rear side of the motor above and to the rear of the fuel pump. The fuel is diluted 10 per cent with kerosene, and all fuel storage tanks are pumped out weekly to remove any moisture which may have collected. Sleeves are replaced when the taper exceeds 0.010 in. Pistons are given 0.005 in. clearance when new or rebored sleeves are installed. As the oversize increases, the motor becomes rougher. Motors 0.040 in. over are so rough that we have never attempted to go over that. Pistons are discarded after 150,000 miles regardless of condition due to danger of breakage from fatigue. At that mileage pistons show no appreciable wear. We have tried aluminum-alloy pistons but found that oil consumption increased and the pistons could not be trusted over 45,000 miles. Rings are replaced whenever the pistons are removed for any reason. They are given 0.001 in. side clearance, 0.018 in. end clearance on the top ring, and 0.014 in. end clearance on the others.

The crankshaft is replaced when 0.007 in. or more flat. Building up worm shafts by the metal-spraying process is successful in most cases and we do it regularly. However, there is no denying the fact that, in a small percentage of cases, the metal comes loose with disastrous results. The camshaft is replaced when worn 0.004 in. or more. We are experimenting with a camshaft which has been built up with metal spray but have no experience with it as yet. There is a tremendous field for a reliable method of reclaiming work parts of this type.

Troubles with Connecting-Rods

Connecting-rod bearings present the most serious problem in the Cummins motor. Ring-sticking really has ceased to be a cause for concern. The motor could be run considerably farther between checks if the bearings could be trusted. The trouble lies in the failure of the bond between the cadmium silver and the back of the bearing, and is not confined to either the top or bottom of the bearing. We have, of course, experienced the usual bearing troubles from lugging and overspeeding, but they have been corrected by proper driver instructions and supervision. We set the connecting-rod bearings at 0.0035 in., the 3¼-in. main bearings at 0.004 in. and the 4½-in. main bearings at 0.006 in. The bearing failures do not appear to be the result of faulty lubrication. The trouble seems to be in the bearings themselves. No. 1 main bearing on the 4½-in. shaft also requires replacement at almost every check.

Connecting-rods always should be checked for cracks by the use of whiting and by tapping with a hammer while suspended by a string. The wrist-pin eye frequently is stretched 0.008 to 0.009 in., particularly if the motor has run away. We have found it sound practice to pull a motor down immediately if it has run away and check it thoroughly for cracked connecting-rods or pistons. We permit a maximum wrist-pin clearance of 0.004 in. New ones are given 0.0015.

For checking a motor with low oil pressure or high oil consumption, we use a pressure tank filled with S.A.E. 20 oil. With the pan removed this tank is connected to the main pressure line and inflated to 125 lb. per sq. in. Any excessive leaks will be located easily by this method.

Timing gears must have sufficient clearance to avoid cracking the block when hot. We lap them in with grinding compound, using an air drill on the crankshaft. The fan drive pulley requires frequent close checking as it is very hard to keep tight.

By the use of service units two men can overhaul a Cummins motor in 5½ hr. – without, of course, replacing the crankshaft, camshaft, or timing gears. The motor is very accessible and, if driven and serviced properly, can be operated at a maintenance cost no greater than that for a gasoline motor of the same size.

The 24-volt electrical system used with the Cummins has long been a source of trouble and expense. We have overcome this by the use of a 12-volt system. We now have 59 12-volt systems in service. The generator fields are rewound to make it 12-volt, and only two brushes are used. The starter fields are rewound to give what amounts to two 12-volt starters in one housing. The voltage regulator is replaced. The batteries are charged in parallel. Circuit breakers are used rather than fuses. With this system, the batteries are kept charged fully. Even if the batteries are run down while starting the motor, they will come up to normal in less than a hundred miles with all the lights on. The lights are much more satisfactory with the 12-volt system, and we have not broken a single starter spring to my knowledge on a 12-volt starter on a Cummins Diesel motor. The voltage regulator setting must be checked every 5000 miles and kept at exactly 14.5 volts.

Clutch and transmission problems are caused, for the most part, by the frequent shifting of gears in motion rather than by stopping and starting. On one check we found that the driver shifted gears 577 times in 57 miles although he came to a complete stop only once. This operation was done in 4 hr., 10 min. driving time. Particularly in the lower gears the slightest carelessness with the clutch throws a tremendous strain on the entire drive line which is multiplied by the trailer. We find that much smaller differentials, drive lines, and transmissions can be used successfully in a solo truck or a semi-trailer than can be used in a truck pulling a full trailer.

Grease on the clutch plates is a serious problem, so much so that we dare not lubricate the pilot bearing properly and, as a result, have occasional failures at that point. We have found that washing out with Pyrene is the only possible remedy for grease in the plates unless the entire assembly is removed. The clutch throw-out bearing must be lubricated frequently as it is subject to very severe usage. The clutch brake, also must be kept in the best possible condition. The splines on the clutch shaft present the most serious problem. They become worn where the plate hubs ride and cause a great deal of difficulty.

We use two transmissions in all our road equipment, giving us the usual twelve speeds forward and three reverse. In both transmissions the main source of trouble lies in the bearings. They are not heavy enough to withstand the hours of pulling in the lower gears. This trouble we have overcome largely by the use of roller bearings to replace ball bearings. The gears are replaced only when there is danger of actually breaking a tooth. We do not change gears when they become noisy, as no one can hear it but the driver and he is paid to listen to it.

Drive lines, if equipped with a tubing having about 50 per cent thicker side wall than that recommended by the manufacturer, give very little trouble. The needles wear grooves at about a 15-deg. angle in the cross, but we have been unable to discover why, and they do not seem to fail

because of it. The standard tubing, however, just will not stand the service. We have had drive lines twisted one full turn in extreme cases, and slightly twisted or bent shafts are the rule rather than the exception.

In four-wheel trucks we use double-reduction differentials; in six-wheel trucks, four-wheel-drive worms. The weakest point in the double reduction lies in the ball bearings used as side bearings. Replacement with roller bearings does a great deal toward correcting this trouble. The pinion teeth frequently are found with large sections broken out also. We find it essential to check all differentials at least once a month by draining the grease and inspecting the unit as well as possible through the inspection hole. Worm differentials give very little trouble as long as they are lubricated with castor oil. Various experiments have all shown that worm-drive differential temperatures run higher with heavier lubricants. In experiments we found that S.A.E. 250 ran at a much higher temperature than S.A.E. 160, and S.A.E. 160 ran much higher than castor oil which is about S.A.E. 90. Rivets occasionally come loose although they are riveted cold. The roller bearing on the worm shaft is the weakest point in the assembly and must be watched closely.

When Axle Shafts Break

Axle shafts quite frequently are broken and we carry a spare shaft on each truck for this reason. Most breakage is traceable to a poor clutch or a careless driver. However, we have a good many trucks with rear axles so designed that the axle sleeve can be removed without first removing the brake spider. It is almost impossible to keep the sleeve tight in the housing in these trucks and, as a result, they break many more axles and have more wheel-bearing trouble than the other equipment.

It is very hard, as you all know, to keep castor oil where it belongs. The task is almost impossible unless the sleeve and axle shaft are constructed to use an oil seal on the inner end of the sleeve. The grease retainer in the hub gives very little trouble. Hubs should all be of cast steel. Malleable hubs have caused us innumerable road failures. Particularly on front axles they are a dangerous source of accidents. On rear hubs the flange will shear off, and it is sometimes mistaken for a broken axle or a differential failure.

Rear-axle housings and axle sleeves are bent frequently by hitting chuck holes or running over curbs. This practice results in broken axle shafts also. We find that the axle will start to break shafts if the housing is bent over ¼ in. Temporary relief can be obtained by leaving the axle stud nuts loose, but it is not good practice to do so.

Wheel bearing cups often become loose in the hub. When this trouble occurs, we build up the hub and bore it again to the proper size. Wheel bearings are checked with the wheels jacked off the ground as a part of the 2000-mile service.

Brakes present many problems in the maintenance of our fleet. We carry 135 to 150 lb. per sq. in. of air and put 90 to 110 lb. per sq. in. to the diaphragm. This pressure gives us really adequate brakes. At the pressures recommended by the manufacturers we were unable to secure satisfactory performance.

Slow application and release have long been a problem to us but they have been overcome largely by the use of the largest tubing and the elimination of many elbows and curves in the lines.

The Cummins motor is designed so that the compressor is engine-lubricated. A real advance will have been made when an interchangeable self-lubricated compressor is designed for this application. It will eliminate a great deal of trouble with carbon and with oil in the air valves and lines due to the rapid cylinder wear in the compressor from improper lubrication.

For a long time brake drums checked so badly that they acted almost like a file on the lining and even broke and fell completely off on the road due to the extreme heat generated on long hills. We now use nickel-cast-iron drums which have been left in the weather for six months before the final machining, and they have eliminated our trouble. Brake lining is giving us an average of over 75,000 miles as compared to 35,000 miles only a few years ago. We do not reline all the wheels at the same time; we line only the wheel or wheels actually requiring relining. Right-hand wheels use more lining than left-hand wheels.

We use a 16½- x 6-in. brake with ⅜-in. lining as our standard and find that it gives us much better mileage and performance than the 17¼- x 5½-in. brake using ¾-in. lining. On the 17¼-in. brake the drum is so thin that it will expand away from the shoes when the brakes are hot descending a long grade. For this reason we use thicker brake drums and ⅜-in. lining on the 5½-in. shoes. We use service unit shoes so that relining requires only about 45 min. per wheel. The lined shoes are not ground to fit the drums but give us very little trouble as the shoe bushings, anchor pins, cam bushings, cams and shoe plates are checked closely at each reline and replaced if found worn. Brakes are vital to our service as we must descend hills as rapidly as possible to maintain our schedule. Fully loaded our trucks and trailers descend most long winding grades at 20 to 25 m.p.h.

The diaphragm on the axle should be of pressed steel rather than of aluminum. Flying rocks frequently break the aluminum pressure plates and a severe application will not infrequently break the base off the aluminum non-pressure plates.

The diaphragm should be mounted so that the push rod is tilted downward even if only at a slight angle. If it is not, the diaphragm will fill with water, mud, snow and, in cold weather, with a block of ice which renders your brake utterly useless. Anyone who has ever found himself with no brakes of any kind because of this trouble will agree that it is a mighty important thing and that the non-pressure plate should be drilled in two or three places near the bottom for drainage as well. Linkage between the diaphragm and the slack adjuster is very undesirable, and we eliminate it entirely wherever we can.

Air lines must be installed so that there are no dips whatever in them and all the lines drain back into the tanks. Moisture will collect and freeze in any dip or pocket in the lines. When it does so the brakes will not function at all; an accident is almost certain to result. The lines are inspected for leaks as a part of the 2000-mile service. Dirt in the lines, particularly on the trailer, is a source of trouble in the valves. We are experimenting with traps of various kinds, the most successful being 4-in. in diameter, 9-in. long and filled with hair. The air tanks must be drained at least every 200 miles, particularly in cold weather, and the drain cocks for this purpose must be protected from flying rocks.

Front axles present many problems. One cause of trouble is the fact that the load carried changes the caster. We give ½ in. more caster on the right than on the left to make it easier to hold on a high-crowned road. The trailer will accentuate the slightest defect in the steering. We operate over many hundreds of miles of high-crowned road with uneven surfacing near the shoulders and most of it is so narrow that we have as little as 2 in. to spare when meeting another truck. Under such conditions the steering requires constant care and must be kept as nearly perfect as possible. Our ex-

perience has been that a truck using a front axle with a vertical kingpin gives much less trouble than one with an inclined kingpin. We do not use front brakes as they have proved unsafe on icy or slippery roads. The truck with the inclined pin is harder to steer and has a very definite tendency to wander or dive on uneven roads. Front springs and the angle of the drag-link must be correct, of course. The steering gear must be centered exactly. Frequently one spline on the Pitman arm makes a tremendous difference. The tie-rod and drag-link ends must be kept in the best possible condition and the toe checked frequently. The steering gear itself requires very little service if lubricated properly. In extreme weather it is sometimes necessary to use motor oil as a lubricant in the gear to make steering possible.

We use adjustable curtains on the radiators of all road equipment. Shutters have proved very unsatisfactory. The curtain with an ordinary window-shade roller has proved more effective and simpler to maintain and operate. It is controlled from the cab by a chain, and the driver uses the curtain to keep the water temperature between 160 and 180 deg. fahr. at all times. Whenever a radiator is removed for repairs we have it boiled out and rods run through each tube. Radiators are thoroughly flushed out about twice a year. The problem of overheating is not serious because Diesel motors run very cool when performing properly. It is usually difficult, particularly in cold weather, to keep the motor warm enough. Radiator cleaners are not very effective for our work because the only real trouble we have is caused by pieces of rubber from the inside of the hoses. They lodge in the tubes, plugging them. Of course, all radiator cleaners are made harmless to rubber so do not correct our difficulty.

Radiators and outlet connections are designed so that the hose extends across from left to right instead of using a short hose parallel to the frame. Before making this change, the hose would pull off and let out all the water whenever the frame was twisted by a deep chuck hole or some similar cause.

We do not use any anti-freeze in our road equipment. We have found the cost to be prohibitive. In cold weather the water is rarely shut off at all. When it is, it must be drained immediately, and the motor run a few minutes to dry out the water pump. In order to keep the truck warm enough in extremely cold weather, the bottom of the motor is covered completely back to the bell housing. Over half of the area of the radiator core is covered with sheet metal also.

When a truck has been standing for some time in sub-zero weather, all the units must be warmed before starting as the grease will channel and bearing failures will result if this warming is not done. It is common practice in case of road failures to build fires under the motor, transmission, and differentials with Diesel-fuel-soaked rags before starting out again. It is hard on paint but it eliminates the danger of another road failure within a few miles.

Tire service is another big problem, of course, and time is too short to discuss it fully here. We recap our tires and repair damaged casings in our own plant and find it very successful. We re-groove tires in wet or cold weather but do not do so in summer when the roads are clear and dry. Our average mileage per tire, including mileage after recapping, is over 67,000 miles. We use only new tires on front wheels. As soon as they are worn 25 per cent or 30 per cent, they are removed and used on the inside drive wheels with recaps, which have a larger diameter on the outside to take care of the crown of the road. On four-wheel drive six-wheel trucks the tires on the drive wheels must be measured carefully and matched to avoid differential wear. Old or repaired tires are used on trailers. When it is necessary to overload, the air pressure is increased 10 per cent if the load is increased 10 per cent. Outside tires are inflated to about 10 per cent higher pressure than inside tires.

During the winter months we carry tire chains on all our road equipment. They are carried on three hooks under each side of the body. A chain with a small hook attached is installed on the cab floorboards beneath the brake pedal.

Then, when the driver loses traction on a hill, he ties down his brake pedal, leaves the cab and has only to go to the side of the truck for his chains. We find that a truck will stand with the brakes on where it would roll backwards if left in gear with only the hand brake set. We make and repair our own tire chains, using three side chains and cross chains short enough to hold the center chain away from the tires. Our chains are much lighter than standard chains for the same size tire and are, therefore, much easier to handle and install. Also they cause less vibration than the larger chains.

We design and build our own bodies and they require very little maintenance. They are all insulated so that they can be refrigerated in summer and heated in winter. The tops are of 16-oz. duck and are coated once a year. Occasionally repairs to the rear doors and the rear corner posts are about all the maintenance required aside from repairing damage due to accidents or careless handling.

Synthetic Enamels for Painting

We use fast-drying synthetic enamels for painting our equipment. Time is so limited that many times we are forced to use equipment which is only partially painted or is not completely dry. However, our results are very satisfactory, considering the conditions. We mix 20 per cent clear with the synthetic to give a lasting luster.

The trailers that we use are of our own design and manufacture. They give very little trouble and require very little attention, but greasing, checking the wiring and air brake lines and valves, and occasional brake relines. The rear spring eyes on our six-wheel trailers were a source of considerable trouble but it has been overcome by the use of "safety eyes."

Safety is a watchword in the design and maintenance of our equipment. Every possible precaution is taken to eliminate the possibility of accidents. We use front springs double-wrapped on both ends. All windshields are of safety plate to avoid possible distortion of the driver's vision. Lights are tested regularly with a light output meter. Brakes are tested with a machine which shows feet required to stop from 20 m.p.h. We carry considerably more equipment and use more clearance lights than are required by the I.C.C. safety regulations. Everything possible is done to make our equipment safe.

Our 2000-mile inspection and service is the most important of the service operations. It must be thorough, and we do not permit any small repairs to be left undone. If a cab is not kept tight and properly weather-stripped or the heater is not kept in good condition, it may result in paying compensation to a sick driver. If the hood lacing is not kept in good shape, the hood itself may require replacement. If the head and spotlights are not in the best possible condition, the driver cannot make his schedule. If the fuel lines have the slightest leak, a road failure may result. If bearings are not kept adjusted properly, they may fail and cause a great deal of damage.

All of these facts have been considered in forming our maintenance policies. It is our policy to keep all our road equipment in the best possible condition at all times and to take preventive action before a failure occurs rather than to wait until the failure makes repairs mandatory. We believe that this policy is cheaper over a period of years and that its additional benefits in improved service and dependability make it the only possible course to pursue.

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The Diesel as a High-Output Engine for Aircraft

By E. G. Whitney and H. H. Foster
National Advisory Committee for Aeronautics

FROM single-cylinder engine tests, power and fuel-consumption curves have been calculated for a highly boosted multicylinder Diesel engine. The influence of inlet-air temperature and maximum cylinder pressure on attainable mean effective pressure for a given specific fuel consumption is demonstrated. Theoretical Diesel-cycle performance curves are included to show gains possible by further improvement and control of combustion rates. Performance to be expected from the two-stroke-cycle Diesel is estimated.

It is concluded that the boosted four-stroke-cycle Diesel engine can exceed the maximum sea-level output of the 100-octane gasoline engine and give lower specific fuel consumptions at any power output. A similar comparison holds for the altitude performances of the two types, although the Diesel will impose greater demands on the air blower. Adoption of the turboblower by the Diesel engine offers further advantages. Greatly increased outputs for a given specific fuel consumption result as inlet-air temperatures are reduced, indicating the desirability of more efficient blowers or improved aftercoolers. Further substantial gains in performance are possible if materials, bearings, and design technique are improved to allow of higher cylinder pressures. The two-stroke-cycle engine offers a decrease in engine weight per horsepower at the expense of somewhat increased fuel consumption.

THE Diesel engine has achieved little success as an aircraft engine in this country or any leading nation in aviation with the single and notable exception of the Junkers engine development in Germany. That the Junkers success has been so outstanding and its development and progress so unswerving in the face of the recent rapid strides made by the gasoline engine with improved cooling and better fuels, cannot fail to attract the attention of even the most skeptical concerning the merits of the Diesel engine for aviation. Especially is this true when well-qualified authorities predict for the 100-octane gasoline engine in the next five years specific fuel consumptions no better than 0.38 lb. per b.hp-hr. and specific engine weights in a 2000-hp. unit no less than 1 lb. per hp., whereas, the Junkers Jumo engines are operating today at specific fuel consumptions as low as 0.36; and a 2000-hp. engine development is in progress to weigh less than 1 lb. per hp.

The object of this paper is to analyze and present results obtained at the N.A.C.A. laboratory in order to draw a clearer picture of the present state of development of the Diesel engine and to point out gains to be expected from its further development and, hence, its relative suitability as a powerplant for all types of aircraft.[1]

Four-Stroke-Cycle Diesel

The Diesel engine investigations at the Committee's laboratory have been conducted on single-cylinder engines and have included both the two- and four-stroke-cycle types. Considering the four-stroke-cycle first, tests have been made on 5- by 6-in. and 5- by 7-in. cylinders over a speed range up to 2500 r.p.m. The variables investigated have included: combustion-chamber types with and without forced air flow; fuel-injection systems including various pumps, valves, and nozzles; ground boosting; performance at altitude; factors affecting engine friction; effects of inlet-air temperatures and pressures; and the effect of exhaust back pressure.

The importance of the combustion-chamber form has been shown, and the superior performance characteristics obtainable from the displacer-piston type (Fig. 1) have been demonstrated. Ten years of work have resulted in an increase in indicated mean effective pressure from 100 lb. per sq. in. at 1500 r.p.m. to 260 lb. per sq. in. at 2500 r.p.m. and have enabled every-day Diesel engine operations at outputs equal to or in excess of those obtained from our best gasoline aircraft engines. From a study of the engine performance that has been obtained and the apparent likelihood of continued increases in specific power output, the authors are convinced that the Diesel engine can now be, and should continue to be, a serious competitor of the gasoline engine for aircraft.

Single-Cylinder Performance

In starting the analysis, consider the performance obtained from the 5- by 6-in. test engine at the Committee's laboratory. A complete ground-boosted calibration of the single-cylinder engine is given in Figs. 2 to 5, inclusive, the figures representing constant engine speeds of 1600, 1800, 2250, and 2500 r.p.m., respectively. Fuel quantity injected per cycle is used

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1938.]

[1] A chronologically arranged bibliography listing all publications of the National Advisory Committee for Aeronautics relative to its work on the Diesel engine, may be had upon request to S.A.E. Headquarters.

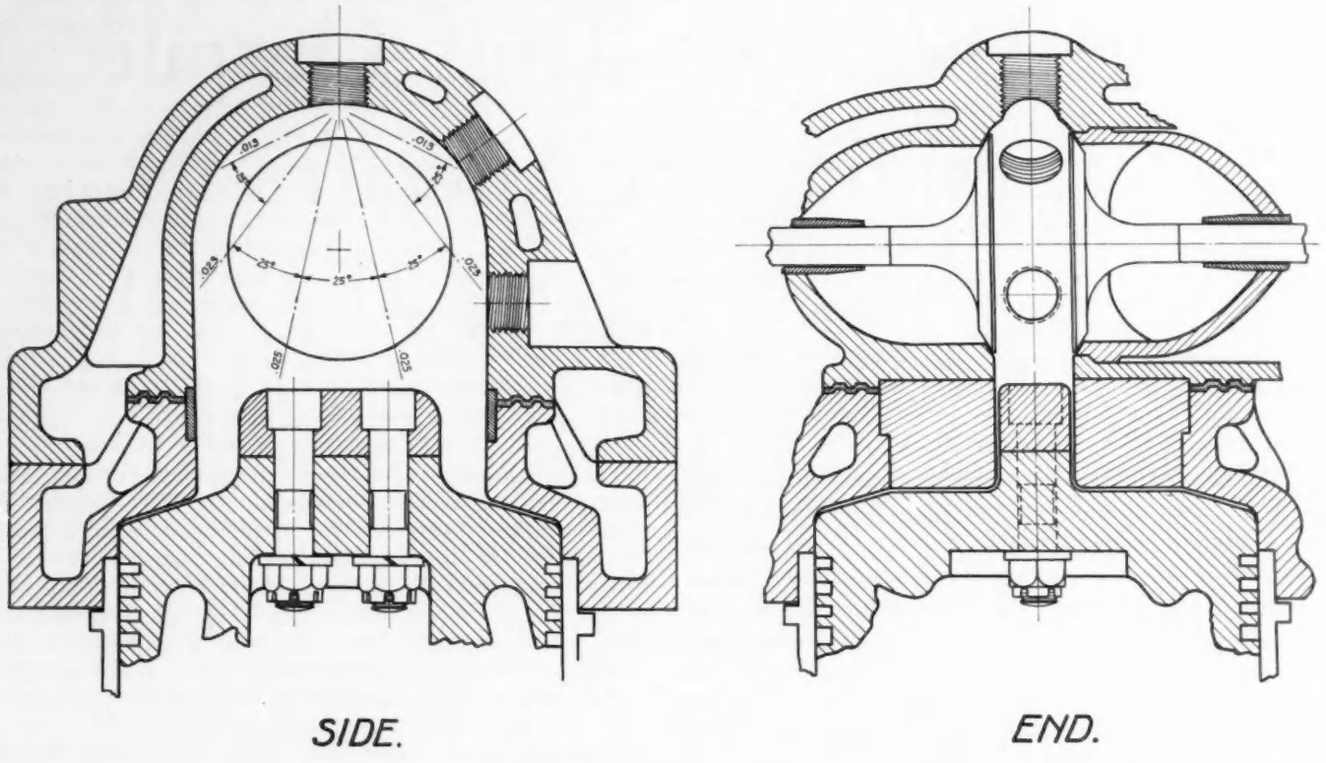


Fig. 1 – Combustion-Chamber of Four-Stroke-Cycle Engine – Compression-Ratio, 14:1

as the abscissas and lines are drawn to indicate the beginning of exhaust haze and the air-fuel ratios. Indicated-performance values shown were calculated from measured power plus motoring friction. The blower was driven separately in all tests. Inlet-air temperature was held constant at 66 deg. fahr. Maximum cylinder pressures, about which more will be said later, were adjusted to the values shown by controlling the timing of injection.

Multicylinder Performance

From the curves of Figs. 2 to 5, an estimate of multicylinder Diesel-engine performance has been derived. Inasmuch as a high boost is desirable for high take-off performance, a radial engine is assumed with a gear-driven single-stage centrifugal blower proportioned to give the inlet-air pressures and temperatures of the boosted, fuel-injection Wasp R-1340-33 engine. The assumed engine will then have the intake-air pressures and blower-temperature-rise values for the crankshaft speed range of 1600 to 2500 r.p.m. as shown in Fig. 6. The higher compression ratio and forced air flow for combustion of the Diesel engine will result in a somewhat higher f.m.e.p. (friction mean effective pressure) curve than that of the gasoline engine, although considerably less than the single-cylinder measured friction values. From certain reported friction studies of the Diesel engine and from as yet unreported breakdown tests of the friction of a radial gasoline engine, a multicylinder Diesel f.m.e.p. curve is calculated for the assumed engine and is plotted in Fig. 6. This curve represents reasonable extrapolation of trends determined in tests and is considered a fair estimate.

Interpreting the test-engine performance in terms of the inlet-air temperatures and pressures and assumed friction curve of Fig. 6, the multicylinder-engine performance curves of Figs. 7, 8, and 9 are obtained. Fig. 7 shows the maximum values of b.m.e.p. with corresponding maximum cylinder pressures and brake fuel consumptions for each speed and the minimum brake fuel consumption with the corresponding b.m.e.p. obtainable. For the maximum-output throttle setting, the exhaust, of course, will be smoky. No difficulty should result from such operation other than the unsightly smoke trail, which could well be tolerated for take-off and emergency conditions.

In Fig. 8, part-load fuel consumptions resulting at any particular speed are clearly shown for any percentage of power down to 35 per cent of maximum. Such curves enter into the determination of economical cruising speeds for long-range aircraft, the final solution involving the performance against engine-power characteristics of the airplane as well as the most economical operating condition of the engine considering wear and endurance. The curves are based on a b.m.e.p. of 200 lb. per sq. in. at 2500 r.p.m. for 100 per cent power, or a specific output of 0.625 b.hp. per cu. in. displacement. The influence of higher friction losses and blower power required at the higher speeds is reflected as increased specific fuel consumption. It is seen, however, that, in the present state of the Diesel engine development, for the assumed engine 75 per cent maximum power or a specific output of 0.47 hp. per cu. in. can be obtained without exceeding a fuel consumption of 0.39 lb. per b.hp-hr.

The data are translated into a somewhat different form in Fig. 9, which shows the specific fuel consumption that would be obtained at various values of b.m.e.p. for a speed of 2000 r.p.m. at 10 lb. per sq. in. boost pressure. The solid curve depicts the performance with inlet-air temperature at 247 deg. fahr. with the maximum cylinder pressure limited to 1100 lb. per sq. in. The dashed curve is included to show the comparative performance with the same maximum cylinder pressure but with 87 deg. fahr. inlet-air temperature, that is, the hypothetical case of no temperature rise through the blower

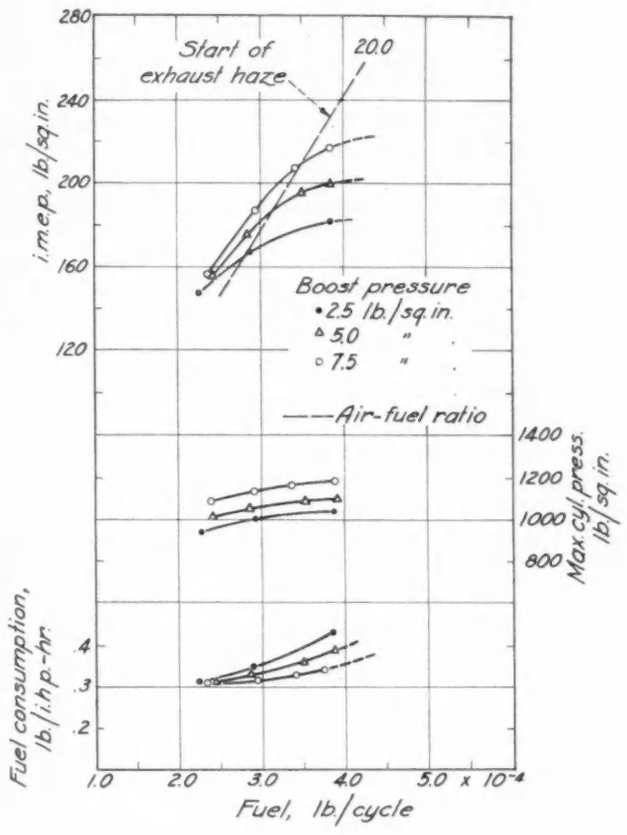


Fig. 2 – Boosted Engine Performance – 1600 R.P.M.

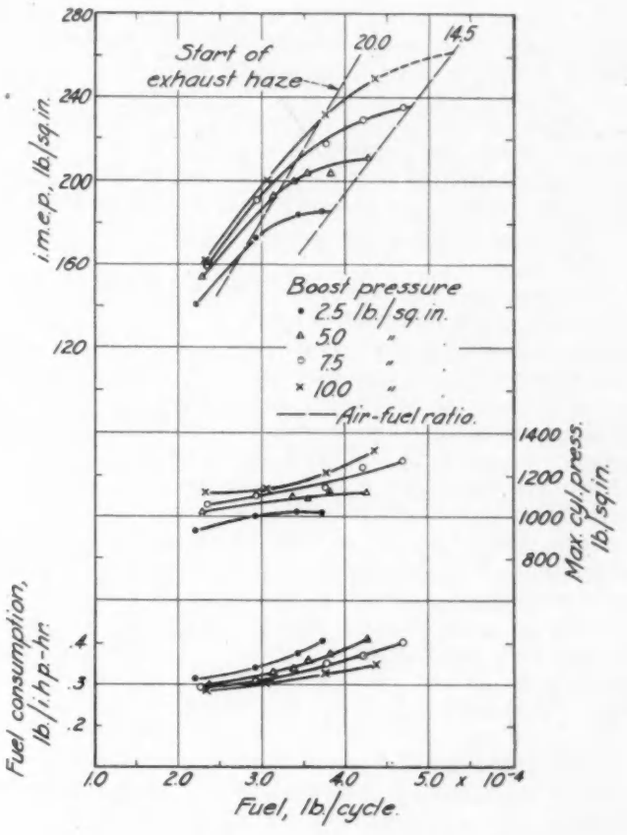


Fig. 3 – Boosted Engine Performance – 1800 R.P.M.

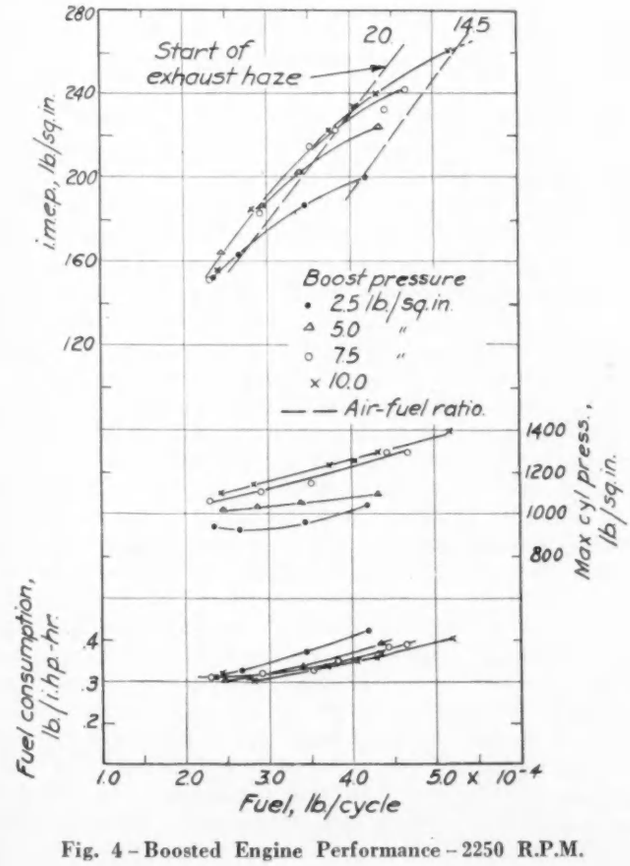


Fig. 4 – Boosted Engine Performance – 2250 R.P.M.

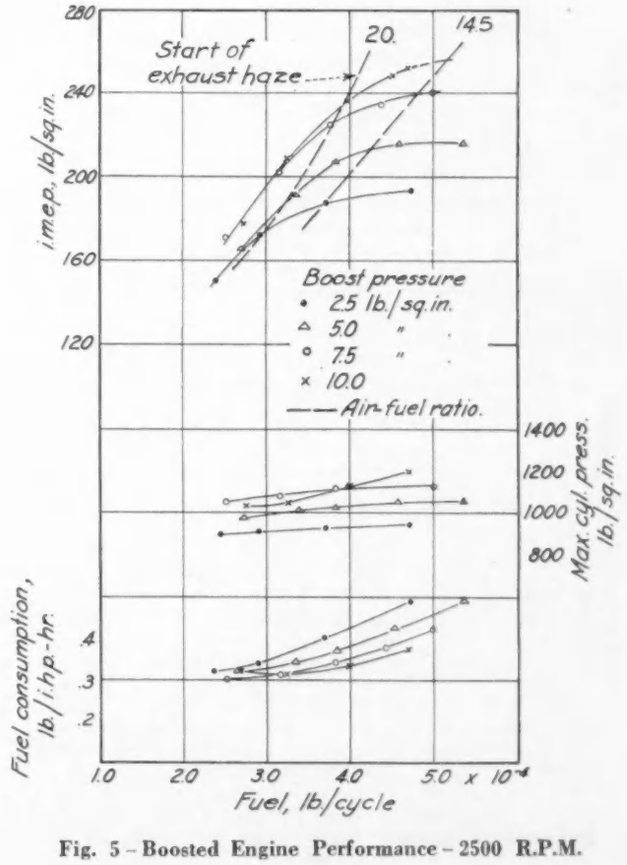


Fig. 5 – Boosted Engine Performance – 2500 R.P.M.

or with the use of a 100 per cent efficient aftercooler. Although knocking is not introduced by high inlet-air temperature in the Diesel engine as in the gasoline engine, the importance of low inlet-air temperatures to give high brake outputs at low fuel consumptions for the Diesel engine is indicated clearly. At a specific fuel consumption of 0.4 lb. per hp-hr., decreasing the inlet-air temperature from 247 to 87 deg. fahr. results in an increase in b.m.e.p. from 163 to 198 lb. per sq. in.

Thus far, little has been said of maximum cylinder pressures, the values shown being taken as not far in excess of those encountered under take-off conditions with some high-performance gasoline engines. With the high boost pressures assumed for the Diesel engine, considerable efficiency is sacrificed by retarded injection in order to avoid higher maximum cylinder pressures. A test point shown in Fig. 9 gives an idea of the improvement in performance that could be expected from an increase in maximum cylinder pressure from 1100 to 1600 lb. per sq. in. allowing the 247 deg. fahr. air temperature. The probable curve at lower loads and the high cylinder pressure is dotted in to demonstrate the substantial increase in mean effective pressure obtainable at a given specific fuel consumption by raising the limiting cylinder pressure 500 lb. per sq. in. Carrying the analysis one step further and assuming a 100 per cent efficient aftercooler in conjunction with the high maximum cylinder pressures, an additional test point is plotted and an estimated part-load curve is shown dotted. This performance is superior to that obtained with present gasoline aircraft engines.

100-Octane Engine Comparison

To draw a fair comparison between the Diesel engine and the 100-octane gasoline engine performance is somewhat difficult for two reasons: In the first place, a combination of compression ratio and boost pressure within the knocking limit must be chosen for the gasoline engine to favor either the maximum take-off power or minimum fuel consumption, since both are not optimum in the same engine. In the second place, conditions of operation of the gasoline engine in the airplane require the engine manufacturer to shade somewhat the best obtainable performance in his advertised charts to take care of poorest operational conditions. The very careful manipulation and control of engine conditions possible in some airplane installations, however, would permit substantial reduction in fuel consumption to be realized. These items must be borne in mind when considering the advertised performance curves of 100-octane gasoline engines. The take-off, the continuous emergency, and the climb consumptions obviously indicate fuel cooling, and show tremendous fuel savings for the Diesel under these conditions since the latter is required to take no precautions against knocking.

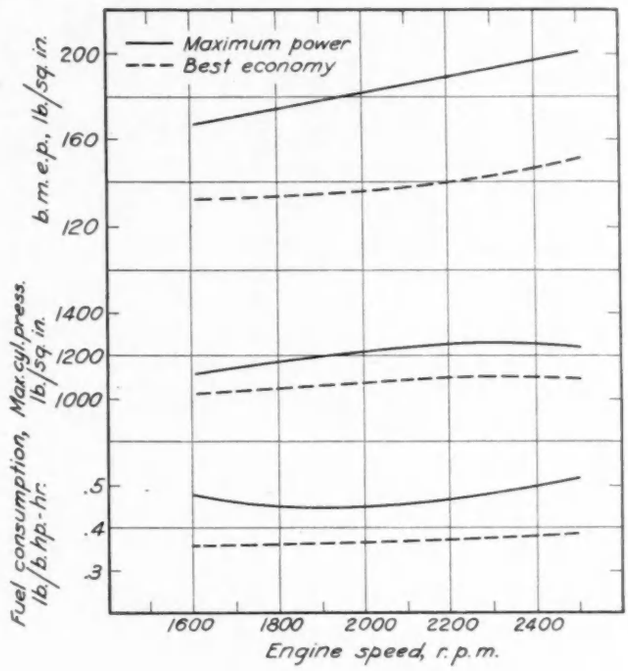


Fig. 7 – Performance at Maximum Power and Best Economy for Various Speeds – Four-Stroke-Cycle Engine

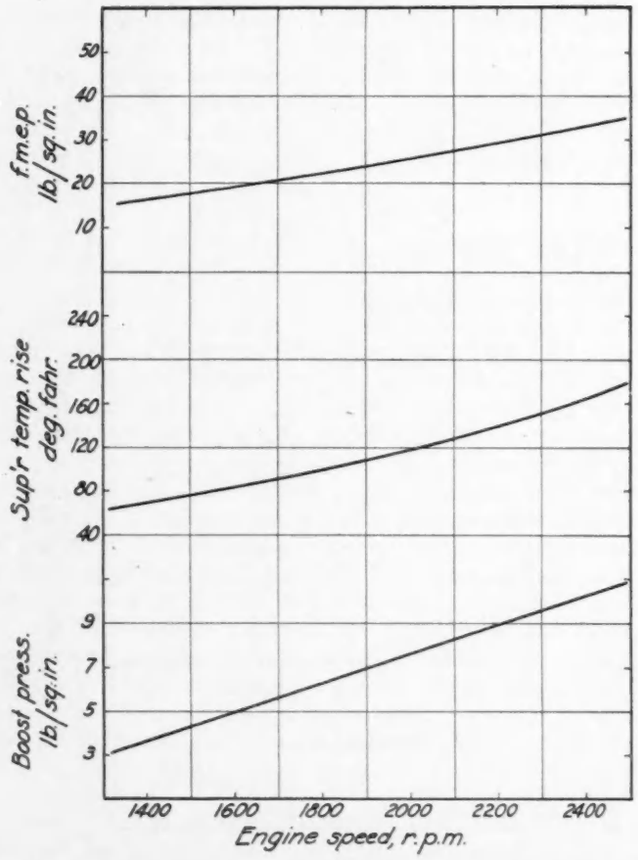


Fig. 6 – Supercharger and Friction Curves for Four-Stroke-Cycle Engine – Supercharger Inlet Air, 65 Deg. Fahr.

The fuel consumptions of the gasoline engine at lower mean effective pressures possibly may be reduced somewhat under favorable operational conditions, although the Diesel engine values still will show substantially higher mean effective pressures obtainable at equal consumptions. It must be remembered that minimum consumptions of 100-octane gasoline engines are not to be expected below 0.38 lb. per hp-hr. The take-off b.m.e.p. for such an engine will be limited by fuel knocking to about 185 lb. per sq. in. Also, at this low specific consumption with present gasoline engines, knocking is encountered at cruising conditions. It seems safe to conclude that the Diesel engine offers take-off powers equal to, or in excess of, the best 100-octane engine values, and substantially higher cruising powers for equal specific fuel consumption with resultant increased airplane cruising speeds or increased cruising ranges. These advantages are obtainable at the expense of a somewhat higher maximum cylinder pressure at take-off and a considerably increased maximum cylinder pressure under cruising conditions. The influence of these items on engine design, materials, and engine life remains to be determined.

Altitude Performance

The discussion thus far has considered only the sea-level performance of the ground-boosted Diesel engine employing a single-stage blower. Few aircraft will be flown at sea level. It is, therefore, unfortunate that complete comparable information is not at hand to calibrate the hypothetical Diesel engine at any altitude. From reported tests under simulated altitude conditions, however, an estimate of the Diesel engine performance at altitude can be made.

The Committee's researches have shown that, for equal inlet-air temperatures, the Diesel and the gasoline engines lose power in about equal proportions as operating altitudes increase. Thus, if the two engines operate at full throttle at sea level and the gasoline engine is not required to add carburetor heat at the lower air temperatures encountered at altitudes, the change in Diesel engine performance may be estimated as similar to that found in the altitude calibration curves of the gasoline engine. The gasoline engine will, in most cases, be designed to operate at full throttle at a certain altitude and will approach that altitude from take-off at some gradually diminishing condition of partial throttling. In other words, exclusive of a greatly increased power for a restricted time at take-off, no loss in sea-level power will be encountered up to the designed altitude; in fact, available power will increase slightly as exhaust back pressure is reduced. Contrarily, the Diesel engine will operate at full throttle during take-off and climb with diminishing power as the altitude increases. At first glance, then, the gasoline engine appears to hold an altitude advantage; however, closer consideration reveals the opposite to be true. Partial throttling at sea level is required of the gasoline engine to avoid high manifold pressure and resultant knocking. As the inlet-air pressure to the supercharger decreases with altitude, the throttle may be opened wider to maintain the allowable absolute manifold pressure. For any partly throttled condition, the engine is losing power by compressing air from a reduced initial pressure in order to avoid generating blower pressures that cannot be used in the engine, resulting in sacrificing fuel economy at the ground to maintain engine power at altitudes. The Diesel engine makes no such sacrifice at the ground and is able to operate at full throttle regardless of altitude. Hence, when designed for the same critical altitude, the Diesel engine would offer substantially increased power for take-off, rather than be held to equal or to slightly less power below rated altitude by throttling.

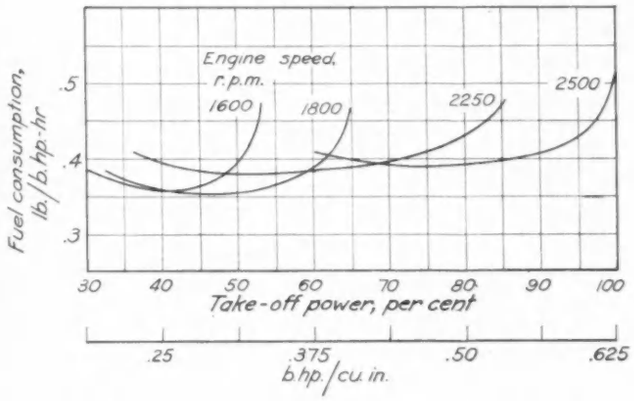


Fig. 8 – Fuel Consumption at Various Power Outputs

The point not to be overlooked, however, is that the present Diesel engine, because of its inability to burn all of its inducted air, will require a higher degree of boost than the gasoline engine for equal mean effective pressures. Consequently, under some conditions, an additional stage blower may be required by the Diesel engine to equal the altitude performance of a gasoline engine. The more rational solution lies in the use of the exhaust turboblower in conjunction with the geared blower. Better economies at altitude will thereby be possible, and the prevalent heat-stress difficulty of the turboblower's application to the gasoline engine will be much reduced by the lower exhaust temperatures of the Diesel engine. For very high altitude airplanes, such as are commanding considerable attention at present, the turboblower will be a requirement, and the lower exhaust temperatures of the Diesel engine may permit the attainment of a high-altitude Diesel engine more readily than a high-altitude gasoline engine. In any event, the added engine weight (¼ lb. per hp., approximately) involved by the use of the turboblower results in lower weight per horsepower at altitudes.

Limits of Diesel-Engine Performance

The values thus far shown for the Diesel engine performance are based on those obtained at the N.A.C.A. laboratory and fall considerably short of performance obtainable through further development. This statement is evidenced by the necessity of higher boosts with the Diesel engine to equal the output of the gasoline engine, albeit the gasoline engine operates at a lower compression ratio and, consequently, should have a lower thermal efficiency. A brief consideration of possibilities of further improvement is offered. Fig. 10 shows the i.m.e.p.'s and specific fuel consumptions that are obtainable theoretically with the Diesel or constant-pressure cycle at 10 lb. per sq. in. boost pressure under the following conditions: compression ratio, 16.5:1; maximum cylinder pressure, 1200 lb. per sq. in.; and compression ratio, 20:1; maximum cylinder pressure, 1560 lb. per sq. in. Previously in this paper the term "Diesel" has been used to signify "compression ignition"; actually, the Diesel or constant-pressure combustion cycle has yet to be attained in high-speed

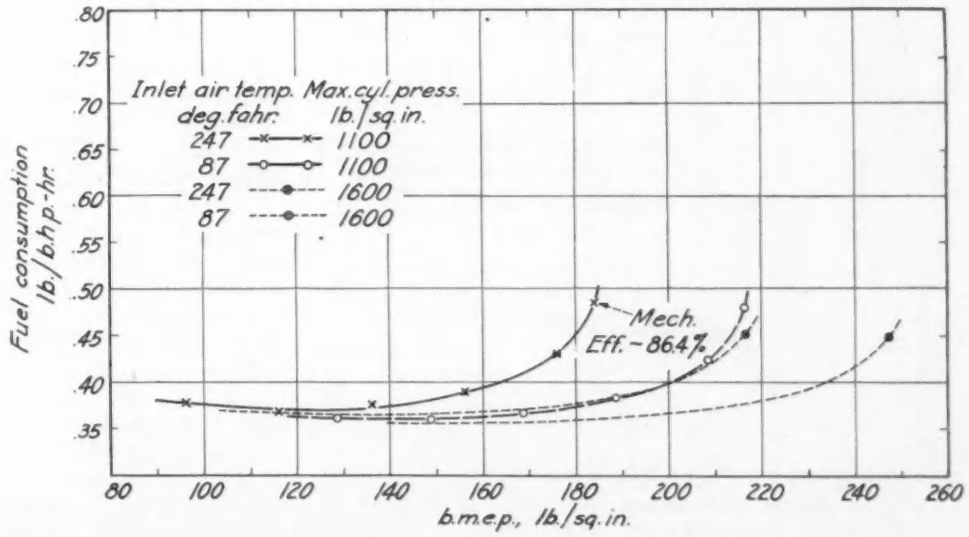


Fig. 9 – Performance at 10 Lb. Per Sq. In. Boost – 5- x 7-In. Engine; 2000 R.P.M.

engines. These curves, however, show the theoretical performance possible of attainment (100 per cent volumetric efficiency; inlet-air temperature 95 deg. fahr.; no heat losses) for the relatively low maximum cylinder pressures indicated, if combustion rates can be controlled to maintain cylinder pressures equal to the maximum compression pressure throughout the burning period. A comparison with the performance already obtained and shown on previous figures shows the possibilities for further development.

Attention is called to the well-known fact that the performance of Fig. 10 could be increased substantially by allowing cylinder pressures to rise, approaching the constant-volume cycle. If efficiency were the only consideration to the exclusion of engine weight, temperatures, and bearing loads, the effort certainly should be directed toward ever-higher cylinder pressure allowed by the Diesel engine in the absence of knocking. Should materials and design technique permit the use of cylinder pressures double those at present obtained, the gasoline engine with its knocking limitation could not hope to compete with the Diesel type in either power or economy. The indication of better efficiency from higher pressures should encourage the development of materials and design ingenuity.

The question is raised immediately as to the reason for the wide difference between attained and attainable performance from the Diesel engine. The answer lies directly in the failure of the present Diesel to complete combustion sufficiently early in the cycle to take full advantage of the expansion stroke. The dotted curve of Fig. 11 shows the theoretical rate of burning required to maintain the constant-pressure combustion, whereas the full curve illustrates actual burning obtained in tests. It may be seen that only about 30 per cent of the fuel burns at the desired rate; the protracted burning of the remainder throughout the cycle results in lowered efficiency. Progress in correcting this condition has been made, and further improvements are subject only to the limits of man's ingenuity.

Two-Stroke-Cycle Diesel

Inasmuch as the Diesel engine inducts only air rather than a fuel-air mixture, the Diesel offers a unique opportunity for the development of the two-stroke-cycle engine. The intriguing possibilities of the development of much greater outputs from a given displacement invite a comparison of the relative potentialities of the two- and four-stroke-cycles for the high-output Diesel engine. The authors are acquainted sufficiently with the complications and possible combinations of variables of the two-stroke cycle to avoid flat conclusions as little more than flights of fancy. Much work has been done at the Committee's laboratory with a uniflow two-stroke-cycle single-cylinder engine with piston-controlled inlet ports and poppet exhaust valves with the result that certain trends are becoming more definite and reasonable extrapolations can be made leading to conclusions meriting some degree of confidence.

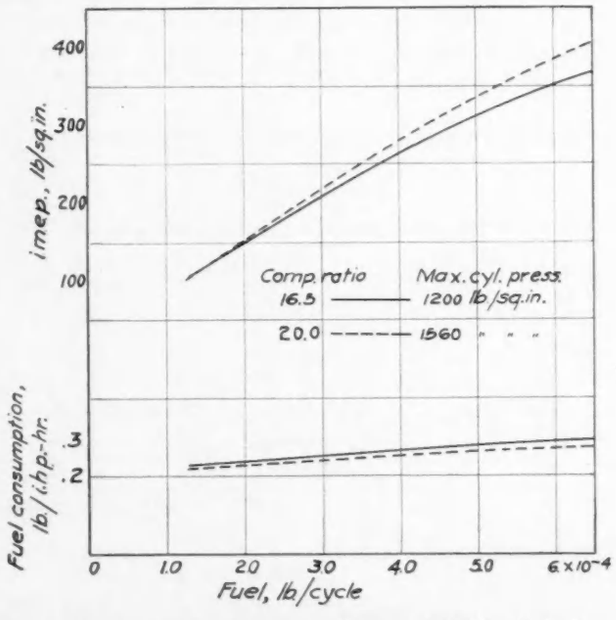


Fig. 10 – Engine Performance – Theoretical Diesel Cycle – Inlet Air Temperature, 95 Deg. Fahr.; Volumetric Efficiency, 100 Per Cent.; Boost Pressure, 10 Lb. Per Sq. In. (No Heat Losses)

In the first place, it is well to consider performance already obtained with the 4⅜- by 7-in. two-stroke-cycle Diesel engine. The lower mean effective pressure curve of Fig. 12 shows the indicated performance from single-cylinder tests at 1800 r.p.m. with 10 lb. per sq. in. scavenging air pressure

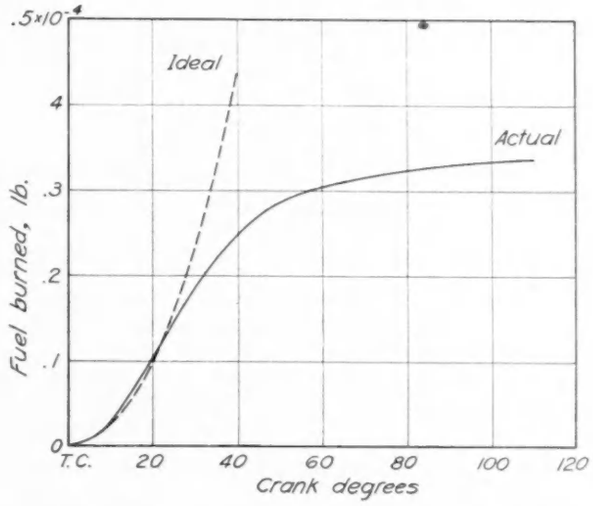


Fig. 11 – Fuel Burned in Actual and Ideal Diesel Cycles – 2000 R.P.M.; Boost Pressure, 4 Lb. Per Sq. In.

supplied to the engine at 115 deg. fahr. and maximum cylinder pressure limited to 1000 lb. per sq. in. From a series of tests, it is indicated that equal breathing, and hence equal i.m.e.p.'s, can be obtained at 2350 r.p.m., corresponding to a piston speed of 2750 ft. per min. It is difficult to predict the limiting piston speed of the four-stroke-cycle radial engine, but a comparison of cycles at 2750 ft. per min. as a limiting piston speed probably will involve little error. Valve-gear accelerations of the two-stroke-cycle are important in setting speed limits, but the Committee's tests have shown speeds of 2400 r.p.m. to be possible, which correspond to a piston speed of 2800 ft. per min. The curve of boost pressure against engine speed used in the four-stroke-cycle Diesel engine discussion can, therefore, be assumed for the two-stroke-cycle engine. Leaving Fig. 12 temporarily, Fig. 13 shows a comparison of the air consumption and f.m.e.p. estimates for the two- and four-stroke engines over the chosen speed range. (It seems likely that the assumed two-stroke boost curve and air-consumption curve will be only approximate and will adjust themselves to a probable net result of straight lines for both. However, for this analysis, the boost curve will be retained as for the four-stroke-cycle engine.)

Retaining the foregoing assumption that the present 1800 r.p.m. breathing can be obtained at 2350 r.p.m., the i.m.e.p. and specific output in hp. per cu. in. are obtained as shown in Fig. 14. There is reason to expect certain improvements in scavenging technique to result in a further performance

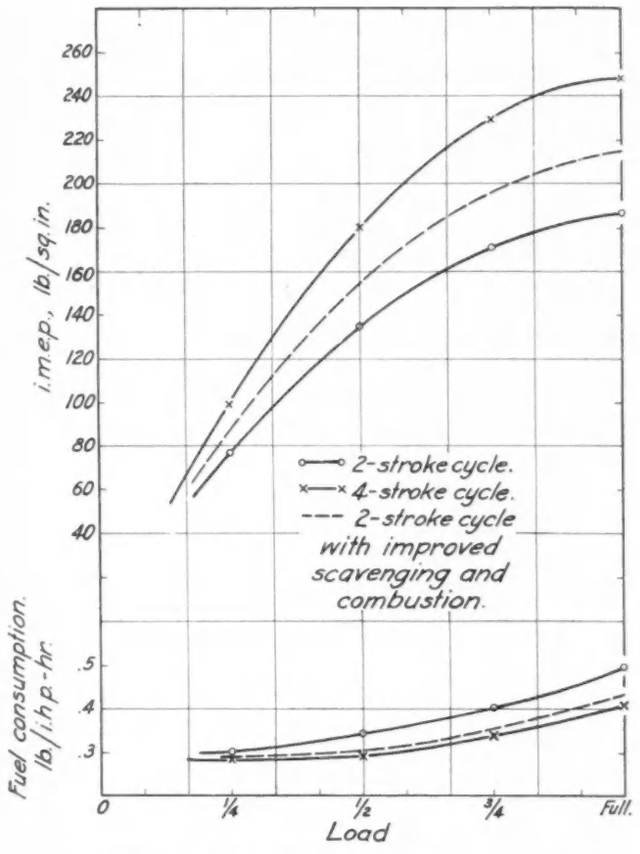


Fig. 12 – Variable Load Performance of Two- and Four-Stroke-Cycle Engines; 2350 R.P.M.; Boost, 10 Lb. Per Sq. In.; Inlet Air Temperature, 115 Deg. Fahr.

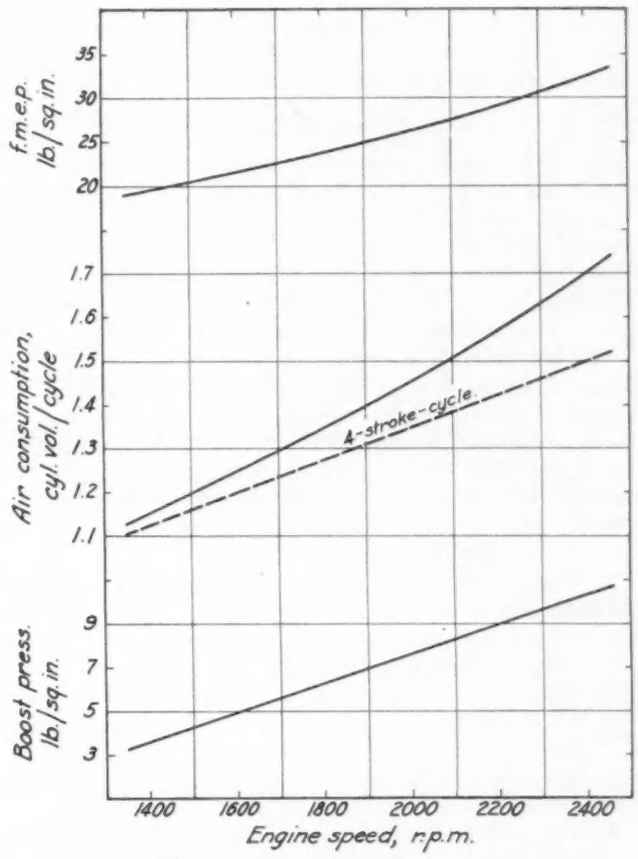


Fig. 13 – Supercharger and Friction Curves for Two-Stroke-Cycle Engine

increase of, say 15 per cent. Outputs reasonable of expectation may then be seen as dashed curves on Fig. 14.

Returning to Fig. 12, it is estimated that the combustion rates of the two- and four-stroke-cycle engines are not developed equally, and that application of known combustion aids plus improved scavenging can lower the two-stroke-cycle fuel consumption to within 5 per cent of the four-stroke-cycle values. This loss in efficiency is indicated by analysis of the effect of decreased length of expansion stroke required by exhaust timing. Plotting the four-stroke-cycle fuel consumption curve and increasing the values by 5 per cent produces an estimated two-stroke-cycle consumption curve for combustion comparable with that of the four-stroke-cycle. The improvement in consumption results in the improved mean effective pressures as shown. Since the two-stroke engine produces these mean effective pressures twice to each power stroke of the four-stroke engine, the two-stroke engine may be expected to develop not less than 43 per cent and reasonably 60 per cent more power than the four-stroke engine at the expense of some 5 per cent increase in specific fuel consumption.

This advantage in output per cubic inch for the two-stroke-cycle engine is not obtained without disadvantages. For example, the two-stroke-cycle will require a blower of over twice the capacity and hence more weight; two-stroke-engine pistons and valve mechanism will be heavier, connecting-rods and crankshaft probably will be lighter. A substantial advantage for the two-stroke engine seems likely in radial form where the heavy inertia loads of the four-stroke engine will be avoided. The poppet-valve mechanism for the two-stroke radial will be very difficult, if not impossible; however, a sleeve valve likely could be employed. In general, it is thought that the specific weight of the two-stroke-cycle would be less.

It must be remembered that these estimates are based on an inlet temperature of 115 deg. fahr. and a cylinder of 1.51:1 stroke-bore ratio. No information is yet available concerning the effect of variation in either of these factors. Higher temperatures would be obtained in practice and lower stroke-bore ratios would be desirable from weight and

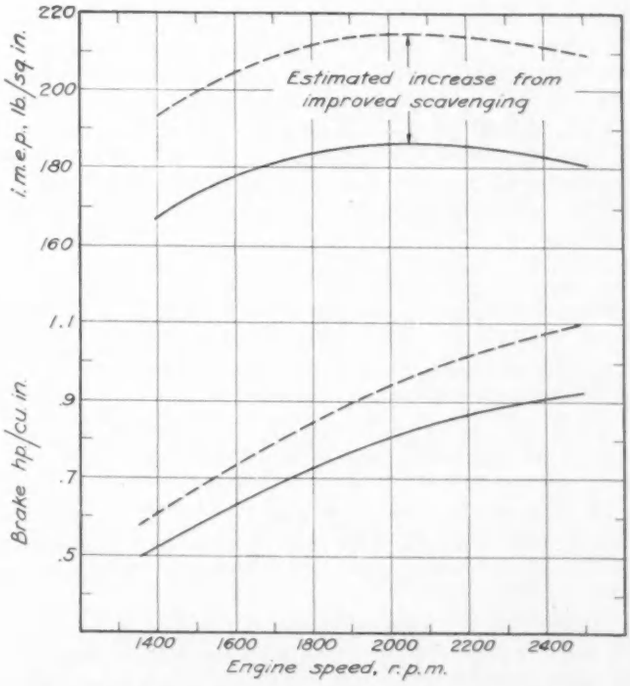


Fig. 14 – Maximum Performance of Two-Stroke-Cycle Diesel Engine with Boost Curve of Fig. 13

overall dimension considerations. Both changes would combine to decrease the performance advantage of the two-stroke over the four-stroke-cycle engine.

Additional Advantages

Before closing, a word relative to certain advantages thus far unmentioned in this paper is considered appropriate.

From considerations of power and economy, the Diesel engine holds the unique advantages that no limitations of boost, air temperature, or air-fuel ratio are imposed by the knocking value of the fuel with a consequent high power and low fuel consumption attainable in the same engine.

On the basis of simplicity, the absence of knocking and carburetor icing removes all requirements for complicated automatic controls of boost, temperature, and mixture. This item is important not only from manufacturing and maintenance considerations, but also from the pilot's viewpoint in military aircraft where his chief consideration should be defense and offense in military tactics rather than watching and controlling engine conditions.

Increasing the power of the Diesel engine does not increase the demands on the fuel-injection system to the extent that demands on the spark-plugs and ignition system of the gasoline engine are increased as its power is increased. Likewise, factors acting to increase the power of the Diesel engine demand no increase in fuel quality.

Cheaper fuels and better economy result in economic savings to operators. Reduction in fuel evaporation losses in handling constitutes an advantage in some localities.

Relative to reduced fire hazard with Diesel fuel, the authors feel that the importance of this point is not appreciated fully in commercial aviation and that the public some day will be no more required to fly in gasoline-fueled airplanes than in hydrogen-filled airships. The potential danger of gasoline in high-output engines was impressed very definitely in the minds of the authors at recent air maneuvers at Langley Field when an airplane with a glycol-cooled engine and an exhaust turboblower caught fire in the air and crashed in an attempted landing, immediately bursting into such intense flames as to preclude any possible rescue of personnel by the fire-fighting equipment immediately available at the scene of the crash. Although the general opinion prevails that fire in the air is not very probable, the risk is always present and the passengers' chances for escaping with their lives from an airplane afire in the air would be nil. The danger of fire in the forced landing of a commercial airplane is apt to result in fatalities otherwise avoidable. To date, no records can be found of a Diesel-powered airplane having burned in a crash, nor is such an accident likely to occur with the non-volatile fuel used.

Conclusions

From the foregoing analysis it is concluded:

(1) The four-stroke-cycle Diesel engine with the present technique of combustion control can, through the use of higher boost pressures and with maximum cylinder pressure not over 1400 lb. per sq. in., provide a maximum power for take-off equal to that obtainable from present gasoline engines using 100-octane fuel to maximum power advantage.

(2) The same Diesel engine can attain lower specific fuel consumptions than the lowest obtainable from the gasoline engine using 100-octane fuel to maximum economy advantage.

(3) Because the antiknock value of the fuel does not limit the performance of the Diesel engine, it offers the possibility of using very high cylinder pressures to the advantage of power and fuel economy, if engine materials and design technique are developed to use them.

(4) The use of aftercoolers with the highly boosted Diesel engine offers substantial increases in mean effective pressures obtainable for a given specific fuel consumption.

(5) The altitude performance of the Diesel engine will be fully equal to that of the gasoline engine, although the higher boost required of the Diesel may require more stages in the blower.

(6) The two-stroke-cycle Diesel engine offers the possibility of greater power per unit of size and weight than a similar four-stroke-cycle Diesel engine at the expense of increased fuel consumption.

(7) Successful development of the Diesel engine requires research in improving combustion, improving materials and design technique to accommodate higher cylinder pressure, and improvement of blowers and aftercoolers.

Discussion

Compares Service Performance of Two Types

– Robert Insley

Pratt & Whitney Aircraft, Division of United Aircraft Corp.

THE authors are to be congratulated not only upon the presentation of this paper but also upon the great quantity of painstaking laboratory work which must have preceded it. Under the circumstances a high degree of enthusiasm in favor of the compression-ignition engine for aircraft is to be expected, but I believe a few of the statements in this paper might be modified without detracting from its effectiveness. In fact, as I first read the paper I felt strongly that the cause might have been espoused even more effectively with a little less ardor.

A particularly provocative statement occurs early in the paper as follows: "Ten years of work have resulted in an increase in i.m.e.p. from 100 lb. per sq. in. at 1500 r.p.m. to 260 lb. per sq. in. at 2500 r.p.m. and have enabled every-day Diesel engine operations at outputs equal to or in excess of those obtained from our best gasoline aircraft engines." This statement refers obviously to every-day operation of N. A. C. A. single-cylinder laboratory compression-ignition engines in comparison with service performance of completely developed and approved multicylinder gasoline engines. The authors must be aware that, in at least five engine laboratories in this country alone, there is *"every-day"* single-cylinder, and even multicylinder, gasoline engine performance exceeding by a wide margin the 260 lb. per sq. in. i.m.e.p. at 2500 r.p.m. cited here, and that not with inlet air temperatures of 87 deg. fahr. but with temperatures corresponding to those actually obtained at equivalent "boost" without intercoolers.

My reaction was similar later in the paper when, after assuming "a 100 per cent efficient aftercooler" the statement is made that "This performance is superior to that obtained with present gasoline aircraft engines." Of course it is. And similarly gasoline engine performance would be enormously superior to what it is if a 100 per cent efficient cooler could be used. In fact, there are published results showing something over 600 lb. per sq. in. i.m.e.p. obtained with a gasoline cylinder with fuel roughly equivalent to commercial 100-octane fuel and with room temperature carburetor air. And I have seen recently the data on a cylinder which unfortunately cannot be named showing an indicated specific fuel consumption of 0.39 lb. per hp-hr. at an i.m.e.p. of 242 lb. per sq. in. with crankshaft speed of 3000 r.p.m. and compression ratio of 6.5:1; obtained with 100-octane gasoline and a carburetor air temperature of about 240 deg. fahr.

I must quarrel also with another statement: "It must be remembered that minimum consumptions of 100-octane gasoline engines are not to be expected below 0.38 lb. per hp-hr." This was an assumption of the authors and will find little support among those gasoline engine manufacturers who are regularly recording brake specific fuel consumption well below 0.38 lb. per hp-hr. in the laboratory. The next sentence to the effect that the take-off b.m.e.p. of such engines will be limited to 185 lb. per sq. in. apparently refers to engines of extremely high compression ratio and high temperature mixtures because, with either normal compression ratio or the intake temperatures required to produce the compression-ignition performances quoted, the take-off mean effective pressure can be carried well above 200 lb. per sq. in.

Briefly it might be interesting to consider the following conclusions which might be derived from comparing compression-ignition and gasoline engines under *similar* operating conditions:

(1) At equal pressures and temperatures the spark-ignition aircraft

engine produces higher output than the compression-ignition engine has attained.

(2) Actual service fuel consumptions of spark-ignition aircraft engines are below 0.42 lb. per b.hp-hr. and test results indicate the possibility of lowering cruising consumptions to 0.36 lb. per b.hp-hr. under favorable conditions. Compression-ignition aircraft engines in actual service reach a value of 0.36 under favorable conditions.

(3) Although many compression-ignition aircraft engines have been *designed* to weigh less than 1.5 lb. per hp. there is no compression-ignition engine in flight service weighing less than approximately 2 lb. per hp. (without including cooling system weight). Those engines noted in this paper which are expected to weigh much less are liquid-cooled engines and must include a cooling system which will increase the weight by at least 0.25 lb. per hp.

I am one of those uncommon individuals in the aircraft-engine industry who can be convinced without great difficulty that the compression-ignition engine some day will play an important, though perhaps somewhat restricted part, in aircraft activities, and I am delighted to see the progress being made at N. A. C. A. and elsewhere in that direction. But, as many of us are painfully aware, there is a long, rough journey between single-cylinder performance and a marketable engine and it seems to me that one of the most serious mistakes that can be made just now is to raise false hopes about the propinquity of the aircraft Diesel engine. A very pertinent and judicious statement in this connection was made recently by R. K. Evans, vice-president of General Motors Corp., and I hope he will forgive me if I quote a portion of it:

"But the millennium has not arrived nor is it just around the corner, and I am beginning to believe that overstimulation of the public's expectation with regard to the application of Diesel power is not good for the business. If anything can delay temporarily the march of the Diesel, it is popular disappointment."

Possibilities of Round-Engine Aircraft Diesels

– E. S. Hall

Manager, The Round-Engine Patents

IN spite of the encouragement of this paper, if my horizon were limited to radial and in-line engine types, I might agree with the feeling expressed by those now building high-output gasoline engines, that the Diesel has not enough to offer to justify the change from the present production. But my horizon is not so limited. As most of you know, I have believed in the round or barrel-type engine with cylinders parallel to the shaft, and this type of engine is peculiarly excellent as a two-stroke Diesel engine for aircraft.

My faith in round engines has been founded on the inherent advantages of the type rather than on any particular mechanism for it. Of the several mechanisms possible, most are impractical, but we know two or three practical solutions of this problem, and one in particular is definitely good engineering – a more efficient and better way to convert piston thrust into shaft rotation than provided by the connecting-rod and crank.

Applying that excellent mechanism to the two-stroke Diesel cycle, two arrangements may be had: the opposed-piston engine, and the "normal" type with double-ended piston units and working cylinders at both ends of the engine. In either case, the scavenging air would be furnished by centrifugal blowers.

The opposed-piston arrangement is free from valves and uses a combustion system accepted by some as the best, but two engine mechanisms on the same shaft are required and mechanical conditions in them are more severe than those in the "normal" engine. Moreover, in smoothness and freedom from vibration the opposed-piston engine cannot approach the normal engine and, on the whole, the opposed-piston arrangement, good as it may be, can be considered only second best.

The normal round engine as a two-stroke Diesel for aircraft makes possible the attainment of results not possible in any other known way. For example, on the basis of the performance outlined in Figs. 12, 13, and 14 of the paper, an engine of the following specifications should deliver 3000 hp. at 2000 r.p.m.: 18 cylinders 5.5 x 8 in.; 3420 cu. in. displacement; body of the engine 35 in. in diameter by 68 in. long; and specific weight less than 1 lb. per hp. and possibly as low as 0.6 lb. per hp.

Such an engine is essentially double-acting. Every stroke of the piston unit is a working stroke. Piston inertia forces serve the useful purpose of offsetting the peaks of cylinder pressure, producing two moderately heavy impulses per stroke. The torque curve shows 36 impulses per revolution from the 18 cylinders and a variation from the mean of less than 2 per cent – incomparably smooth. Only the piston heads and cylinders know the engine to be a Diesel; the mechanism connecting the pistons to the shaft is subject to no higher loads than it would be in a gasoline engine.

This engine at 2000 r.p.m. has an average piston speed of about 2700 ft. per min. – considered satisfactory by the authors for radial engines. But this round engine has a cross-head construction. The pistons run straight and true, without cocking or slap, and without bearing on the cylinder walls. The rings have a fair chance to function. Probably much higher piston speeds can be used successfully.

Moreover, the cylinder arrangement is convenient for a peculiarly simple and effective exhaust-valve system which can give exceptionally free breathing and yet be free from any speed limitations within the range of desirable engine speed. The possibility exists, therefore, of operating effectively at more than 2000 r.p.m. If so, the bulk and weight of the engine could be reduced correspondingly.

Without going into further details, it is clear that, in thinking about the development of the Diesel for high-output aircraft engines, it is obviously good sense to consider carefully that type of engine which offers such unique mechanical advantage to supplement and enhance the advantages of the Diesel cycle itself, which have been so well established in the paper.

Discussion of Snead Paper on "Truck Maintenance Problems"

(*Continued from page* 160)

Maintenance Experience with Similar Equipment

– Peter Glade

Superintendent of Equipment, Purity Stores, Ltd.

TRUCK maintenance problems differ in many respects due to the types of business in which they are used, and most of our service work is performed while the equipment is being loaded.

On the basis of our experience we give our road equipment inspection service every 5000 miles – adjust valves, injectors, blow down filters, and change oil. Valve-grinds, pistons, rings, cylinder liners and connecting-rod bearings are never touched except on major overhauls – which at present stand at 120,000 miles – and complete records are kept of all work done.

We operate 14 Cummins Diesel-powered trucks, and two of the first Cummins automotive Diesel motors on the West Coast have already run in excess of 1,700,000 miles.

As a result of our experience with this type of powerplant, these motors are driven within the range of 1250-1550 r.p.m. at water temperature of about 160 deg. fahr. Lugging motors down below 1000 r.p.m. is not permitted. The fuel pump setting is never over 26½ cc., and the average fuel-pump mileage is 60,000 miles. The cylinder liner wear does not exceed 0.007 in. tapered and 0.004 in. out-of-round.

The injector maintenance and repair are very important. The inner cup must be ground into the outer cup with care so that the two seats will hold simultaneously (a special tool has been made up to hold these cups true to one another as they are ground) or else the check-valve ball will carry the power-stroke pressure and become damaged in a very short time. The plunger blowby cannot exceed 10 per cent.

We have built a hydraulic-injector testing machine by which injectors are tested as repaired. This method cuts down the time required to repair them and also takes out most of the guess work. Every 25,000 miles the injectors are changed in the motors.

Ring-sticking has been overcome in the following manner: After the first 50,000 miles, as the oil is changed every 5,000 miles, the crankcase is refilled with 3 gal. of fuel oil and 1 gal. of valve oil, then the motor is run for 20 min. with the radiator blanketed and the rocker arms, valves, and so on, are washed with a small brush, using fuel oil. We find that, with this type of flush, the motor's performance picks up and it stays clean inside, thus adding longer life to rings, cylinders, camshaft bushings, and main bearings, and we have no rod bearing failures. We were never able to take up connecting-rod bearings before 75,000 miles. Connecting-rod journals are reground when 0.005 in. or more flat. We set connecting-rod bearings at 0.003-in. clearance and mains at 0.004-in.

Fuel-pump governor linkage failures have been eliminated by using a small filter in the oil line between the motor and the governor housing. This filter has a cartridge set on a spring so that, in case the cartridge is filled, the oil pressure will raise it and by-pass the oil to the governor housing.

Camshaft gears having wear in excess of 0.005 in. are replaced, due to the fact that this condition retards the injection, causing loss of power, poor fuel economy, and so on. Camshafts with injector lobe lifts having less than 130 deg. are replaced. Injection should start 40 deg. before top-center and end 12 deg. after top-center, checking by a timing disc.

New Technique for Noise Reduction

By Ernest J. Abbott
Physicists Research Co.

GREAT simplification of understanding and unusual results in production often follow new approaches to old problems. When noise problems are stated in terms of the familiar physical units of pressure, velocity, weight, and stiffness, basic ideas are obtained which can be applied directly to practice. In this way, most of the mysteries and the contradictions of noise problems are eliminated.

In their elements, noise problems involve only simple physical factors which are understood easily, and which can be measured with available equipment. Similarly, the solutions involve the straightforward application of known and definite engineering principles.

Although simple in their elements, most practical noise problems are very complex because of their combinations. Often much ingenuity is required to measure the physical characteristics of the noise which determine the human impressions obtained from it. Such determinations can be made by proper modification of the attack and the measuring equipment, and are the key to successful noise reduction.

This paper gives a classification of the various means of quieting, and outlines the technique of applying them to production. The viewpoint is quite different from that usually employed, and the practicability of the method is illustrated by examples of actual jobs.

PRACTICAL quieting jobs consist of two parts:

(1) Diagnosis of the problem to determine the important noises present and the mechanism by which they are produced.

(2) Application of quieting means.

An earlier S.A.E. paper by the author dealt with the diagnosis of noise problems in terms of the human sense of hearing, and the physical characteristics of sound.[1] The present paper deals solely with practical quieting means.

There is nothing mysterious or complicated about the means used to reduce noise. Noise problems are often complicated and may require much work to unscramble, but the difficulties are almost entirely in connection with the diagnosis, and not in the treatment. Quieting means involve simple and familiar engineering quantities like weight, stiffness, pressure, velocity, time, and so on, and no others. The quantities follow the ordinary rules of mechanics, and disturbing factors like relativity and quantum theory are not encountered.

In general, quieting means can be classified as follows:

Quieting Means

(1) Reduction of mobility-force product.
(2) Reduction of radiation resistance.
(3) Enclosure.
(4) Acoustical absorption.
(5) Introduction of mechanical damping resistance.
(6) Greater accuracy of parts.
(7) Shifting of frequencies to less objectionable ranges.

Although these terms may be unfamiliar to many engineers, examples of their application are not. I know of no type of machinery which has been subjected to the amount of work and expense for quieting as has the automobile. Probably most readers will find parallels in their own experiences of the principles discussed here. In many cases these experiences represent the sad ending of very costly experiments. A better understanding of the principles involved reduces the losses caused by such experiments.

Acoustics is a reasonably complex science, and can be considered from different angles. It is quite possible to express formulas and ideas in terms which are technically correct, but which are awkward to apply. For practical use in reducing noise, the author and his associates have used special concepts and technique. Obviously, these concepts must be consistent with the general theory of acoustics, but they are formulated for practical application. Experience on actual jobs has proved the value of the methods which are outlined herewith. The paper consists of three sections:

(1) Concepts.
(2) Outline of Quieting Methods.
(3) Examples of Practical Quieting.

Section 1 – Concepts

What Constitutes an Appreciable Reduction in Noise? – One of the first questions asked in connection with noise is: "What is an appreciable reduction?" This amount depends somewhat on what the observer expects, and the relative cost of the various amounts of quieting.

The following table lists the usual reactions to various reductions:

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1938.]

[1] See S.A.E. TRANSACTIONS, August, 1934, pp. 271-287: "The Place of Sound Measurements in Automotive-Noise Reduction," by Ernest J. Abbott.

Table 1 – Usual Human Impression of Various Decibel Reductions

Reduction in Sound Level, db.	Judgment
1 or less	Barely detectable, not worth spending money to obtain.
2	Just noticeable, hardly worth considering.
3 to 4	Noticeable, some improvement.
5	Definite improvement, worth some expenditure.
10	Striking improvement, worth considerable cost.
20	Outstanding improvement, is an altogether different class of machine than before.
25 and more	Almost like "off and on."

The preceding relation of decibel reduction to human impression applies quite generally when *all* components of a sound are reduced in proportion. In most practical instances, a change of level is accompanied by a change in the relative proportions of the different components of the sound, and the effect is more complex. If one is primarily interested in a particular component of the noise, as is often the case, the improvement *in this component* follows closely the data just given. On the other hand, the component in question may contribute very little to the total sound level. In such cases, even large changes in the component may affect the total sound level by less than 1 db. These facts were discussed at some length in the earlier paper already mentioned.[1]

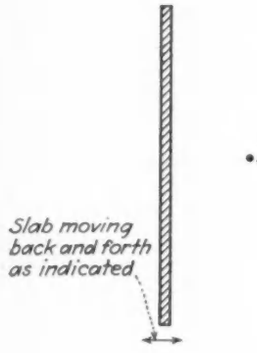


Fig. 1 – Oscillating Slab as Simple Sound Source

It is common experience that marked improvements in noise-reduction can be made by removing components which reduce the total sound level by an amount hardly more than the accuracy of measurement. The solution of this difficulty is to measure the changes in the components in which one is interested. These changes must be of appreciable magnitude (several decibels) if the ear is to hear appreciable differences in the sound.

Ear versus Instruments – The ear is very poor for making noise comparisons if time elapses between observations. On the other hand, if rapid "back-and-forth" comparisons can be made, the ear can detect easily differences in sound level and sound quality which are often next to impossible to measure instrumentally. For best results, the interval between ear comparisons should not exceed a few seconds.

The value of sound meters does not lie in their ability to measure differences which cannot be heard by ear, but in determining how to make reductions which can be heard by ear, and in making measurements where rapid "back-and-forth" comparisons are impossible.

Velocity and Sound Pressure – Although decibels are a much-used and convenient unit for expressing sound measurements, they often tend to obscure the physical dimensions and quantities on which noise reduction must rest. This section deals with those physical quantities.

Consider the case of a simple sound source such as the slab shown in Fig. 1. Assume that it moves back and forth as a unit in the direction indicated. If the size of the slab is large compared with the wave length of the sound, it will generate essentially a plane wave for nearby points. The air particles next to the slab will move back and forth with the slab and this motion will be communicated to adjacent particles in the form of pressure waves. A suitable pressure gage placed at the point *P* would indicate the pressure fluctuations shown in Fig. 2. If the sound level is 80 db. (a common value for automobiles), the range of this pressure fluctuation is from 2.8 dynes per sq. cm.[2] (0.000041 lb. per sq. in.) above atmospheric pressure to 2.8 dynes per sq. cm. below atmospheric pressure. In order to produce this pressure, the air particles are vibrating back and forth with a maximum velocity at their center of travel of 0.07 cm. per sec. (0.028 in. per sec.). As stated previously, this velocity is the same as that of the slab.

The sound pressure in air is proportional to the particle velocity and, close to the vibrating source, the particle velocity is essentially the same as the velocity of the source. At greater distances from the vibrating object, the particle velocity is less than that of the vibrating source because of the spreading of the sound waves. In general, however, the particle velocity and hence the sound pressure are proportional to the velocity of the vibrating source.

On the other hand, the loudness of a sound as heard by the human ear is very nearly proportional to the sound pressure. If the pressure is doubled, the loudness also is doubled.[3]

These fundamental relations make the physical quantities of velocity and sound pressure most useful in dealing with noise as they serve as practical measures of cause and effect. The cause of noise is the velocity of vibrating sources, and the effect produced is the sound pressure. This condition is

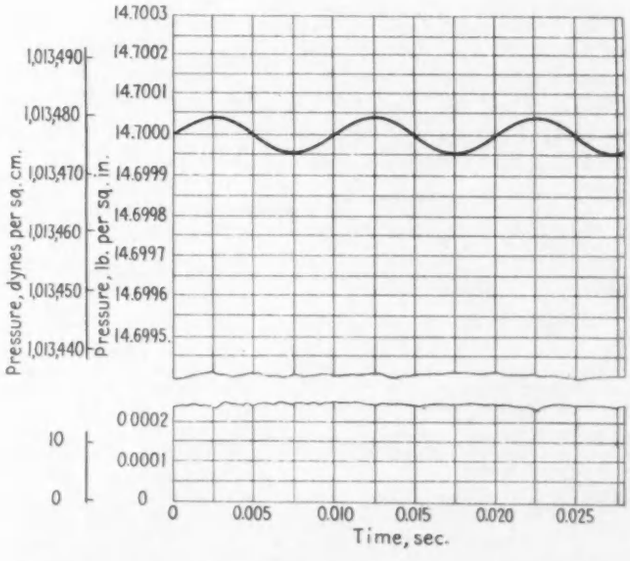


Fig. 2 – Pressure-Time Plot of 100-Cycle Sound of 80-Db. Level.

[2] By definition the decibel level of a plane sound wave is:

$$\text{db.} = 20 \log \frac{P}{0.0002,} \qquad \text{or db.} = 20 \log \frac{v}{0.000005}$$

where P = r.m.s. (root mean square) sound pressure, dynes per sq. cm.
where v = r.m.s. sound particle velocity, cm. per sec.
By definition also, the maximum instantaneous value of P or v is $\sqrt{2}$ times the r.m.s value.

[3] For sounds of ordinary level (60-90 db.) and a frequency range of 80-200 cycles per sec. which are common in automobiles, this relation is extremely close. For other levels and other frequencies, the values spread to either side. See *The Journal of the Acoustical Society of America*, July, 1933, pp. 25-30: "The Estimation of Fractional Loudness," by P. H. Geiger and F. A. Firestone.

especially fortunate as instruments are available for the direct measurement of both of these quantities.

In the foregoing discussion, no mention was made of the frequency of the vibration. For most of the practical ranges of audio frequencies and sound levels, the relations between velocity, pressure, and loudness are essentially independent of frequency so that frequency measurements are not required to determine levels of noise or vibration. On the other hand, frequency measurements are one of the most powerful tools for diagnosis, and find great application for this purpose.[1]

The relations outlined in the previous two paragraphs are basically correct and can be used to practical advantage. Complete simplicity of this sort is, of course, too good to be true, and there are other factors which are important in specific cases. One of the most important is the small amount of sound emitted from sources at low frequencies. A subsequent section of the paper deals with this effect under the heading of "Radiation Resistance." A second deviation concerns the pressure-loudness relation. Loudness is a psychological quantity, and not a physical one, and the relation between sound pressure and loudness depends considerably upon the composition of the sound.[4] A discussion of these effects is beyond the scope of this paper. In spite of second-order effects which are sometimes important, the foregoing relations of velocity, pressure, and loudness are fundamentally correct and serve as useful concepts.

Relation of Noise Level to Physical Dimensions – On the basis of the foregoing, the data of Table 1 can be translated into relative vibrational velocities of sound sources.

Table 2 – Approximate Relations of Quantities Involved in Noise Reduction

Reduction in Sound Level, db.	Per cent reduction[5] in: Velocity of Source Vibration, Sound Pressure, Approximate Loudness	Usual Ear Impression
0	0	Original sound.
1	11	Barely detectable.
2	21	Just noticeable.
3-4	29-37	Some improvement.
5	44	Definite improvement.
10	68	Striking improvement.
20	90	Outstanding improvement.
25	97	Almost "off and on."
80	99.99	Complete inaudibility for initial level of 80 db.

These data show that, in order to obtain worthwhile quieting, the velocity of the vibrating source, and hence the sound pressure, must be reduced by about 50 per cent, and that to obtain striking quieting, the reduction must be around 70 per cent. Even with these large reductions, most noises are still many, many times the least audible value.

Subtraction of Velocity and Sound Pressure – If a sound is composed of two equal components and one of them is removed, the velocity (or the pressure) is not reduced by 50 per cent, but only by 29 per cent. The reason for this fact is purely physical, and herein lies one of the most troublesome aspects of practical noise reduction.

When two oscillating quantities such as vibrational velocities or sound pressures are combined, they add at certain parts of their cycle and subtract at others. Consequently, the sum is not obtained by ordinary addition, but by taking the square root of the sum of the squares of the individual values. This fact is derived easily mathematically, and checked easily experimentally.

On this basis it is possible to compute the size of the component which must be removed in order to obtain a given reduction in vibrational velocity.[6] The results are given in Table 3.

Table 3 – Relation of Removed and Remaining Components of Sound or Vibration for Various Reductions

Reduction in Sound Level, db.	Velocity of Source Vibration, Sound Pressure, Approximate Loudness: Size of Remaining Component (A), per cent of total	Size of Removed[7] Component (B), per cent of total
0	100	0
1	89	46
2	79	61
3-4	71-63	71-78
5	56	83
10	32	95
20	10	99.5
25	3.2	99.9

These figures show that, in order to reduce a noise (velocity or pressure) to one-half its value, it is necessary to remove a component which has an individual value equal to 87 per cent of the original total. To cut the sound pressure to one-third its original value the removed component must individually equal 95 per cent of the original velocity or pressure. It is often awkward that velocities and pressures add and subtract this way, but such is the fact.

Most Practical Noises Have Four or Five Components of About Equal Value – If one of the component sound pressures of a noise is two or three times as great as any of the others, the level is practically determined by this component. Such components are recognized easily, and usually it is quite easy to devise means for reducing this part of the noise. Such changes produce a noticeable reduction in the noise, and usually no measurements are necessary to determine the effect.

However, one can be reasonably sure that such obvious components already have been removed if any efforts whatever have been made to quiet a machine. Accordingly, the practical situation in noise problems is to have several components of about equal size. From this condition it is axiomatic that even the complete removal of any one or two of them will have a comparatively slight effect. This is common experience, and is illustrated numerically by Table 4.

Table 4 – Effect of Removing One or Two Components of a Sound Having Five Approximately Equal Components[8]

Assumed Sound	Level, db.	Sound Pressure, dynes per sq. cm.	P^2
A	74	1.00	1.00
B	71	0.71	0.50
C	70	0.63	0.40
D	68	0.50	0.25
E	65	0.35	0.125
Total	77.6	1.51	2.275

[4] See *The Journal of the Acoustical Society of America*, October, 1933, pp. 82-108: "Loudness, Its Definition, Measurement, and Calculation," by Harvey Fletcher and W. A. Munson.

[5] These values in the second column were obtained by substituting in the formula:

$$\text{db.} = 20 \log \frac{v}{100 \text{ per cent}}, \text{ where } v = \text{vibrational velocity.}$$

[6] The assumption is made that the removed and remaining components have different frequencies, or that they are averaged over a wave-pattern area if of the same frequency. These are the practical cases.

[7] This column was computed from the relation: $A^2 + B^2 = (100 \text{ per cent})^2$. For oscillating values of this type, it is practice to give r.m.s (root mean square) values.

[8] This table is computed by the formulas

$$P_{Total} = \sqrt{P_A^2 + P_B^2 + P_C^2 + P_D^2 + P_E^2}$$

$$\text{db.} = 20 \log \frac{P}{0.0002}$$

Additional substitution in these formulas shows that:

Component Removed	Remaining Sound Pressure, dynes per sq. cm.	Sound Level, db.	Reduction, db.
A	1.13	75.0	2.6
B	1.33	76.5	1.1
C	1.37	76.7	0.9
D	1.42	77.1	0.5
E	1.47	77.3	0.3
B, *C*, *D*, and *E*	1.00	75.0	2.6
A and *B*	0.88	72.9	4.7

It will be remembered that, to obtain worthwhile reductions, the sound pressure should be cut to ½ or ⅓ of its initial value. The preceding table shows the impossibility of doing this job by working on any one or two of the components. In general, all components whose pressures are as much as 25 per cent as large as the total must be reduced to obtain real quieting. This statement is somewhat of a paradox, as the removal of such a component would reduce the total pressure by only 3 per cent. Herein lies the most fundamental difficulty of practical noise reduction. If total sound-level readings are used to measure the effect of a change, it is often impossible to obtain sufficient accuracy to allow the size of the removed component to be determined. If analysis must be used to focus attention on the components under study, the increase of time and cost is usually very great. Practical jobs require much ingenuity and modification of measuring means for the job to obtain usable results in a reasonable time.

Cancellation of Sounds – There is a very widespread theory that one sound can be cancelled by another. Usually no mention is made of the three conditions required.

(1) Two acoustically identical units must be used. (The author has never seen such machines.)

(2) They must be operated exactly ½ cycle apart. (This timing would be very difficult to maintain.)

(3) The two units must occupy exactly the same space at the same time. (This condition is very difficult.)

When two somewhat similar machines are placed near each other, the sound may cancel at certain points, but it inevitably adds at others, and experiments show that, on the average, one obtains simply the sum of the sound pressures which the units produce individually (root mean square addition).

Summary of Section on Concepts – In dealing with sounds it is most useful to think of the sound simply as a pressure variation in the air. This pressure is measured in ordinary units, dynes per sq. cm. being used instead of lb. per sq. in. in order to obtain more convenient numbers. Since the sound pressure is an oscillating quantity, it is convenient to follow electrical convention and use r.m.s. (root mean square) values. The loudness of the sound as heard by the average ear is essentially proportional to this pressure, so that sounds with twice the pressure are about twice as loud to the ear. The second-order corrections to this statement are quite large, especially for complex sounds, but it is useful to use this relation as a basis, and to consider the departures as corrections.

In general, the sound pressure generated by a vibrating object is proportional to the velocity of that vibration and, accordingly, it is useful to deal with vibrations in terms of their velocity rather than their displacement and frequency. There are also important corrections to this concept, but it is a useful basis on which to work.

[9] In my earlier paper this term was called: "Mismatch of Vibrational Impedance," which was a carry-over from electrical terminology. The present concept is more useful. To the best of my knowledge, the term "mobility" was used first by Prof. F. A. Firestone, Department of Physics, University of Michigan.

[10] For oscillating quantities such as these, it is convenient to follow electrical convention and use r.m.s. (root mean square) values. For a simple oscillating quantity, the r.m.s. value is $\frac{1}{2\sqrt{2}}$ (about ⅓) of the "peak-to-valley" variation.

Experience shows that, to obtain worthwhile reductions in sound level, it is usually necessary to reduce sound pressures and vibrational velocities to ½ to ⅓ of their initial values. These values correspond to reductions of about 6 db. and 10 db. respectively.

Experience shows that noise components with pressures two or three times that of other components are located easily and treated, and that most practical noise problems in automobiles and other machinery involve as many as four or more components of about equal importance. The basic relations for combination of oscillating quantities, such as sound pressure, make it impossible to obtain a large reduction by the removal of one or two such components. Accordingly, means must be employed to reduce *all* contributing components by sizable factors (that is, a factor of 2 or 3). Means for obtaining these large reductions are outlined in the next section.

Section 2 – Quieting Methods

As outlined previously, successful noise reduction consists in reducing vibrational velocities of sound sources by at least a factor of 2 to 3, so that the resulting sound pressure will be reduced to ½ to ⅓ of its original value. Means of achieving this result are treated later. The methods used are all straightforward engineering applications of physical principles. Most engineers are familiar with the elements involved, but encounter difficulties because of lack of data on the quantitative relations. Most acoustical problems are so complex that it is almost impossible to compute results with precision and, accordingly, little attempt has been made to interpret results on the basis of fundamentals. Hence, it has been difficult to accumulate usable backgrounds of experience and thumb-rules.

It is feasible to compute the results of many experiments with sufficient accuracy to be of use. Results of experiments which appear entirely inconsistent can be shown to follow directly from elementary principles and, by making such computations, one can improve greatly his background for planning work. Throughout this paper all principles are illustrated by simple computations which explain the results obtained in practice. These computations are not intended to be complete descriptions of the job in all details, but rather to indicate the essential factors. An attempt is made in each case to indicate the simplifying assumptions, and what sort of departures might be expected from the computed values. The author appreciates fully the impossibility of accurate computations in most cases, but trusts that the reader will find these computations useful in understanding the general relations.

(1) *Reduction of Mobility-Force Product*[9] – In ordinary language, the vibrational mobility of an object is simply the ease with which it can be shaken. The mobility of a body is determined by its weight and stiffness and, under some conditions, by its frictional resistance to motion. Large friction may reduce mobility.

As mentioned previously, noise reduction is concerned primarily with reducing the vibrational velocity of sound sources. Mobility and velocity are related closely, the relation being:

Vibrational Velocity = [Mobility] times [Vibrational Force]

where velocity is given in ft. per sec. (r.m.s.)[10]
force is given in lb. (r.m.s.)

then mobility is ft. per sec. per lb.

For simple mechanical elements, the mobility can be computed. For example:

For a mass, mobility is $-\frac{g}{2\pi f W}$ (ft. per sec. per lb.)

For a spring, mobility is $\frac{2\pi f}{k}$ (ft. per sec. per lb.)

Where W is the weight of the mass, lb.
k is the constant of the spring, lb. per ft.
g is 32.2 ft. per sec. per sec.
f is the frequency of the driving force in cycles per sec.

The minus sign results from phase relations, the reason for which may be left for the mathematicians at this point.

Assuming constant vibrational force, the vibrational velocity is proportional to mobility and, therefore, inversely proportional to the weight. Hence, to obtain a reduction of 6 db., it would be necessary to double the weight, and for 10 db., to triple it. Such increases of weight would be out of the question for most automobile parts, and hence quieting by simple increase of mass is not feasible. Similarly, the stiffness of parts would have to be increased by the same factors, and this method is seldom feasible either. From this discussion it is concluded that practicable changes of weight and stiffness can be expected to produce little quieting, even *assuming constant vibrational force*. Although practical examples of this "constant-force" condition exist, the situation most frequently encountered involves changes in vibrational force when mobilities are changed. In such cases the reductions are even less. This brings us to a consideration of force-mobility relations in practice.

Constant-Displacement Versus Constant-Force Sources – Often the sources of vibration are of the "constant-displacement" type. A cam and follower is an example of this type of motion. The amount of movement is practically independent of the load moved by the cam, as any decrease in the mobility of the load is accompanied by a decrease in force required on the cam. Hence, for all speeds of operation, the displacement of the part is the same. (Fig. 3 illustrates and describes a set-up that demonstrates these sources. Fig. 4 shows a detail of the sources shown in Fig. 3.)

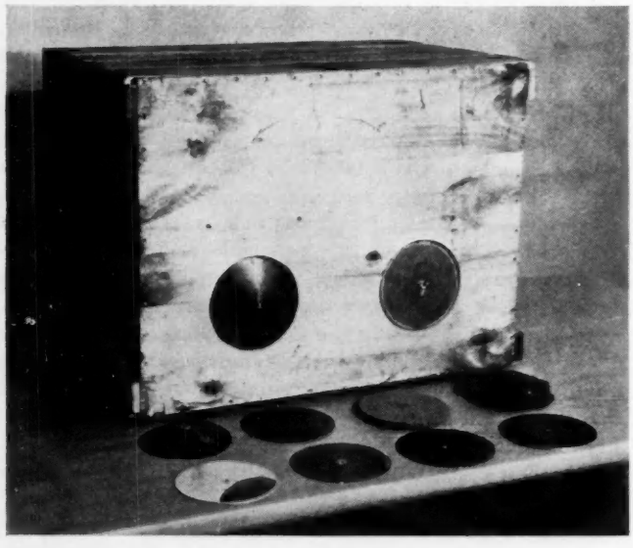


Fig. 3 – Set-Up Used at the 1938 Annual Meeting of the Society To Demonstrate "Constant-Force" and "Constant-Displacement" Sources

Constant force is applied to the disc on the left by means of a loud-speaker driving unit. Constant displacement is given the disc on the right by means of a motor-driven eccentric of 0.005-in. throw. The discs shown in the foreground can be driven from either source. The discs shown are:

Material	Weight, Oz.	Material	Weight, Oz.
1/32-in. cardboard	0.5	1/16-in. aluminum	2.6
⅛-in. dense wallboard	1.8	1/16-in. sheet iron	7.5
½-in. porous wallboard	2.6	1/16-in. iron heavily coated with viscous damping material	8.2
1/16-in. sheet iron with viscous damping material and 1-in. hair felt	9.5	⅛-in. sheet iron	16.5

On the constant-force unit, the sound pressure generated is approximately inversely proportional to the weight, so that the heavy discs produce much less noise than the lighter ones.

On the constant-displacement source, all discs produce the same level of noise. If anything, the heavy discs with the fancy treatments make slightly more noise as they vibrate more nearly as a unit. This simple illustration checks well with results obtained on actual applications.

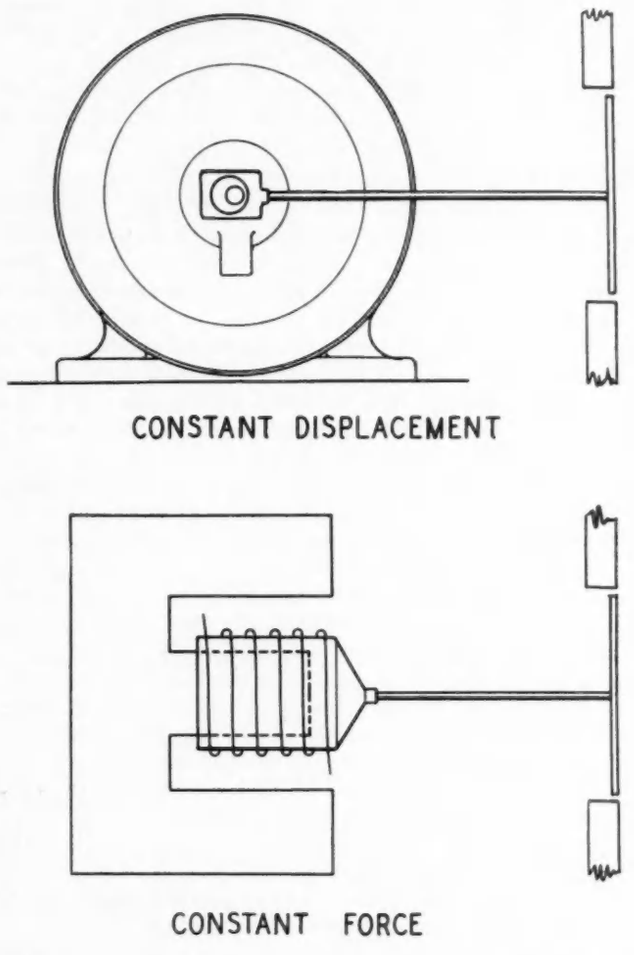


Fig. 4 – Detail of Constant-Force and Constant-Displacement Sources Shown in Fig. 3

In this case, even large changes in the mobility of the sounding object will have no effect whatever on the noise produced by a given area of surface. Effects of changing areas are considered in the next section.

Errors in gears, anti-friction bearings, and many other machine parts produce vibrations which are essentially of the constant-displacement type. There are minute peculiarities in geometry that pull and push the associated parts positively through small, but definite distances exactly like small cam movements. With such sources, it makes little difference whether the driven part of the machine is light or heavy. The sounding areas can be made of steel, brass, wood, aluminum, cardboard, or almost any other material, and the resulting sound is essentially the same. A vast amount of experimenting has been done on constant-displacement sources in trying one material after another in the vain attempt to find one with some inexplicable but magic property which would reduce the noise.

On the other hand, if one is dealing with sources of "con-

stant force" rather than of "constant displacement," great changes can be obtained by varying the mobility. Forces due to lack of balance usually are fairly independent of the resulting motion. Consequently, the motions can be very great if the mobility of the surrounding parts is high (that is, mountings which have little weight, or yield easily). Similarly, the vibration can be reduced by decreasing mobility by the use of heavy, stiff parts.

One of the first items in a noise-reduction job is to determine if the sources of vibration approximate "constant-displacement" or "constant-force" sources. This determination indicates the possibilities of quieting by changing the mobility of vibrating parts.

A practical example of these two types of force might be as follows: Suppose that one wishes to reduce the sound radiated from automobile panels. A typical panel could be set up in a laboratory, and driven by an electromagnetic unit similar to a loud-speaker unit. This is essentially a "constant-force" drive, and experiment would show that large changes in acoustic output from the panel could be obtained by changing the mobility of the panel, particularly at resonances. These changes could be made by attaching materials to the panel, or otherwise by varying the mass, stiffness, and resistance.

Having determined the optimum treatment for the panel, the next step would be to try it in a car. Tests indicate that the panels of a car are driven by essentially "constant-displacement" forces. Accordingly, it would probably be impossible to detect the difference between panels which were at extremes of the laboratory results with the other type of driving force. Much money can be spent making tests under conditions which are of little or no practical significance.

Use of Resilient Elements to Reduce Mobility – The most effective way to reduce the movement of a part is to connect it to the vibrating source by a resilient member such as rubber or springs. The most familiar example is shown in Fig. 5.

A mass m is supported resiliently on a large mass M by means of a spring S. If the large mass M is vibrated, what happens to the small mass m at different speeds?

f is the frequency of vibration, cycles per sec.
W is the weight of the large mass, lb.
w is the weight of the small mass, lb.
k is the spring constant, lb. per ft.
V is the velocity of the large mass, ft. per sec.
v is the velocity of the small mass, ft. per sec.
$g = 32.2$ ft. per sec. per sec.

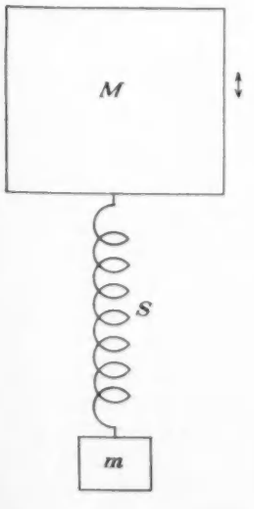


Fig. 5 – Principle of Resilient Mounting

We use the formula

$$\text{velocity} = \text{mobility} \times \text{force}.$$

From the second paragraph of this section, we have

$$\text{mobility of mass} = -\frac{g}{2\pi f w}$$

$$\text{mobility of spring} = \frac{2\pi f}{k}$$

Now, for the spring, the velocity is $V-v$, hence, neglecting resistance:

$$\text{Force on spring} = \frac{k}{2\pi f}(V-v)$$

This same force is applied to m, hence the velocity of m:

$$v = \frac{-g}{2\pi f w}[V-v]\frac{k}{2\pi f}$$

Solving $$v = \frac{V}{1-\frac{(2\pi f)^2 w}{kg}}$$

Now, as f is changed, this formula shows that:

(a) Where the speed (f) is very slow, the two masses move in unison.

(b) As f increases, m moves *more* than M.

(c) When $f = \frac{1}{2\pi}\sqrt{\frac{kg}{w}}$, m moves many times as far as M, and is limited only by friction. (Resonance condition.)

(d) When f is greater than the resonance value, then m moves *less* than M.

(e) At very high speeds, m stands practically still.

From this very fundamental example, it is seen that, to be effective, a resilient mounting must be very, very limber. Usually, a resilient support is so limber that it outrages our engineering ideas of a mechanically satisfactory job long before it is sufficiently resilient to serve as an effective means of reducing vibration. As shown by the formula, a resilient mounting which is too stiff is worse than none at all. Apparently the only remedy is to revise our ideas as to what constitutes a mechanically satisfactory support, and to accept greater deflections. It seems to be a surprisingly general experience that the limber mountings work out in practice much better than everyone expects.

Mathematical juggling of the preceding formula yields the following well-known thumb-rules:

(a) To reduce the velocity of m to ¼ that of M, the natural frequency of the system must be at least a factor of 2¼ lower than the lowest frequency it is desired to isolate. Smaller reduction than this is usually of slight practical importance.

(b) The natural frequency of a single-mass, spring-supported system is related to the static deflection by the following formula:

$$D = \frac{9.77}{f^2}$$

where D is the static deflection, in.

f is the natural frequency of the system, cycles per sec.

Hence, if one wishes a natural frequency of 10 cycles, a static deflection of 0.098 in. is inevitable. For 1 cycle, the deflection must be 9.77 in.

(c) Except for speeds very close to resonance, the effectiveness of a resilient mounting is *decreased* rather than increased by the presence of frictional resistance or damping. See Section 4.

From the foregoing it is apparent that a resilient mounting is essentially a device for transforming a "constant-displacement" source of vibration into a "constant-force" type. By allowing the initial vibrator to work against a very soft connection, there is little reaction on the initial source, so that

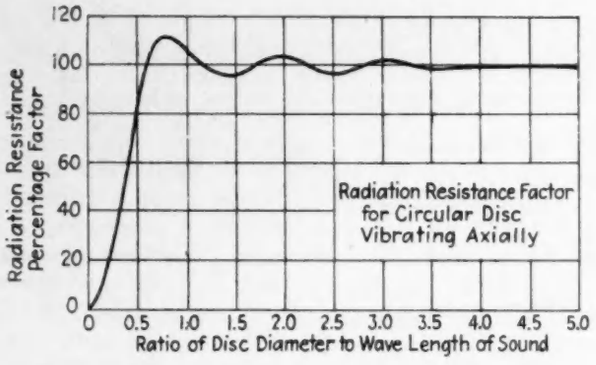


Fig. 6 – Radiation Resistance for a Vibrating Disc

little force is drawn from it. This small force, in turn, is applied to the body at the other end and, if static deflections are sufficiently large, this force is essentially of the "constant-force" type. Hence the mobility of an object which is resiliently mounted will have a large effect on its motion, and quieting can thus be achieved.

Complexity of Audio-Frequency Vibrations – The preceding concepts are fundamental, and the formulas given apply very closely, provided that the masses in the system vibrate as a unit and the weight of springs or resilient material can be neglected. For low frequencies up to about 50 cycles per sec., these requirements usually are met. For higher frequencies, the departures are tremendous. Unfortunately, in noise-reduction work the important frequencies are seldom as low as 50 cycles per sec. Frequencies from 100 to 1000 cycles per sec. are most common, although those outside this range are often of importance.

It is a sad fact that formulas fail utterly to give quantitative results for the frequencies most commonly encountered in practice. One recent paper on this important subject gives experimental results in which the measured vibrations are as much as 1000 per cent greater than the computed values.[11] This finding, of course, simply indicates that the parts were not vibrating as assumed. That such is a fact is demonstrated easily. If one takes measurements on a vibrating machine part with a vibration pickup and suitable amplifier, he finds that even the stiffest and heaviest parts do not vibrate as a unit. Instead, they break up into a large number of vibrating segments of different amplitudes and phases. These complicated vibration patterns make mathematical treatment of most practical cases either very difficult or impossible.

Unfortunately, when objects vibrate in segments instead of as a whole, this condition is equivalent to raising natural frequencies so that actual reductions are usually much less than are computed on the basis of simple vibrations. On the other hand, these complex vibrations greatly reduce the opportunities for prominent resonances, and such resonances are not nearly as common as might be anticipated.

In dealing with actual resilient mountings for audio frequencies, it often is found advantageous to use a resilient material with high internal friction such as rubber. Although the elementary theory just outlined indicates that resistance is a disadvantage rather than an advantage, other theory applies in practice at audio frequencies. In steel springs, which have little internal friction, it often happens that noise "telegraphs" down the spring. This effect is much less for rubber and similar materials with high internal resistance.

[11] See *The Journal of the Acoustical Society of America*, October, 1937, pp. 141-145: "Compliant Materials To Isolate Vibrations," by W. Jack and J. S. Parkinson.

[12] See "Vibration Systems and Sound," by I. B. Crandall, D. Van Nostrand Co., Inc., 1927.

Summary on Mobility-Force Relations – Although the complexity of the audio-frequency vibrations of most machine parts makes it impractical to compute the mobilities, and hence the mobility-force products, this difficulty by no means destroys the usefulness of the concept. The fundamentals apply even though it is often impossible to compute numerical values, and experiment must be used.

One of the first steps in a noise problem is to determine if the vibrational sources are of the "constant-force" type, or of the "constant-displacement" type. If the source is of the constant-force type, quieting can be obtained by changing mobilities. On the other hand, if the source is of the constant-displacement type, any modifications involving only mobility changes will be a fruitless waste of money.

One of the most successful methods of reducing noise is to reduce velocities by reducing mobilities or forces, or both. This method of quieting probably finds more application than all others combined.

(2) *Reduction of Radiation Resistance* – Radiation resistance is a term taken over directly from electrical terminology. It is well known that sound is radiated from vibrating objects, and that this radiated energy must be drawn from the source which drives the vibrating object. Accordingly, there is a greater resistance to the motion of the vibrator than would be the case if the vibration took place in a vacuum so that no sound energy were radiated. Since this increase of resistance can be measured easily, it is practice to speak of the amount of sound radiated in terms of this increase of resistance, which is called "radiation resistance." In common language, the radiation resistance of an object is simply a direct measure of the sound energy radiated from it. The greater the sound energy radiated, the greater the resistance to motion of the radiator. Hence, if we can reduce the radiation resistance of a vibrating object, we can reduce noise.

The principal factors controlling the radiation resistance of objects vibrating in the air are the vibration pattern of the object, and the relation of the wave length of the sound to the dimensions of the vibrating area.

As usual, precise computations are impractical for most cases, especially in view of the complexity of the audio-frequency vibration patterns of most sound radiating surfaces. Computations have been made in special cases, and Fig. 6 shows a reproduction of such a computation taken from Crandall.[12] From this curve it is seen that, as long as the size of the vibrating object is larger than the wave length of the sound, the sound is radiated with about equal efficiency. For frequencies such that the wave length is larger than the object, the amount of sound radiated by a given vibration of the object is very small, and often is only 1 per cent or less of the values obtained at higher frequencies.

These general remarks also apply to actual machine parts, although the exact ratios cannot be computed.

Effect of Area on Noise – Ordinarily, the various parts of a sounding object are vibrating in different phases, and the effect of the various areas can be computed by taking the square root of the sum of the squares of the pressures from the component areas. This method is used in the following section for computing the effect of openings in enclosures, and the formula is given in that section. Reducing the radiating surface by a factor of 2 usually decreases the sound pressure by about 30 per cent.

In general, it can be said that sounds can be reduced materially by reducing the radiation resistance if the dimensions of the vibrating parts can be made smaller than the wave lengths of the sound being radiated. At 1000 cycles per sec., the wave length is about 1 ft. and, at 100 cycles per sec., about 10 ft.

(3) *Enclosure* – One of the most effective ways for re-

ducing noise is to place an enclosure around the source, or the observer. For best results, such an enclosure should *not* be made of sound-absorbing material. Instead it should be made of a very poor sound absorber. The explanation for this apparent paradox is that the action of such an enclosure is not obtained by sound absorption. Let us consider what we mean by a sound absorber.

Fig. 7 shows a section of wall covered by a layer of sound-absorbing material. The best sound absorbers are soft, porous materials, such as hair felt, rock wool, cellulose fibers, cotton, and so on. Assume now a section of plane sound wave of pressure, p_o, incident upon the material. Most of the pressure is transmitted directly through the porosities of the material, but a portion of the pressure is dissipated in forcing air through small intricacies of the material. Assume that r represents the fraction of the pressure reaching the wall. The wall is practically a perfect reflector for sound, so the wave returns through the material and suffers another diminution. Hence,

$$p = r^2 \times p_o$$

where p is the sound pressure of the reflected wave.
p_o is the sound pressure of the incident wave.
r is the fraction of pressure remaining after a single transit of the material.

The absorption coefficient of a material is defined by the formula:

$$E_r = E_i\,(1 - a)$$

where a is the absorption coefficient.
E_i is the energy per unit area of the incident wave.
E_r is the energy per unit area of the reflected wave.

In a plane wave

$$E = \frac{p^2}{\rho C}$$

where E and p are as defined as before

ρ = density of air

C = velocity of sound

Substituting:

$$1 - a = r^4$$

At 125 cycles per sec. (which is about the frequency of the largest components of automobile noise), the best sound absorbers have a coefficient of 0.2 to 0.5. For the exceptionally high value of 0.4, the value of r is therefore $\sqrt[4]{0.6} = 0.88$.

This value means that, after passing through a couple of inches of the best sound absorber, the wave still has 88 per cent of its original pressure (1 db. reduction). As outlined in the previous section, this much reduction is of little or no practical importance. Sound absorbers have their uses, and these uses will be considered presently, but they are not suitable materials for enclosures.

Consider now a single pane of glass about ⅛ in. thick for the enclosing partition. The mobility is essentially determined by the mass, although the stiffness of the mounting enters somewhat. As stated previously, the mobility of a mass is $\frac{-g}{2\pi fW}$. The density of glass is about 150 lb. per cu. ft. Hence, at 125 cycles per sec.:

Mobility of glass per sq. ft. of area

$$\frac{32.2}{2\pi \times 125 \times 150 \times \frac{1}{8\times 12}} = 0.0262 \text{ ft. per sec. per lb.}$$

Now for an 80-db. plane wave, the sound pressure is 2 dynes per sq. cm. r.m.s., or,

$$\frac{2 \times 14.7 \times 144}{1{,}013{,}000} = 0.0042 \text{ lb. per sq. ft. r.m.s.}$$

velocity = mobility × force

velocity = 0.0262 × 0.0042 = 0.00011 ft. per sec. r.m.s.

This is also the velocity of the sound wave on the far side of the partition. The velocity on the near side is that corresponding to 80 db. As previously defined: This is,

0.05 cm. per sec. or 0.0016 ft. per sec. r.m.s.

Hence, the ratio of the two velocities is 14½:1, or 23 db.

Handbook values for actual windows are in the neighborhood of 10:1 (20 db.). This is a very close check, considering that the stiffness of the glass and its support, internal yielding, and similar factors were neglected. As in most work in this field, elementary concepts of the type here outlined give good qualitative comparisons, even though it is impossible to make accurate numerical computations.

Compare now the quieting of 1 db. obtained with an enclosure of a couple of inches of the best sound-absorbent with the 15 to 20 db. obtained with a single thickness of solid material such as glass, and the conclusion is obvious. It should be observed further that the only characteristic of the glass involved was its weight. Consequently, similar results can be expected from sheet iron, wood, plaster, or any similar non-porous material of appropriate weight.

Yielding Partitions, Resonances, and so on – The elementary concept just outlined for an enclosing wall or partition neglects yielding of the panel, resonances, and so on. In most practical cases, these assumptions are justified but, in some instances, resonance can be used to obtain special properties; in particular, very large isolation for light panels. The unusual effects are obtainable only for narrow frequency ranges, and often at the expense of isolation at other frequencies. If one must isolate against certain steady notes as in airplanes, such special constructions can be used to advantage.

Enclosures for Great Isolation – As indicated previously, it is a poor enclosure which will not give a transmission loss of 15 to 20 db., and this is ample for many jobs. On the other hand, for soundproof booths with interior levels of 40 db. or less, and for other architectural applications, it is often desirable to use partitions with 50 db. or more transmission loss. (The pressure transmission through such a partition is only 0.3 per cent.) Such reductions can be obtained by heavy

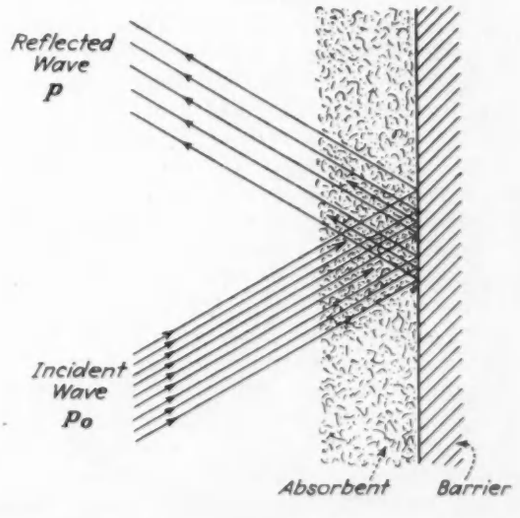


Fig. 7 – Sound Absorption

walls, and by multiple constructions. Such structures are discussed in standard works on architectural acoustics, and are beyond the scope of this paper.

Effect of Lining Enclosure with Absorbing Material – In general, the effectiveness of an enclosing shell can be improved a few decibels by lining the interior with sound-absorptive material. Most of the improvement is obtained with the first absorption added. Often it is more economical to obtain the desired reduction by using a better enclosure. An elementary computation follows:

Suppose one has a source of sound which produces a pressure p at a radius of 1½ ft. in free space. See Fig. 8. Since none of the sound is reflected, this is equivalent to 100 per cent absorption in all directions. Now, if an enclosing shell of radius 1½ ft. with absorption coefficient 0.1 (a poor absorber such as wood), and a transmission loss of 20 db., is placed around the source, the level inside the shell is increased by about a factor of 3 because of reflection.

$$\frac{p_1}{p_2} = \sqrt{\frac{a_2}{a_1}}$$

where p_1 is the average sound pressure in an enclosure of average absorption coefficient a_1.

p_2 is the average sound pressure in an enclosure of average absorption coefficient a_2.

Average pressure inside =

$$p \times \sqrt{\frac{1.00}{0.1}} = 3.16\,p$$

It is assumed that the same acoustical energy is supplied in both cases. The transmission loss of 20 db. is equivalent to a pressure ratio of 10:1, so that the overall effect of the enclosure is to reduce the pressure by a factor of 3.16 or 10 db.

If the enclosure were lined with material of coefficient 0.5, the overall reduction would be 17 db. instead of 10. A similar reduction could be obtained by making the shell of material giving a transmission loss of 27 db. instead of 20 db. and omitting the absorbing material.

Effect of Cracks and Openings in an Enclosure – Generally speaking, the value of an enclosure is very slight if it contains an opening of moderate size. Again, accurate computations are difficult, but the fundamental principle follows:

Consider the enclosure just discussed. For convenient computation, assume that it is a sphere of radius 1½ ft., transmission loss 20 db., and absorption coefficient 0.5 on the inner surface. The enclosed source is of such power that it would produce a sound pressure p at the 1½ ft. radius if the shell were not present. Table 5 shows the approximate average pressures obtained externally when openings of various size are present. The values are computed as follows:

Average absorption coefficient of interior:

$$a = \frac{S_1 \times a_1 \text{ plus } S_2 \times a_2}{S}$$

where S_1 is the area with absorption coefficient a_1.

S_2 is the area with absorption coefficient a_2.

For openings, a_2 is taken as 1.0.

With an opening of 1 sq. ft.,

$$a = \frac{27.2 \times 0.5 \text{ plus } 1 \times 1.0}{28.2} = 0.518$$

$$\text{Average sound pressure inside} = p \times \sqrt{\frac{1.0}{0.518}} = 1.39\,p.$$

The average sound pressure outside can be computed approximately as follows:

$$p_o = \sqrt{\frac{S_1 p_1^2 + S_2 p_2^2}{S}}$$

where S_1 is the area of the opening.

S_2 is the area of the remainder of the enclosure.

p_1 is the sound pressure at the opening (assumed to be the same as inside).

p_2 is the sound pressure over the remainder of the outside of the enclosure, or internal pressure ÷ 10 (for 20-db. partition).

whence

$$p_o = p\sqrt{\frac{1 \times (1.39)^2 + 27.2 \times (0.139)^2}{28.2}} = 0.296\,p$$

Table 5 – Average Reduction with Enclosure with Openings

	Per Cent of Opening	Average Sound Pressure at Radius of 1½ Ft.	Reduction, db.
No enclosure at all	100	1.00 p	0
Complete enclosure (20 db. transmission loss, absorption coefficient, 0.5)	0	0.141 p	17
Opening of 1 sq. ft.	3½	0.296 p	10½
Opening of 2½ sq. ft.	9	0.425 p	7½

Of course these computations do not include all factors, but it is obvious that the effectiveness of any enclosure is practically ruined by openings which total as much as 10 per cent of the surface.

Effect of Mechanical Vibration of Enclosure – It often happens that the noise is increased rather than decreased when an enclosure is placed around a machine. This increase is due to the fact that the enclosing shell is not driven by the sound pressure of the air as assumed in the preceding computations, but is driven by mechanical contact with the machine. When I was a boy, my grandmother used to rap on the window with her thimble to attract my attention. She did not waste her breath yelling. Apparently she knew that she could apply much larger force to the window pane by mechanical contact than by generating sound pressure inside. Since the mobility of the pane was constant, greater velocities could be imparted to it by the mechanical forces. The same is true of most enclosures.

If an enclosure could be hung on a "sky-hook" so that mechanical transmission to it would be nil, then pure acoustic driving would result. In practice, there is a dearth of sky-

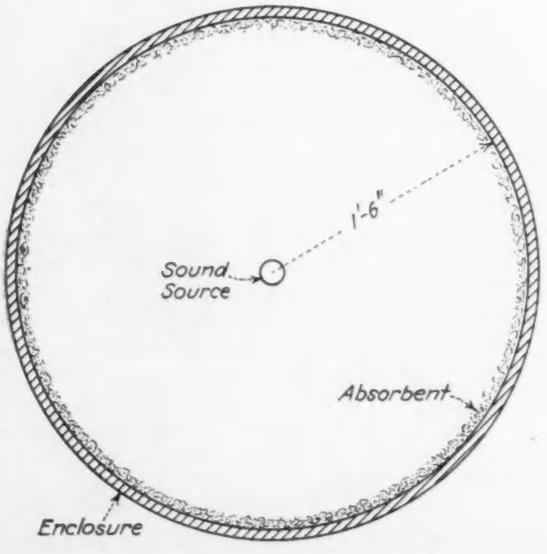


Fig. 8 – Enclosed Sound Source

hooks, and it is necessary to mount the enclosure on the machine, or on something adjacent to the machine. In such cases, the vibrational force transmitted mechanically is often greater than that transmitted acoustically. If the mechanically transmitted force approaches the "constant-displacement" type, as is often the case, the enclosure simply adds to the sound-radiating surface, and the noise is increased.

If a noise source produces large sound pressures in the surrounding air, but comparatively little audio frequency vibration in solid objects, then enclosure may be very effective. If most of the surfaces which eventually radiate the sound are driven by mechanical contact, enclosures are not effective unless they enclose all radiating surfaces, and unless the enclosure is so mounted that it is not shaken mechanically. Obviously, this arrangement requires resilient mounting of the enclosure to a high degree of effectiveness. In practice, it is usually difficult to reduce the mechanical vibration of the enclosure to extremely low levels and, accordingly, there is no point in using a partition with a high transmission loss. Usually, a loss of 20 db. (which is obtained easily with a single layer of non-porous material) will give as much reduction as can be obtained by the method on account of leakage, and mechanical driving.

(4) *Acoustical Absorption* – Acoustical absorbent materials have a most vital place in architectural acoustics, and the improvements obtainable by them are almost unbelievable. In machinery noise reduction they play a far less conspicuous part, although they have valuable applications. Even in architectural acoustics, the essential improvement is not attained by reducing sound level. Although acoustical treatment of rooms usually reduces the average sound level by a few decibels, the real improvement is due to the elimination of persisting multiple reflections or reverberation which may so "hash" the sound that speech is practically unrecognizable, and music is much impaired. In most machinery noise, which is more or less continuous, these multiple reflections are not important close to the machine where the noise is greatest, and their presence at other points is manifested chiefly by a modest increase in sound level.

As outlined in the section on enclosures, the reduction in sound level at a single reflection from even a good absorbent material is very slight. For most practical materials, at frequencies usually found in machinery, the reduction at a single reflection is practically negligible. The practical requirement for quieting by absorption is, therefore, that practically *all* of the sound reaching the ear must strike an absorber *many times* before reaching the ear. These unusual conditions are realized in the case of a noise which comes through a duct or tube which is long compared with its diameter. In such cases, lining of the duct with sound-absorbing material does a tremendous amount of good. In nearly all other cases, it is of slight advantage.

In particular, there is little to be gained in covering a vibrating sound source, such as an automobile panel, with sound-absorbing material. As explained previously, such materials are practically transparent to sound, especially if the sound has to traverse the material only once to reach the ear. Absorbent materials are usually too light to have any appreciable effect on the mobility of the object and, as explained previously, if the vibrational sources are of the constant-displacement type, variations in mobility are of slight consequence.

Results obtained from the simple application of acoustical material to the walls of an enclosure, such as the interior of an automobile or railroad car, follow quite closely the well-known formulas of architectural acoustics. Assuming a sound source of constant energy in such an enclosure, the average sound pressure varies inversely as the square root of the average absorption coefficient of the walls. (Such a computation was made in the case of the lining of the enclosing sphere of the previous section.) Since the interior of an automobile already includes a considerable area of good absorbing material in the form of upholstery, seat cushions, occupants, and so on, the application of additional sound-absorbing material hardly can increase the average absorption coefficient by more than a factor of two, which would reduce the sound pressure by 30 per cent, or 3 db. Actual experiments indicate that reductions obtained in practice are about this value, or less.

From this discussion it appears that the use of sound-absorbing material can be very effective for sounds in tubes and ducts. Modest effects can be obtained in enclosures which initially have very little absorption. As a means of reducing machinery noise, sound-absorbing material is of very little value unless it can be applied under the favorable conditions just outlined.

(5) *Mechanical Damping Resistance* – Mechanical damping resistance is the result of friction between parts, or it occurs internally in materials when they vibrate. The latter is called hysteresis loss. The energy required to overcome this friction appears as heat. In most materials this resistance is small but, in viscous materials, such as tar-like substances, rubber, fluids, and sometimes gases, it can be used to obtain mechanical resistance. As mentioned previously, motions are determined by the mobility of parts, and by the driving forces available. In most instances, there is ample force available to overcome the friction which accompanies the extremely low velocities of audio-frequency vibration. (As stated previously, the velocities usually are measured in thousandths of an inch per second, because of the very small displacements). There are, however, two special cases in which damping is of consequence.

Resonance – The phenomenon of resonance is too well known to require discussion here. When mass-stiffness combinations are such that one of the natural frequencies of the system is close to the frequency of a driving force, large movement results. Increased damping greatly reduces the magnitude of such resonance peaks. Demonstrations of this striking effect are so well known that one instinctively thinks of this method as a powerful means of quieting. Certainly this means is effective in cases of resonance, but instances where it can be applied to machinery quieting do not occur nearly as frequently as might be expected. In fact, the author has been looking for a dozen years for a problem to which this neat solution might be applied but, to date, no such problem has occurred within his experience. Perhaps one reason for this fact is that resonances are observed so easily and so obviously treated that these cases were treated long before the job reached the author. Another important factor appears to be the extreme complexity of audio-frequency vibrations. When the surface of a machine vibrates in many segments instead of as a whole, which is the practical case, the probability of some part coming into resonance and producing a large increase in the total sound is quite slight. On the whole, it appears that quieting through the use of mechanical damping resistance occupies a very minor place in practice.

Transient Vibrations – If an oscillating system is given a displacement and allowed to vibrate of its own accord, the so-called transient dies away much faster if there is more damping resistance. Usually such transients are not nearly as common as continuous driving during which damping resistance plays no part. If one pounds on an automobile panel with his fist, the resulting vibration dies away more quickly if the panel has been treated by the application of damping material. On the other hand, it appears that on the road, the driving forces are essentially continuous, and little difference is noted whether the panels are damped or not.

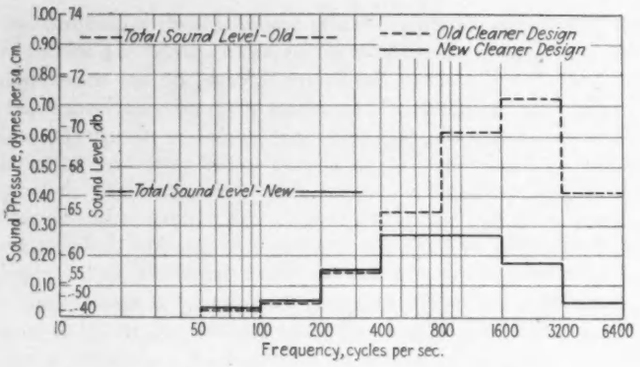


Fig. 9 – Noise Measurements on Vacuum Cleaner

In summary, it is found that mechanical damping resistance can reduce effectively only resonances and transients. If these happen to be the troublesome types of vibrations, damping is useful. Otherwise, it is of little consequence.

(6) *Greater Accuracy of Parts* – Machine parts which are loose, badly out of balance, or poorly built are inclined to knock and rattle and, in such cases, worthwhile reductions can be obtained by tightening, balancing, and better workmanship. On the other hand, reduction of production limits and tolerances is by no means a cure-all for noise. It may be a very costly manufacturing job to reduce a tolerance from 0.001 to 0.0002 in. and, in many cases, it may achieve no quieting whatever. In fact, it is quite common experience that special tool-room jobs where all tolerances are exceptionally small and fits unusually close, may make considerably more noise than units built on ordinary production.

This condition does not seem unreasonable when the dimensions of the acoustical movements are considered. The particle displacement in the sound of 80 db. is approximately 0.00003 in. at 100 cycles per sec. and 0.000003 in. at 1,000 cycles per sec. Since the displacements of the vibrating machine parts are of the same order of magnitude, they are conveniently measured in micro-inches (millionths of an inch), the dimensions just listed being 30 and 3 micro-inches, respectively. The minute size of these deflections makes it easier to visualize the previously mentioned complex vibrating patterns of stiff and heavy parts. It does not seem unreasonable that work on dimensions of the order of 0.001 in. (1000 micro-inches) would have little influence on vibrations which are of the order of 10 micro-inches. In fact, it seems surprising that anything at all can be done with such small vibrations.

The practical approach in connection with greater accuracy of parts is to view this method as a last resort to be attempted only in special cases where definite evidence has been obtained that such improvements in accuracy will produce the desired results at reasonable cost. The author has encountered few such instances.

(7) *Shift of Frequencies to Less Important Ranges* – It is often suggested that quieting can be obtained by shifting frequencies "beyond audibility." In cases of resonance it is often effective to shift frequencies to avoid resonance, but as stated before, resonances are not very common in machinery noise. It is also advantageous if frequencies can be shifted below the low frequency cutoff described under radiation resistance. It is also true that higher frequencies are more objectionable to most people than lower frequencies, and shifts to lower frequencies are desirable, if possible. It is also a fact that, for very faint sounds, the ear is less sensitive to sounds of low frequency than to frequencies in the neighborhood of 1000 to 2000 cycles per sec., and quieting can be obtained through application of this fact.

In most instances, however, the factors listed previously are of minor consequence. For sounds of ordinary level, the ear is about equally sensitive for all frequencies from 50 cycles to 10,000 cycles per sec., a range of nearly 8 octaves. It is manifestly impractical to change speeds by 100 to 1, or any such factor. Even if this change were done, probably new components would be introduced.

In general, recent tendencies have been toward higher speeds in all types of machinery, and this increase tends to increase noise. The possibility of operating at speeds outside the audio-frequency range is very remote in most cases.

Summary of Quieting Methods – A review of these quieting methods shows that, once a noisy machine has been built, about the only cure is to mount it resiliently and to build an enclosure around it. Apparently there is no magic fluid to pour over it, nor no material which can be applied to soak up the noise.

If it is not feasible to enclose it, the problem then becomes one of reducing the noise at the source, or in keeping it from being transmitted to the surfaces which radiate it. These questions finally get back to a consideration of mobility-force products, both in general and in detail. This method may not sound very attractive, but success has been attained by this attack.

We have found on many jobs that the most effective quieting means are inherently simple, and as obvious as A, B, C, *once they are understood*. The complexities of noise reduction exist in the determination of the fundamental mechanisms of noise production, in individual cases.

Section 3 – Examples of Practical Quieting

There is a widely used procedure in attacking noise problems which can be outlined as follows:

(1) Noisy machinery is designed and built.

(2) A brief listen shows that they are noisy, and so little time is spent in examining them. ("Why spend time measuring noisy units? We already know that they are noisy.")

(3) Search is made for some material which can be applied to reduce the noise, and changes are made on the basis of various theories as to the cause of the noise. Such "cut-and-try" is often prolonged to great length.

This method seems logical enough, and the author has no objection to it as long as it produces results.

Many noise problems have been solved by this method, and the automobile has been brought to its present state of perfection chiefly in this way. Eventually, however, the point is reached where this method fails to yield results, and vast amounts of time and appropriation can be spent with very little result to show.

A few years ago, it became apparent that there was a definite need for a more powerful means of attack on these problems which had reached a dead end by ordinary methods. This new method of solving problems which fail to respond to ordinary methods has been developed and used on actual jobs. The results have been most satisfying to our clients, and several of these jobs are described in this section.

The new technique is essentially a reversal of the ordinary method just outlined. It can be outlined as follows:

(1) Noisy machinery is examined to determine which noises are important, and how they are produced.

(2) Quieting methods are devised on the basis of this information.

(3) Quiet machinery is built initially. Usually, it happens that quiet machinery costs no more to build than noisy, and sometimes the quieter machinery is cheaper to build.

In the past three years, the author and his associates have had an opportunity to apply this new method to three noise problems. In each instance the job was a difficult one of long standing and, in each case, the new method yielded a practical solution. Quieting consistent with the criteria outlined

in this paper was obtained, namely, reductions of 5 to 10 db. or more. Not only were these reductions obtained in the laboratory, but they are being duplicated on actual production. These problems are outlined as follows:

(1) *Refrigerator Compressor* – Noise always has been a problem in compressors for domestic refrigerators, and production and engineering departments have worked continually on quieting. Since the first units, great strides have been made but, on the unit here described, the point was finally reached where continued work and redoubled efforts achieved no more results. Many, many experiments were tried, and all parts were changed, but it appeared that the noise was there just the same.

The manufacturer set the following conditions for the job:

(1) The quieting must be appreciable to the ear.

(2) The efficiency of the unit must not be reduced.

(3) The head and suction pressures must not be changed.

(4) The general style of the unit must not be changed.

(5) The dimensions and weight of the unit must not be affected.

(6) Expensive materials or manufacturing operations must be avoided.

(7) An increase in the cost of production of more than a few cents would be prohibitive.

(8) The quieting means used must be permanent, and not affected by the refrigerant.

(9) The speed of the unit must not be changed.

(10) The quieting means must be applicable to production.

The job was undertaken on the basis of meeting the ten conditions just listed. Manufacturing tolerances already were as close as practical. Every part of the unit had been subjected to intensive work over several years.

The job was attacked by devising special means for analyzing the sound of noisy units, and for determining which parts moved to cause these noises, and how these parts were driven. When the underlying mechanism of noise production was determined, the principles listed previously were applied to inhibit the production and to keep the noise from reaching the parts from which it was radiated. The objectionable noise was reduced 10 to 15 db., which practically eliminated it, and the overall noise was reduced considerably. A patent is pending on the means devised.

The quieting means met all of the foregoing requirements. The efficiency and operation of the unit were not affected in the least, no new materials were used, and the new machining operation required was very simple and involved only ordinary accuracy. The method was placed in production within a few weeks without the necessity of redesign or new types of machinery, and has been in use for over two years. The increase in cost of production was considerably less than 1 cent per unit.



Fig. 10 – Old and New Vacuum-Cleaner Designs

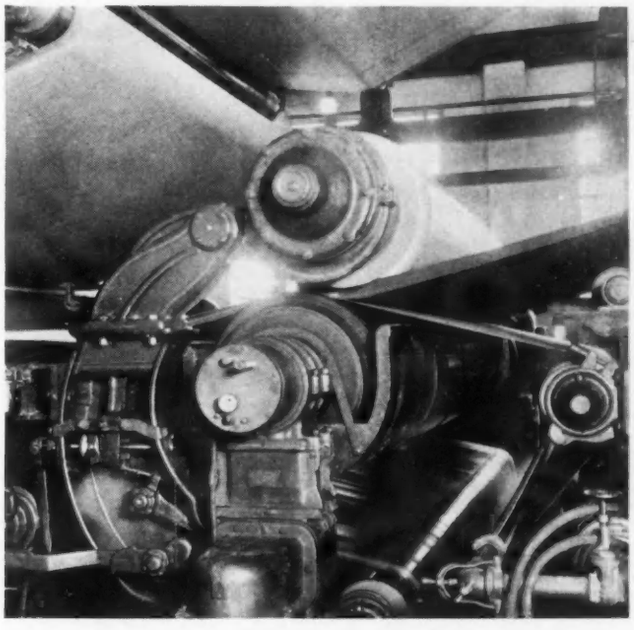


Fig. 11 – Suction Roll on Paper Machine

(2) *Vacuum Cleaner* – Mechanically, a suction cleaner consists of an intake nozzle, a blower, a motor, and a bag. Each of these elements is essential to the operation of the conventional cleaner, and each has been subjected to continual efforts at quieting by practically all manufacturers of such equipment. Most cleaners operating in a typical living room produce sound levels well below those found in present-day automobiles, but nevertheless, noise is an important objection.

The builder of the cleaner here described outlined the following requirements for a practical job of quieting:

(1) The new design must sound appreciably quieter to the ear.

(2) The difference between the old and the new designs must be apparent readily to anyone by ordinary listening, even when considerable time intervals elapse between comparisons.

(3) The new design must not reduce the cleaning efficiency in any way.

(4) The operation of the unit must not be changed.

(5) The power requirements for the unit must not be increased.

(6) The weight of the unit must not be increased appreciably.

(7) The size of the unit must not be increased appreciably.

(8) The cost of the unit must not be increased appreciably.

(9) The general appearance of the unit should not be affected.

(10) The essential manufacturing methods must not be changed, so that the new unit could be produced economically in the same shop.

The motor represented the results of a vast amount of work to obtain a smooth, compact, and efficient unit. Slower speed would have produced quieting, but this reduction was impossible if weight and cleaning efficiency were to be maintained. The blower unit, including nozzle, fan, housing, and outlet, had been subjected to long continued experimentation, and present designs represented the best available compromise on

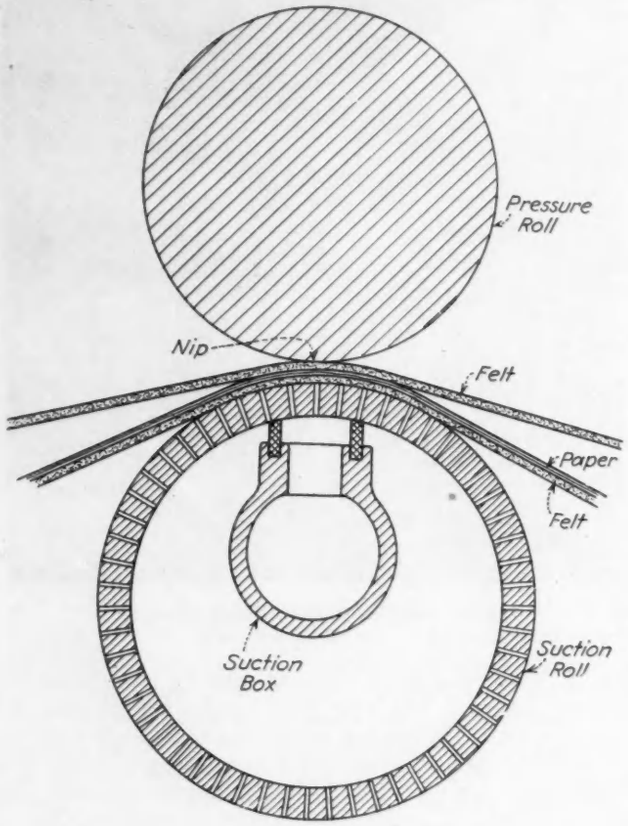


Fig. 12 – Section Through Suction Roll

efficiency, size, weight, and quietness. It appeared that the only part of the unit which could be changed or omitted was the noise itself.

The job was undertaken on the basis of meeting the ten requirements listed previously. The results are shown in Fig. 9. The sound level of the new design is 7 or 8 db. lower than the original, and the quality of the sound has been improved greatly by reduction of the noise at the higher frequencies. Everyone who has heard the two cleaners agrees that there is a marked difference. The new design is slightly more efficient than the old, and the cleaning efficiency and general appearance are the same. See Fig. 10. The size and weight are practically identical. The increase in production cost was slight, and the company took the opportunity to make other improvements in design. The new unit is made in the same shop and with the same personnel. But – the noise is down.

This improvement was made by direct application of the principles listed previously, based on laboratory studies to determine how the various noises were produced and how they reached the surrounding air. Patent applications have been filed on the methods used, and the new unit was placed in production last summer, and is now in satisfactory general use.[13]

(3) *Paper Machine* – Fig. 11 shows a photograph of a machine used for making paper. Essential elements of such machines are the so-called suction rolls which are used to remove the water from the stock as it passes through the machine. Fig. 12 shows a diagram of such a roll. The paper is carried between felts which pass between the "nip" of the two rolls shown. Considerable mechanical pressure is applied, and the operation is improved materially by the use of "suction." The lower roll is hollow, and the shell is perforated with a large number of ⅛-in. holes. A stationary "suction box" is mounted within the roll, and the interior of the box evacuated by suitable vacuum pump. Vacuum of 10 to 20 in. hg. is carried, and the drying action is much enhanced thereby. Unfortunately, however, the construction is simply a siren with a 100-hp. motor attached, and the resulting noise is terrific. Most paper machines employ several of these suction rolls.

The conditions set for this job were as follows:

(1) About 20 db. reduction of sound level necessary for satisfactory conditions.

(2) Suction must be maintained.

(3) A 20 per cent increase in speed must be accommodated.

(4) Nothing must touch the felts, as this contact would cause undue wear.

(5) The quieting means must be impervious to water as the machine operates under a continual spray.

(6) The quieting means must not interfere with the operation of the machine or the convenience of making adjustments.

(7) The size and shape of the suction box must not be changed materially.

(8) The quieting means should be applicable to existing machines without essential change.

(9) The quieting means must not become clogged with loose bits of stock, and must operate 24 hr. a day for months without attention.

(10) No changes must be made in the wrap of felts, roll positions, and so on, and the method must be applicable to a wide range of speeds, suctions, and roll adjustments.

This problem had received considerable attention for several years, and changes had been made in all of the essential parts of the roll assembly without obtaining the desired reduction.

The job was undertaken on the basis of meeting the ten requirements just listed. The resulting installation has been operating in production for several months. The results are shown in Fig. 13, which shows measurements taken on an actual machine operating in a mill. Everyone who has heard the new installation has been much impressed with the reduction. This development permitted the elimination of partitions and other expensive and inconvenient means for acoustic control by reducing the noise at the source.

This development was obtained on the basis of work on an experimental machine in the laboratory to determine the mechanism by which the noise was produced, and the paths by which it was radiated to the surrounding regions. With this material at hand, the principles listed previously were applied

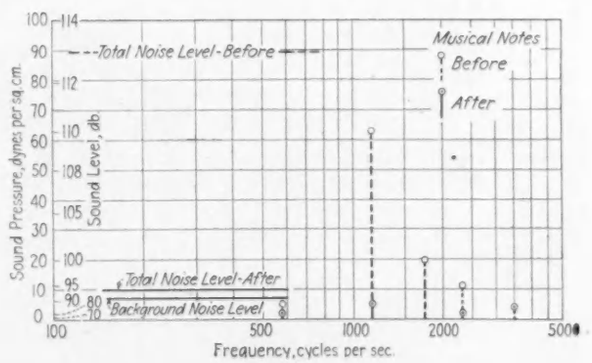


Fig. 13 – Noise Measurements on Paper Machine Operating in Mill

[13] This material was released through the courtesy of the Regina Corp., Rahway, N. J.

to reduce the noise at the source. A patent application is pending on this development.[14]

Conclusion – As outlined previously, the new technique is of great advantage in obtaining practical results on difficult problems which have failed to respond to other attack. A record of three successes on three difficult jobs in three starts appears to be conclusive proof of the effectiveness of the method. To date, the opportunity to apply it to automobiles has not occurred, but there is every reason to believe that it will be just as effective for automobiles as it has been for other machinery.

Discussion

Explains Different Method of Attack

– **Stephen J. Zand**
Sperry Gyroscope Co., Inc.

EXPRESSING any new technical unit by physiological reactions of the human body is a very good but a very old idea. Anyone has seen bath thermometers marked, at 70 deg. fahr., "cool," and, at 80 deg. fahr., "warm," and so on. Several years ago, when the public began to object to excessive vibrations on airplanes, we made up a human-reaction scale and classified vibration phenomena by such expressions as "vibrations which can be seen, felt if touched by a gloved hand, and so on." In the early stages of acoustical work on airplanes, now six years old, we defined noise levels by comparing them to the understandability of speech at predetermined distances. Thus a noise level of 90 db. was compared to a conversation level of "shouting at 3-ft. distance," whereas 30 db. was considered as a "whisper at 12 ft., still intelligible." Gradually, as we become accustomed to new terms and units, we are able to drop consciously the physiological reactions to the unit in question; but it is well known that, subconsciously, the bodily reaction to the physical phenomena always is taken into account.

Some additional remarks may be of value and will broaden Section 2 of Dr. Abbott's paper. From the test it would appear that absorption of sound by porous, flexible materials has only limited uses, and that the only way to attenuate a disturbance properly is by the application of very heavy homogeneous materials wrapped around the noise source. Upon analysis of the whole problem of noise reduction in closed rooms, such as airplane cabins, buses, trains, and so on, we can conclude that the well-trained designer will have to work with both principles in order to get desirable acoustical characteristics with a minimum of weight.

If we denote by i_1 the intensity outside, and by i_2 the intensity which is transmitted into the cabin or room, then the ratio of $i_1:i_2$ will give us the reduction of the noise accomplished by the passage of sound waves into the surrounding parts of the body, such as panels, walls, bulkheads, doors, windows, ceilings, and so on. But the intensity of noise transmitted into a room is equal to the rate of flow of sound into the confined space divided by the total absorption present in that room. Or, expressing it mathematically, if ΣT represents the sum of all transmittances, and if ΣA is the total absorption, then:

$$\text{The noise reduction in decibels is } 10 \log \frac{\Sigma A}{\Sigma T}$$

The transmittance of component parts can be obtained from fairly simple laboratory tests. The absorption is expressed in units known as "sabines," which is the coefficient of absorption times area in square feet.

The question which interests the airplane designer is weight economy, and this formula invariably will give the answer to the problem of which line of approach to choose – greater transmission loss, or greater absorption.

Let us illustrate this thought by an example taken from an actual airplane. In a given design we had available 60 lb. of extra weight, and the question arose of what to do with it. It was suggested that we use a construction calling for double windows, which would decrease transmittance; or, we could add absorption by treating certain cabin parts with suitable materials. The original figure for transmittance of this cabin was 0.835. The original absorption was 390 sabines, hence the reduction in decibels was $10 \log \frac{390}{0.835} = 26.69$ db. Double windows would decrease the figure of 0.835 to 0.794, hence the total reduction would become $10 \log \frac{390}{0.794} = 26.91$ db. or a gain of 0.22 db. Materials having absorption coefficients of 0.8 and weighing 0.2 lb. per sq. ft., although not common, can be procured. Thus we can obtain an addition of 240 sabines by using 300 sq. ft. (60 lb.) of a material having an absorption coefficient of 0.8. In that case the noise reduction will be $10 \log \frac{390 + 240}{0.835} = 28.76$ db., or a decrease of the cabin noise level of about 2 db. It is impossible to generalize; different conditions may necessitate entirely different methods. Since the mass of a wall offers a reaction to the sound wave similar to the magnetic reaction of a self-inductance in an electrical circuit, obviously then, the mass of a partition ought to be the principal factor in determining the insulating value of the partition. But it is possible to increase the transmission loss through a solid partition by combining properly the two known principles of sound insulation, that is, by inertial losses in rigid partitions and viscous losses in porous flexible material in an air space between two rigid partitions.

Among the newly developed systems of sound absorption is one which has received very little attention from industries outside of the airplane field. Through the joint efforts of J. S. Parkinson, of the Johns-Manville Corp., and our staff, we were able to develop a combination of two materials which weigh very little and show remarkable sound absorption in low frequencies.

The construction consists of a highly stretched diaphragm which can be made of any material impervious to air, behind which felt is located, the thickness of the felt varying with the amount of absorption wanted and, of course, with the limitations of weight. Neither material alone will give the desired efficiency. The membrane by itself will have a certain absorption at its resonant frequency, and this absorption will be sharply critical with frequency. The application of the felt, we have found, causes the absorption to be spread well across the frequency range, due partly to the damping effect of the felt fibers and partly to the added mass reactance and what might be called an inertance reactance. It will be apparent that, for full effectiveness, this construction should be spaced from the wall or surface behind. Our experiments indicate that, the greater this spacing, the more effective the low-frequency absorption. Our experiments likewise show that, the heavier the felt, the more effective the low-frequency absorption. For practical purposes, ¼-in. to ½-in. felt will be satisfactory. The weight of this construction varies from 0.15 lb. per sq. ft. to 0.3 lb. per sq. ft.

Somebody with a sense of humor called this system of absorption "diaphragmatic dissipation." If we were to advertise our system under this name, I am sure the automotive industry, which adores such mysterious expressions, would probably adopt it – not for the sake of its acoustical efficiency, but for the purpose of further mystifying the consumer.

In closing, it is hoped that Dr. Abbott's efforts toward popularizing the science of acoustics will continue. His efforts already have shown results on present-day automobiles as, with the increase of automobiles employed and people driving in them, noise, for the sake of our nerves, has to be eliminated.

Outlines Problems Awaiting Solution

– **W. A. Wagstaff**
Norge Division, Borg-Warner Corp.

DR. ABBOTT has practically confined his discussion to some of the methods employed by the acoustical engineer in actually accomplishing an appreciable reduction in some objectionable noise. In my opinion, this phase of noise correction is rather a simple part of the problem. When a noise problem has been analyzed correctly and accurately, most any engineer can foresee at once several practical methods of effecting a correction, and the only factor limiting the choice is one of cost or ease of application. From my experience with noise reduction projects, it appears that the most important and vital part of the problem is the analysis, or the obtaining of a complete description of the noise.

To obtain a correct analysis of a noise problem, however, some sort of instrumentation is frequently necessary. It is in connection with this phase of the noise-reduction problem that I wish to offer a few comments.

As a rule the type of instrumentation used in noise analysis is not only very expensive but it requires the services of an expert to operate them. The unfortunate part of this "expert" business, from an organization and management viewpoint, is that acoustical experts are primarily hard to find and, if a satisfactory type of vulnerable embryonic engineer is discovered, it requires several years of additional study and practical experience to develop such a man into a full-fledged expert. Suppose a manufacturing concern is fortunate enough to have in its organization such an acoustical expert, and further suppose this manufacturing concern has made available to its acoustical expert a number of expensive noise-analyzing instruments. I still question whether the noise problems of the engineering department of such a corporation will be satisfactorily, or entirely, eliminated. The time required, even with the most expensive instrumentation and the best possible acoustical expert and analyzer, to describe the noise completely, is generally far too great.

Assuming that the expert eventually is able to improve the noise prob-

[14] This material was released through the courtesy of the Beloit Iron Works, Beloit, Wis.

lem, do we as a business concern have any assurance that the noise reduction as made will be satisfactory for the market at the time the corrections are incorporated in the finished product? If the answer is "yes," then do we have any assurance that the noise level, as now corrected, will be satisfactory two years, or three years, from now so that we can realize a just profit on our expenditure in accomplishing this noise reduction? In answering this second question, we must keep in mind the rapid and sometimes radical changes in design, which are necessary each year to improve our product. Should it require even six months of constant effort on the part of the acoustical expert finally to effect a correction for one of the noises of our product as it now exists, I am afraid we will discover that, by the time such a correction becomes part of our production, other changes will bring to light another noise source which, in itself, must be studied thoroughly and painstakingly. If this is not the case, then the public consciousness of the noise level of our product will change in a year's time and, even though the noise level as it originally existed has been reduced, the present noise level is now considered by the public as too loud. The inability of the human ear to retain for any length of time the value of any noise is only too well known, and our fast-moving civilization demands improvements each year, not just the correction of the past year's noise faults.

Acoustical scientists and engineers – like Dr. Abbott – have made a distinct contribution to industry in exploring the possibilities of noise correction. However I feel that, before their work can be accepted fully, it will be necessary for them to direct their efforts more definitely toward the solution of the following problems:

(1) Effect an agreement among themselves, as to the many theories, terms, units of measurement, and descriptive items of acoustical science so that the production of acoustical engineers in our universities and laboratories can be made as simple, positive, and economical as is the case at present with mechanical or electrical engineers.

(2) Simplify the noise-analyzing instruments so that all such instruments, no matter who the manufacturer or vendor, will always give identical answers when identical noises or the same noise are being measured. Further, these instruments should be made so practical that their use in industry is not limited to the laboratory. It should be possible for an ordinary service, inspection, or production man to use them intelligently and rapidly in connection with each day's production or service problems. Their initial cost, as well as their upkeep, should certainly be reduced considerably.

(3) Further improve the technique for analyzing noise problems so that a complete analysis and correction of any noise problem can be made in a day or two – rather than in several months.

Until these improvements are made, I am afraid the place of thorough acoustical treatment of production noise problems will continue to be filled by the so-called "guess" system now employed so generally.

Differs on Effects of Damping Body Panels

– J. S. Parkinson

Johns-Manville Corp.

THERE are certain points on which our data show some disagreement with those of Dr. Abbott.

In particular, Dr. Abbott says that "tests indicate that the panels of a car are driven by essentially constant-displacement forces. Accordingly, it probably would be impossible to detect the difference between panels which were at the extremes of the laboratory results with the other type of driving force. Much money can be spent making tests under conditions which are of little or no practical significance." This appears to be a somewhat unfortunate generalization. There may be circumstances where the panels are not driven by constant-displacement forces and where damping materials do perform a distinct function. In general, we agree that it is not possible to quiet an automobile by the use of damping materials. However, in so far as the noise given off by body panels is excited by shock excitation, damping will serve to reduce both the amplitude and the duration of such sounds. It also will have the advantage of removing almost entirely the high-frequency components which are most annoying. The most obvious example of this type of quieting is the difference which is observed between the noise given off by treated and untreated doors on closing.

Certain types of road noise, such as the noises produced by crossing expansion joints in concrete, belong in the category of shock excitation. For noises of this type, damping materials can be expected to be effective.

It likewise may be shown that, even for constant-displacement forces applied to the edge of the panel, the vibrations of the center of this panel will be affected materially by the use of damping material. It is presumed that, in making the statement that "the panels of a car are driven by essentially constant-displacement forces," Dr. Abbott is assuming that the panel is essentially rigid and that all parts of the panel are vibrating with constant amplitude and phase. Our laboratory tests show that a 24 x 30-in. panel of body metal clamped in an iron frame and driven from the edge by a small electric motor gives off considerably less energy when treated with an effective damping material. The reduction in overall energy in this test was 4 db. In certain bands the loss was negligible. In others it was as high as 23 db. This example further illustrates the point that an important part of the effectiveness of damping materials is in removing certain annoying components of the noise.

Most of those concerned in this type of work have observed that, if one of the front wheels of an automobile be struck with a hammer, certain of the natural periods are observable readily within the car. These same periods will appear as the speed of the car is increased to pass through the resonances. Dr. Abbott himself points out that the vibrating systems encountered in practice are extremely complex and present a large number of resonant frequencies. It seems obvious, therefore, that the effectiveness of a damping material in reducing these resonances, particularly in the higher frequency ranges, will be a distinct advantage in producing overall quieting.

To summarize, it is my belief that the vibrations in body panels do not always represent constant-displacement forces; first, because many of these vibrations are the result of shock excitations and, second, because it may be shown that, even when a panel is vibrated from the edge, the amplitude of the center of the panel may be affected readily by damping treatment. I further believe that there are a variety of speeds at which panel resonances respond to road or motor driving frequencies. In all of these cases, it seems obvious that damping material has an important function.

There is one other point in Dr. Abbott's discussion which we believe should be amplified. This has to do with the effect of sound-absorbing material. The statement is made at one point that "the application of additional sound-absorbent material hardly can increase the average absorption coefficient by more than a factor of 2, which would reduce the sound pressure by 30 per cent or 3 db." This statement may be true, providing that the type of absorbent material used is one which has high-frequency efficiency only, and providing also that the quantity of absorbent material which can be introduced is not large in comparison with that which is already present. Reductions greater than 3 db. have been measured on several occasions in passenger cars where the absorbent material was designed to take out a specific band of frequencies which the normal construction of the car did not absorb adequately. Likewise, in buses where a large area of absorbent material of a suitable frequency characteristic may be used, considerably greater reductions have been obtained.

I am happy to note that Dr. Abbott points out the fallacy of using a sound-absorbent material as a means of enclosure. I do not altogether agree, however, with his implication that it is not economical to obtain reductions by lining the enclosure on the inside with sound-absorbent material. In the first place, the example given assumes an enclosing shell with an absorption coefficient of 0.1. The normal materials used for enclosures such as wood, steel, asbestos board, and so on, have absorption coefficients of the order of 0.02 or 0.03. In consequence, the substitution of a lining material having a coefficient of 0.5 as in his example, would increase the average coefficient by approximately 20 times and effect a difference in level of 13 db. Thus, if the original enclosure had the stated attenuation of 20 db., the net effect would have been to reduce the level 20 — 13 db. or only 7 db. To counteract this loss by the method suggested of increasing the transmission loss of the shell would require a shell having an attenuation of 33 db. The original shell having a transmission loss of 20 db. would presumably have weighed about 0.9 lb. per sq. ft. This value is based on the common relation between transmission loss and the logarithm of the weight for homogeneous materials. To have increased this transmission loss to 33 db. would have involved using a material having a weight of 6 lb. per sq. ft., an increase of about 7 times in weight. In most cases when the required reduction in level makes it necessary to go to heavy types of enclosures, it is not only more economical but also more practical to rely on sound-absorbent materials.

Finds Quick Aural Comparisons Useful

– Peter Loomis

Norge Division, Borg-Warner Corp.

IT seems to me that especial emphasis should be placed on the difficulty in working with constant-displacement forces, the correct use of absorbing materials, and the proper use of ear and instruments.

In connection with the quick aural comparisons which Dr. Abbott mentioned, we have found this an especially useful technique for testing theoretical solutions. In some of our work on inaccessible sources we have been able to ascertain and demonstrate the causes and cures of very small but irritating noises, which were especially elusive because of irregular occurrence. By means of magnetic or hydraulic forces, externally controlled, very definite and convincing audible differences were obtained where instrumentation was impracticable.

As Dr. Abbott said, he has never seen or heard two acoustically identical machines. If comparisons are to be made between machines in which the improvements are small and variable, a statistical number of samples should be used. This makes the quick-change comparison in one machine very valuable if the set-up is well planned.

Motion Pictures of Engine Flames Correlated with Pressure Cards

By Gerald M. Rassweiler and Lloyd Withrow

Research Laboratories Division, General Motors Corp.

THIS paper represents a continuation of the work with the high-speed motion picture camera described before the Semi-Annual Meeting of the Society, June, 1936.[1]

The experimental observations consist of pictures showing successive positions of the flame at intervals of 2.4 crankshaft deg. during single explosions, and pressure-time records of the same explosions.

A method finally is described for sorting out the pressure changes due to combustion from an observed pressure card. When the pressure changes resulting from combustion are summed and put on a percentage basis, it is found that the per cent of pressure rise due to combustion is approximately equal to the per cent of charge burned (by weight) at the corresponding instants in the combustion period.

A PREVIOUS paper[1] describes a high-speed motion picture camera which photographs pictures of engine flames at rates as high as 5000 frames per sec. It was pointed out in this paper that these pictures yield unusually complete information about the flame propagation in the engine for the following reasons:

(1) The pictures are recorded through a window which allows an unobstructed view of the entire combustion space.

(2) The progress of the flame is followed throughout the combustion period in single explosions.

(3) The rate at which the pictures are taken is such that about 20 observations of the form and position of the flame may be made during flame propagation in a single explosion.

Since the power output, knock, and roughness in engines appear to be related in some way to the rates of pressure rise and to the magnitudes of maximum pressures attained, it is highly desirable to interpret the flame pictures in terms of pressure changes produced within the engine. Considerable effort has, therefore, been made to obtain accurate pressure data from the engine cylinder during the photographing of the flame pictures. A description of the methods developed to measure these pressures constitutes the major portion of the experimental work described herein.

In the analysis of the data it has been found convenient to express the progress of combustion both in terms of inflamed volume of charge and in terms of burned fraction of the weight (or mass) of charge. Although the latter quantity cannot be obtained directly from the flame pictures, its use aids materially in understanding the significance of the relationships. In studying the relationships between inflamed volume, inflamed fraction of the mass, and pressure, an effort has been made to keep the analysis direct and straightforward, to avoid introducing assumptions which have not been tested experimentally, and to determine experimentally the constants used in the equations.

Engine Conditions

All of the data in this paper were obtained under one set of engine conditions as follows:

Speed	900 r.p.m.
Spark Advance	25 deg.
Fuel	Iso-octane (2,2,4 trimethylpentane)
Air-fuel Ratio	13:1
Compression Ratio	4.6:1
Throttle Setting	Full open

The Flame Pictures

Fig. 1 presents a typical sequence of 24 consecutive frames from a motion picture of a single engine explosion. Beneath each frame in Fig. 1 is the crank angle at which the exposure of that frame ended, the minus signs denoting angles before top dead-center and the plus signs indicating angles after top dead-center. The angular interval between the end of the exposure of one frame and the end of the exposure of the next was 2.4 crankshaft deg., the duration of each exposure being 2.2 deg. The forward edge of the flame front as recorded in each frame was the position occupied at the end of exposure of that frame; that is, at the crank angle indicated beneath the frame. Thus, for example, Frame 13 (the first picture in the third horizontal row) shows that, at +0.8 deg., or approximately top dead-center, about three-quarters of the combustion space was filled with inflamed charge.

The result of previous studies of engine flames with the sampling valve[2] and the spectrograph[3], indicates that the

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1938.]

[1] See *Industrial and Engineering Chemistry*, June, 1936, pp. 672-677: "High-Speed Motion Pictures of Engine Flames;" also S.A.E. TRANSACTIONS, August, 1936, pp. 297-303: "Slow Motion Shows Knocking and Non-Knocking Explosions," both by Gerald M. Rassweiler and Lloyd Withrow.

[2] See *Industrial and Engineering Chemistry*, September, 1930, pp. 945-951: "Following Combustion in the Gasoline Engine by Chemical Means," by Lloyd Withrow, W. G. Lovell, and T. A. Boyd.

[3] See *Industrial and Engineering Chemistry*, July, 1931, pp. 769-776: "Spectroscopic Studies of Engine Combustion;" also May, 1932, pp. 528-538: "Emission Spectra of Engine Flames;" both by Lloyd Withrow and Gerald M. Rassweiler.

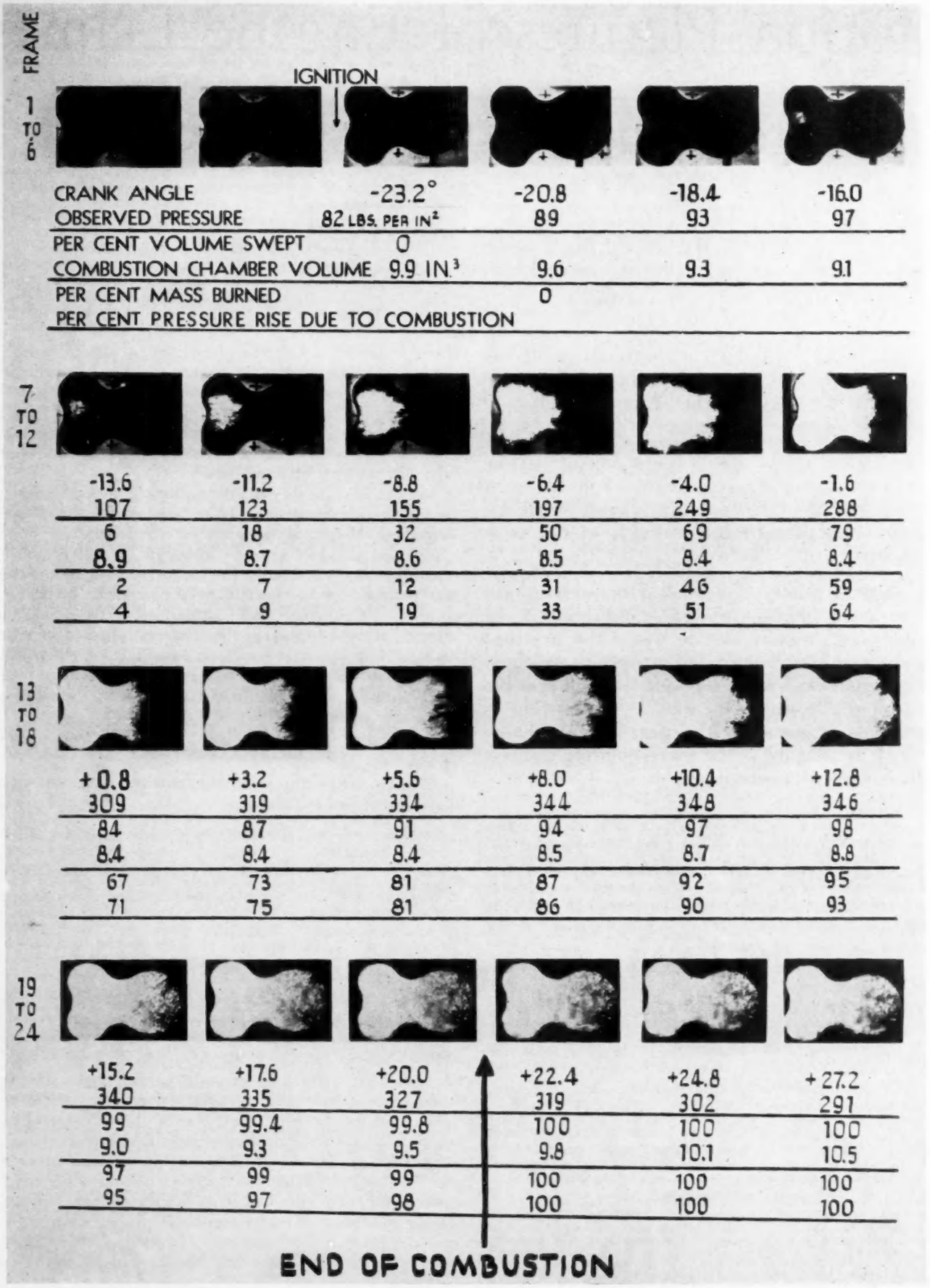


Fig. 1 – Typical Set of Flame Pictures of a Non-Knocking Explosion

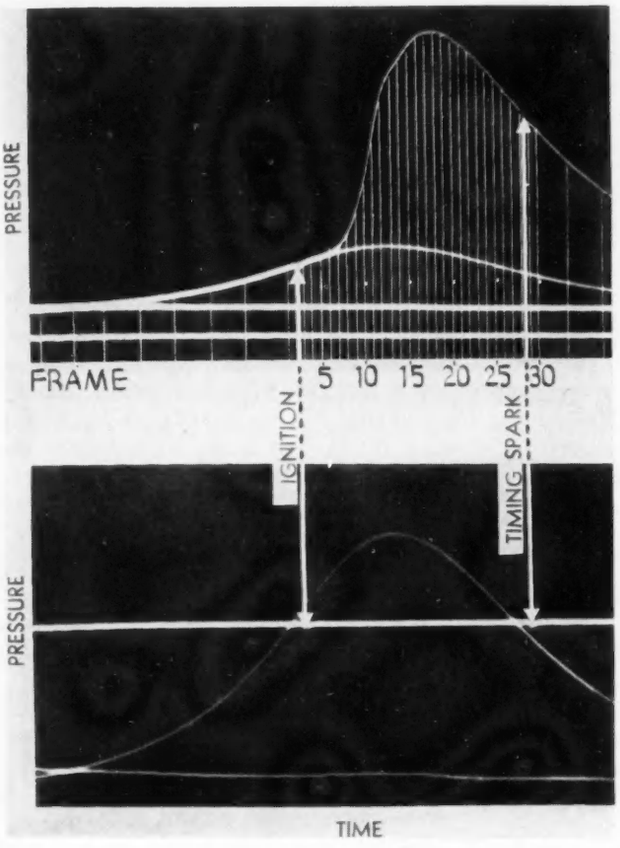


Fig. 2 – Firing-Pressure Card for the Explosion in Fig. 1, Together with Two Firing-Compression Cards

gasoline burns to completion, in so far as the oxygen present and the conditions of temperature and pressure permit, near the forward surface of the inflamed volume of charge. Hence, the inflamed portion, as shown in Frame 13 of Fig. 1, for example, is composed chiefly of burned gas or more exactly of gaseous constituents which are in chemical equilibrium. (The terms "burned" and "inflamed" are used interchangeably in this paper.) The dark portion of charge ahead of the flame in Frame 13 is unburned. The temperatures in the inflamed portion range between 3500 and 4500 deg. fahr., while the probable temperatures of the non-inflamed portions range between 500 and 1000 deg. fahr. This temperature difference results in a several-fold difference in density of the non-inflamed and inflamed portions of the charge at any instant during an engine explosion, which difference must be taken into account in the investigation of the relationship between the flame propagation and the pressure development that is under consideration.

The Pressure Records

A set of pressure records, typical of those obtained when photographing each set of flame pictures, is reproduced in Fig. 2. The records were secured with a carbon-stack indicator[4] used in connection with a Duddell oscillograph. The upper half of Fig. 2 shows a pressure-time record of a single engine explosion recorded simultaneously with the photographing of the sequence of motion pictures reproduced in Fig. 1. Immediately thereafter, two "firing compression cards" were recorded. One of them was superimposed on the firing card with the indicator sensitivity unchanged, whereas the other, shown in the lower half of Fig. 2, was recorded separately using the maximum sensitivity of the indicating apparatus. (The term "firing compression" as used in this paper refers to those conditions prevailing in the engine when the ignition is cut off for a single explosion. The fuel enters the engine and the initial conditions are the same as for a firing card.) Pressure scales and time scales are required to evaluate these three pressure cards.

Establishing the Pressure Scales – The method of establishing a pressure scale for records obtained with the carbon-stack indicator consists essentially in the use of an improved type of balanced-diaphragm indicator to locate a series of pressure levels. The carbon-stack indicator which is mounted in the engine block is not disturbed when a calibration is to be made. In order to mount the balanced-diaphragm indicator in the engine, a solid steel plate is substituted for the quartz window. A $\frac{7}{8}$-in. hole in the plate permits the balanced-diaphragm indicator to be mounted in the roof of the combustion-chamber just above the base of the carbon-stack indicator. Prior to the calibration, the engine is operated for some time under conditions simulating as closely as possible those under which the carbon-stack indicator is to be used for measuring pressures in single engine explosions. The pressure above the balanced diaphragm is then set and held at some arbitrary value which is read on a Bourdon gage. The apparatus is arranged so that, at the instant during each explosion when the pressure in the engine equals the value registered on the gage, the deflection produced by the carbon-stack indicator can be observed. A set of such observations of the deflections corresponding to a series of arbitrary pressures above the balanced diaphragm constitutes a calibration of the carbon-stack indicator.

The circuit for accomplishing the foregoing purpose is shown in Fig. 3. The conventional Wheatstone's bridge circuit (at the right of Fig. 3) usually employed with two-stack indicators is coupled to the balanced-diaphragm indicator circuit (at the left of Fig. 3) by means of a small transformer. When the current in the balanced-diaphragm circuit is either established or interrupted by making or breaking contact between the diaphragm and the high-tension terminal in this indicator, a surge of current is induced in the oscillograph circuit. The effect of this surge on the pressure-time curve produced by the carbon-stack indicator is shown in Fig. 4. The first break in the pressure line in Fig. 4 was produced when the engine pressure built up to 199 lb. per sq. in., which was the pressure maintained on the upper side of the balanced diaphragm. The second break was produced when the pressure in the engine dropped to 199 lb. per sq. in., and the balanced diaphragm opened the circuit. The deflection produced by the carbon-stack indicator corresponding to 199 lb. per sq. in. and with pressures increasing, is the distance from the first break in the pressure-time curve to the upper side of the base line, whereas the corresponding deflection produced with pressures decreasing is the distance from the second break in the pressure-time curve to the lower side of the base line. A series of such deflections, corresponding to various pressures, are read from the screen of the oscillograph by visual observation. Data obtained in this manner for the evaluation of firing pressure-time curves (see Fig. 2, for example) are presented in Fig. 5 where deflections of the oscillograph have been plotted against absolute pressures in lb. per sq. in. The base line (recorded during the intake event and shown in Fig. 4) is selected arbitrarily to represent zero deflection both in calibrating the indicator and in measuring pressures; thus, no correction need be applied for departure of this datum from atmospheric pressure.

Before and after each calibration of the carbon-stack indicator, the calibrating instrument, that is, the balanced-diaphragm indicator, is checked against a third indicator.

[4] See S.A.E. TRANSACTIONS, Vol. 23, 1928, pp. 384-394: "A New Electrical Engine Indicator;" also *The Electrical Journal*, Vol. 27, February, 1930, pp. 87-91: "An Electrical Engine Indicator;" both by E. J. Martin and D. F. Caris.

The purpose of this check is to determine the pressure difference required on the two sides of the diaphragm to deflect it against the contact. Obviously, the greater the pressure required to deflect the diaphragm, either against or away from the high-tension terminal in the indicator, the greater will be the difference at the time the break appears on the pressure line in Fig. 4, between the pressure in the engine and the pressure applied to the top of the diaphragm as read on the Bourdon gage. The pressure difference required to close contact may be measured readily with the indicator cold and removed from the engine, but the value so determined may have no significance when the indicator is operating under engine conditions. To obtain the required information under engine conditions, the balanced-diaphragm indicator is compared with another pressure-measuring instrument which is particularly free from the influence of temperature effects.

Tests on a sampling-valve type of pressure indicator ob tained from the Commercial Engineering Laboratories showed that the time of opening and closing of the valve was not influenced appreciably by the temperature changes accompanying the transition from engine-motoring to engine-firing conditions. The sampling-valve indicator operates most accurately when measuring the maximum compression pressures. Under these conditions, the rate of change of pressure is low, and the maximum pressure is duplicated closely in successive cycles. Therefore, the balanced-diaphragm indicator is checked against the sampling-valve indicator by measuring simultaneously with the two instruments maximum firing-compression pressures. This test is carried out in the following manner: With the two instruments mounted in the steel plate which replaces the window while calibrating, the engine is operated in the normal manner except that a rotary switch driven by the engine opens the ignition circuit every sixth cycle, causing the engine to miss. Other rotary switches are arranged so that the sampling-valve and balanced-diaphragm indicators operate only during the cycles when the engine misses. If the measurements of maximum firing-compression pressure by the two instruments check within a pound or two per square inch, as is usually the case, the balanced-diaphragm indicator is considered to be in satisfactory condition for a calibration.

The merits of this calibration procedure may be summed up as follows:

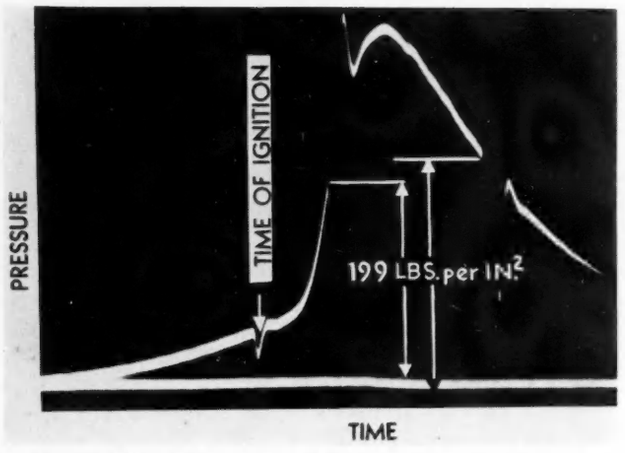


Fig. 4 – Typical Breaks Produced with a Balanced-Diaphragm Indicator on a Pressure Card of a Single Explosion As Obtained with a Carbon-Stack Indicator

(1) The carbon-stack indicator is calibrated under the engine conditions to which it is subjected in service.

(2) In the interval between the calibration and the recording of the pressure records, the indicator is neither removed from the engine nor disturbed in any way.

(3) The number of pressure levels at which deflections are read can be made large, thus reducing the random observational errors.

(4) The accuracy of the calibration depends upon the operation of the balanced-diaphragm indicator which seems to be capable of considerable accuracy for the establishment of given pressure levels in a sequence of successive explosions. By checking this instrument repeatedly against a third instrument in the manner just described, the uncertainties regarding its operation under engine-firing conditions may be minimized.

Even though the carbon-stack indicator is calibrated under engine-firing conditions and without removing it from the engine, it is still desirable to check the constancy of its sensitivity between calibrations. Three tests are applied for this purpose:

(1) Calibrations made at the beginning and end of a day's work should agree.

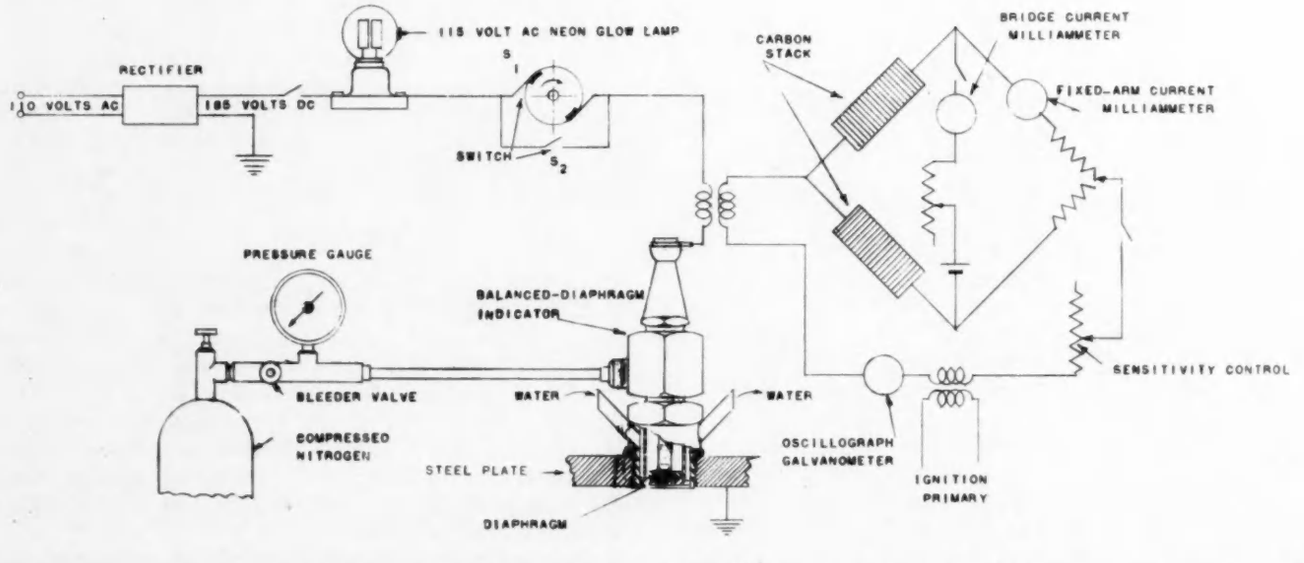


Fig. 3 – Circuit Used for Calibrating Carbon-Stack Indicators Under Engine Firing Conditions

(2) It has been found that changes in sensitivity are accompanied frequently by changes in stack resistance which may be checked as follows: (a) For a constant bridge current with the oscillograph circuit open the "fixed-arm current" (see Fig. 3) will remain constant if the sum of the stack resistances is constant. (b) The engine pressures during the intake event, when the base line is put on the cards, are very nearly repetitive from cycle to cycle; thus, if there is no shift in the base line, the ratio of the resistances of the two stacks for a given pressure is constant.

(3) A third check on the constancy of the indicator sensitivity is the comparison of deflections corresponding to maximum firing-compression pressure as recorded on various cards throughout the day. Since maximum compression pressures show very little change under a given set of engine conditions, a change in the corresponding deflection, as recorded by the indicator, is usually a sign of a change in indicator sensitivity.

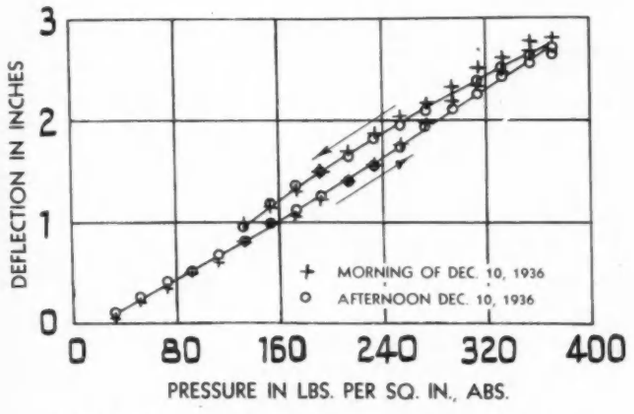


Fig. 5 – Indicator Calibration Curve Obtained Under Engine-Firing Conditions

The Time Scale – The correlation of the pressure measurements and the flame pictures requires an accurate time scale. Here, the problem is to determine the points along the time axis of a pressure record corresponding to the end of the exposure of each frame in the flame picture. Such a scale is shown in Fig. 2, each vertical line representing a frame of the flame picture shown in Fig. 1. The problem of laying out such a scale is simplified to some extent by the fact that both the camera and the oscillograph are driven from the engine. The camera disc is fastened rigidly to one end of the engine crankshaft. The revolving mirror, which spreads out the time axis of the pressure cards in the oscillograph, is coupled through a set of reducing gears and the dynamometer to the other end of the engine crankshaft. Consequently, a time templet can be laid out under static conditions by setting the camera disk at the respective crank angles at which the focal plane shutter cuts off the exposure of each successive frame and then by photographing the spot of light on the oscillograph screen at each of these positions. In determining the engine pressures, each pressure record is superimposed upon the negative of the templet and viewed by transmitted light before an illuminated screen. The templet is first shifted with respect to the pressure record until the flick put on the pressure card at the time of ignition – which time is read from a spark protractor as the record is photographed – coincides with the corresponding angle on the templet and until the horizontal lines (the lower line in Fig. 2), which are photographed with zero current through the oscillograph, are parallel to each other. As a further check on the accuracy of the time correlation, a timing spark is photographed on the flame pictures and registered on the pressure records.

As soon as the templet and pressure record are oriented correctly with respect to each other, the calibration curve shown in Fig. 5 can be used for determining the pressures at the end of exposure of each frame in the flame picture. Such pressure values are listed as the second item of data beneath the frames in Fig. 1.

Volume Swept by Flame

Determination of the Per Cent of the Combustion Space Swept by Flame – Referring again to Fig. 1, it will be noted that the third item beneath each of the flame pictures consists of the per cent volume swept by flame. The procedure for determining these values now will be described.

A series of plaster casts of the combustion space is prepared, each cast representing the clearance volume plus the displaced volume for a particular crank angle corresponding to the end of the exposure of some flame picture. Three views each of three such casts are shown in Fig. 6. The flame picture to be measured is projected in natural size upon the upper face of the corresponding cast (see the left cast in Fig. 7). The position of the forward edge of the flame front is marked as shown on the center cast in Fig. 7; the cast is cut along this line; the two portions are weighed; and, the percentage by weight of each part is determined. These latter values, which are considered equal to the volume per cent, are multiplied by the total volume of the combustion space at the end of exposure of the particular frame under consideration to obtain cubic inches of charge burned and unburned, respectively. (The total volumes at each crank angle constitute the fourth item beneath each frame in Fig. 1.)

This simple method of determining the inflamed volumes from the pictures takes care of three of the principal difficulties encountered, namely: (1) the irregular form of the flame fronts, (2) the variation in depth of the combustion-chamber from one end to the other, and (3) the variation in piston position from one exposure to the next.

On the other hand there are a number of possible sources of error in the values for the inflamed and non-inflamed volumes of charge as determined by the foregoing method. These possible errors will now be enumerated and discussed.

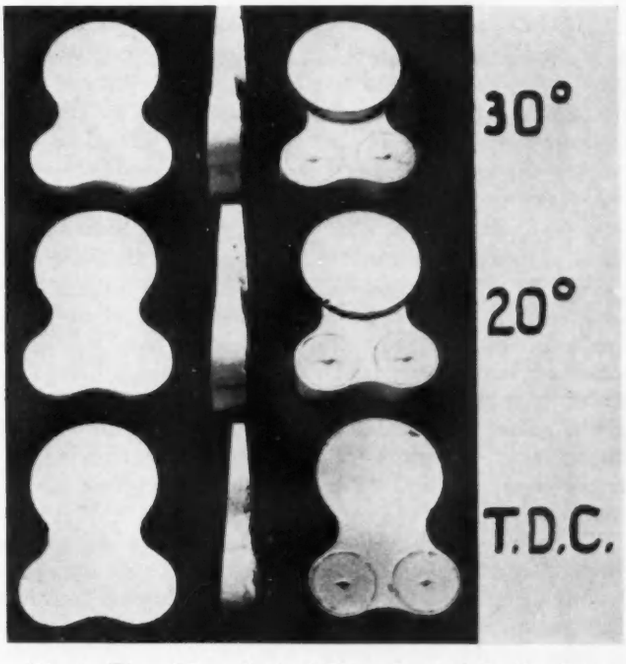


Fig. 6 – Three Views Each of Three Casts of the Combustion Space Showing How the Volume Changes Produced by Piston Motion Were Taken into Account in Determining Inflamed Volumes

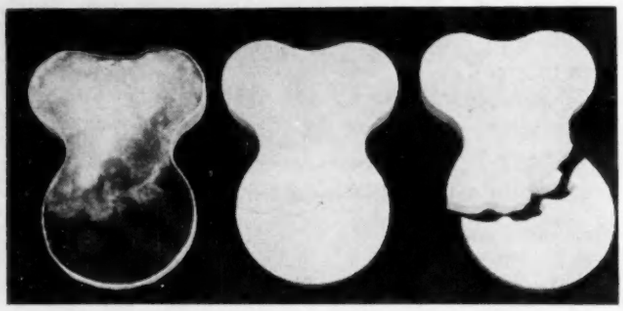


Fig. 7 – Method of Finding Inflamed and Non-Inflamed Volumes from Flame Pictures

(1) Errors due to curvature of the flame front in the vertical plane. Since the flame is viewed only from above, the amount of this error is not known, but previous observations in an engine used for studying absorption spectra indicated that, throughout most of the flame propagation, the curvature of the flames in the vertical plane is not very marked. However, when the flame first starts from the spark-plug and before it touches the top and bottom of the combustion-chamber, the errors introduced by the curvature of the flame front may be quite large.

(2) Errors due to insufficient photographic exposure at the forward edge of the flame. On account of the rapid forward motion of the flame front as the picture is taken and on account of the finite time required to register a photographic image of the flame, the actual position of the flame front, when the exposure ends, may be somewhat ahead of the position registered on the picture. The more intense is the light from the flame, the less this error will be. In photographing the flame pictures shown in this paper, a trace of sodium was introduced into the engine. This addition brightens the flame and makes the image of its forward edge much more definite.

(3) Slight errors might be introduced by non-uniformity of density throughout a cast. A few of the casts have been checked, and it is believed that errors from this source are negligible.

(4) There are undoubtedly random and personal errors introduced when the casts are marked. In some cases the flame front is not defined very well, and there may be some variation in marking the casts between one trial and the next, or between one observer and another. To minimize these errors, two complete sets of casts from each set of flame pictures have been marked independently by two observers; hence, the values of per cent volume swept by flame, as presented in Fig. 1, are the average of two sets of determinations.

In regard to the several sources of error just listed, it is not possible at present to determine the magnitude of the errors from sources (1) and (2). But it should be noted that errors from these two sources tend to neutralize each other. Some idea of the magnitude of the errors from sources (3) and (4) may be gained from the fact that the maximum difference in volume swept at a given frame as obtained by two independent observers for the six explosions described herein is about 7 per cent.

The corresponding values of pressure, inflamed volume, and total combustion-chamber volume for single-engine explosions determined by the methods just outlined constitute the experimental data. Fig. 1 presents such data for one explosion. Similar data for five other explosions are presented in the appendix to this paper (Fig. 22). The next objective is to investigate the relationships existing between the data from the flame pictures and from the pressure records. For this purpose the explosion represented by Figs. 1 and 2 will be used as an illustrative example.

Volume Swept Compared with Pressure

In Fig. 8 the original data obtained directly from the flame pictures and pressure record of the explosion represented in Figs. 1 and 2, are compared graphically. The observed pressure and the per cent volume swept by flame are plotted against crank angle on separate ordinate scales which are arranged so that the two curves are in coincidence at the beginning and end of combustion. Despite this arrangement of the curves they follow quite different courses over a considerable portion of their lengths. It is thus apparent that the progress of combustion can not be deduced quantitatively in terms of per cent volume inflamed by visual inspection of pressure-time records.

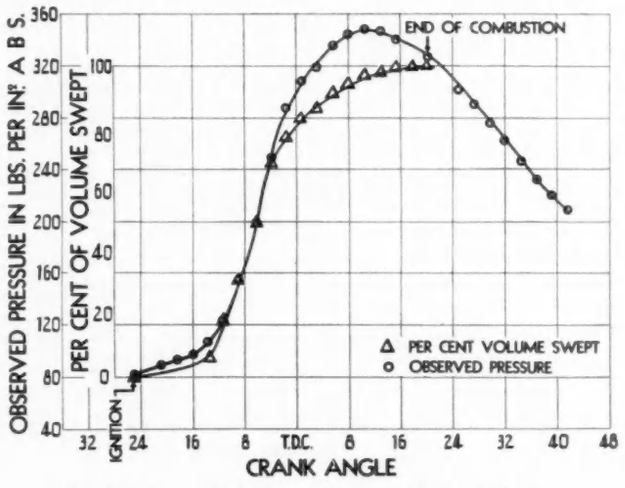


Fig. 8 – Observed Pressure Compared with Inflamed Volume in Each Frame of Fig. 1

On account of the differences in the densities of the inflamed and non-inflamed charge at any given time during the occurrence of an engine explosion, the fractional volume of charge inflamed is not equal numerically to the fractional mass of charge inflamed. Inasmuch as the pressure developed in a given combustion-chamber depends upon the heat liberated and therefore upon the weight of charge burned, the lack of coincidence between the two curves in Fig. 8 is to be expected. On the basis of these facts the next problem will be the determination of the per cent mass of charge burned at each of the crank angles shown in Fig. 1.

Use of Pictures to Determine Mass Burned

The use of two quite similar methods for calculating per cent mass burned in any given flame picture will be illustrated by considering Frame 13 of Fig. 1. According to the data listed below this frame, the pressure was 309 lb. per sq. in. absolute; the per cent volume of combustion space swept by flame was 84 per cent; and the total combustion-chamber volume was 8.4 cu. in. On the basis of these data, it follows that the volume of the inflamed portion of the charge was 7.0 cu. in. and the volume of the non-inflamed portion of the charge was 1.4 cu. in. To facilitate the description of the methods for obtaining the per cent mass burned, Frame 13 has been reproduced in Fig. 9 flanked on the left with Frame 2 which was exposed at the time of ignition and on the right with Frame 22 which was exposed at the end of combustion (22.4 deg. after top dead-center). Below these frames are listed the symbols which will be used to represent the various quantities. (The values in parentheses are the quantities used in calculating per cent mass burned for Frame

13 of this explosion.) The two methods of deducing values of per cent mass burned from these data will now be considered separately.

Method 1 for Calculating Mass Burned – The first method of calculating mass burned depends upon the existence at the time of ignition of a charge density of such a degree of uniformity that any fraction of total volume of charge contains a numerically equal fraction of the total mass. The first step is to find the volume that the section of charge U, which is non-inflamed in Frame 13 of Fig. 1, occupied at the time of ignition. The problem may be stated very simply as follows: Given section U containing a constant fractional mass of charge occupying volume V'_t (1.4 cu. in.) at pressure P_t (309 lb. per sq. in. absolute), find the volume occupied at pressure P_{ti} (82 lb. per sq. in. absolute).

The solution of this problem requires a knowledge of the pressure-volume relationship in the non-inflamed portion of the charge. This relationship may be found by measuring the pressures on the firing-compression card which is reproduced in the lower half of Fig. 2, by converting these values to absolute pressures and by plotting them against corresponding values of total combustion-chamber volume on logarithmic paper. Such a plot is shown by the solid line in Fig. 10, the plotted points representing data taken from both the compression and expansion sides of the firing-compression card. With this engine operating under the conditions maintained while photographing the explosion shown in Figs. 1 and 2, there is approximately a straight-line relationship between the logarithms of the absolute pressure and of the total combustion-chamber volumes throughout the pressure range covered by the compression card. It will be assumed that the same relationship holds for any section of non-inflamed charge throughout the combustion period. Thus, the volume of section U when changed from 309 lb. per sq. in. absolute to 82 lb. per sq. in. absolute will follow the course shown by the dashed line which is parallel to the solid line in Fig. 10. The volume at 82 lb. per sq. in. absolute, as read from the curve, is 3.7 cu. in. Since the total combustion-chamber volume is 10.2 cu. in. at the time of ignition, U represents at this time a fractional volume of 0.36. Because of the uniform density at the time of ignition, the fractional volume of 0.36 contains a fractional mass of 0.36. Hence, there is present in Frame 13 of Fig. 1 a fractional mass of 0.36 in the non-inflamed section and a fractional mass of 0.64 in the inflamed portion; accordingly, the per cent mass inflamed is 64 per cent.

Instead of solving the problem graphically as has been just done, the procedure constituting Method 1 may be expressed in the form of an equation which now will be derived. The polytropic relationship between pressure and volume represented by Fig. 10 may be written, using symbols from Fig. 9,

$$V'_{ti} = V'_t \left(\frac{P_t}{P_{ti}}\right)^{1/n'} \tag{1}$$

Dividing both sides by V_{ti}

$$\frac{V'_{ti}}{V_{ti}} = \frac{V'_t}{V_{ti}} \left(\frac{P_t}{P_{ti}}\right)^{1/n'} \tag{2}$$

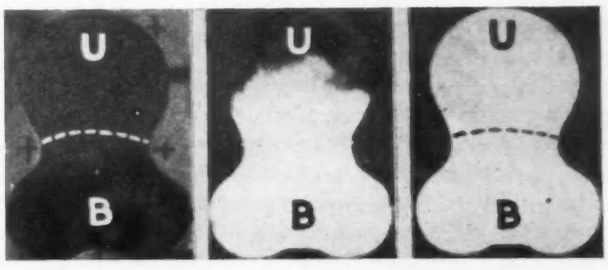
The left side of Equation 2 expresses the fractional volume occupied by section U at the time of ignition. Subtracting both sides of Equation 2 from unity gives

$$1 - \frac{V'_{ti}}{V_{ti}} = 1 - \frac{V'_t}{V_{ti}} \left(\frac{P_t}{P_{ti}}\right)^{1/n'} \tag{3}$$

The left side of Equation 3 represents the fractional volume occupied by section B at the time of ignition. The density of the charge is uniform and thus we also may write:

$$\text{Fractional mass inflamed} = 1 - \left(\frac{V_t - V''_t}{V_{ti}}\right)\left(\frac{P_t}{P_{ti}}\right)^{1/n'} \tag{4}$$

(By Method 1)



TIME	AT IGNITION	DURING COMBUSTION	AT END OF COMBUSTION
OBSERVED PRESSURE	P_{ti} (82)	P_t (309)	P_{tf} (319)
TOTAL VOLUME OF COMBUSTION CHAMBER	V_{ti} (10.2)	V_t (8.4)	V_{tf} (9.8)
VOLUME OF SECTION U	V'_{ti} (3.7)	V'_t (1.4)	
VOLUME OF SECTION B	V''_{ti} (6.5)	V''_t (7.0)	V''_{tf} (6.8)
PER CENT MASS BURNED FROM U FROM B		(64) (69)	

Let n equal the polytropic exponent.

In the above scheme the single prime denotes noninflamed charge and the double prime denotes inflamed charge.

Sections U and B are defined by the fractional mass of charge contained therein. Section U, for example, contains the same fractional mass in all three frames.

Fig. 9 – Diagram Illustrating the Two Methods of Calculating Per Cent Mass Burned from Flame Pictures

The value of n', the polytropic exponent for the unburned charge as determined experimentally for this engine, is the slope of the solid line in Fig. 10. An average value of 1.36 was obtained from several firing-compression cards photographed on the same day and under the same conditions as the cards in Fig. 2. Here, it should be mentioned perhaps that the values of n', as observed over a period of several months, at times have been rather disconcerting. In some instances, for example, the plotted points from the compression and expansion sides of the compression card either have not fallen on the same line or else have shown a marked variation from a straight-line relationship. Such discrepancies have sometimes been traced to slight gas leaks around the valves or window of the engine; but, at other times, the causes of such discrepancies are not known. Even when all the conditions appeared to be satisfactory, the observed values of n' have still varied from 1.20 to 1.36. An alternative to using these experimentally observed values of n' is to assume adiabatic compression of the unburned charge and to assume a reasonable value for the adiabatic exponent. It seems preferable, however, to continue to determine the exponent from firing-compression cards if for no other reason than that such an exercise acts as a continual check on the accuracy of the pressure measurements and on the condition of the engine.

In connection with this first method of obtaining the fractional mass of charge inflamed from the flame pictures it should be noted:

(1) That all the quantities involved in making this calculation are either measured experimentally or derived from experimental observations by straightforward processes.

(2) That the polytropic exponent, 1.36 or the slope of the line in Fig. 10, is determined for compression and expansion of unburned gas from compression cards taken with initial conditions identical with those prevailing for the firing cards.

However, there are two possible objections that might be offered to this first method of calculating mass burned:

(1) The values of pressure used to determine the polytropic exponent are lower than the values of pressure realized during the explosion. In other words, the line relating logarithm of pressure to logarithm of the volume is extrapolated from compression to firing pressures. (2) When the explosion is in progress, it is possible that the temperature of the non-inflamed charge may be rising as a result of absorption of heat radiated from the inflamed portion or as a result of the occurrence within the non-inflamed charge of slow chemical reactions which do not emit light. On account of this latter possibility, the application of the first method to the calculation of the mass burned in pictures of knocking engine explosions would appear to be quite hazardous.

Method 2 for Calculating Per Cent Mass Burned – The second method used for computing mass burned depends upon the assumption that, at the end of combustion, the charge density is of such a degree of uniformity that any fraction of total volume contains a numerically equal fraction of total mass. Hence, the first step in the second method consists in finding the volume of the inflamed section, B, in Fig. 9 at the end of combustion. In other words, if the inflamed volume, V''_t, is 7.0 cu. in. when the observed pressure is 309 lb. per sq. in. absolute, what volume does this same fractional mass of charge occupy at the end of combustion when the pressure is 319 lb. per sq. in. absolute?

Obviously, answering this question requires some knowledge about the pressure-volume relationships in the inflamed portion of the charge. A logarithmic plot of the total combustion-chamber volumes against corresponding absolute pressures is presented in Fig. 11, these plotted pressures being measured during the early part of the expansion stroke from the firing card in Fig. 2. All of the points fall on a straight line having a slope of about 1.30. It will be assumed that this line relates the pressure and volume in either compression or expansion changes in the inflamed charge even though this assumption is subject to certain criticisms which will be considered in some detail later. By a graphical procedure similar to that discussed in connection with Method 1, the volume of section B at the end of combustion is found to be 6.8 cu. in. Since the total combustion-chamber volume at the end of combustion, V''_{tf}, is 9.8 cu. in., the fractional mass burned as determined for the burned section of Frame 13 of Fig. 1 is 69 per cent, which value is to be compared with 64 per cent obtained by the first method.

This second method of deriving per cent mass burned likewise may be expressed in the form of an equation derived as follows: Using the symbols from Fig. 9, the polytropic rela-

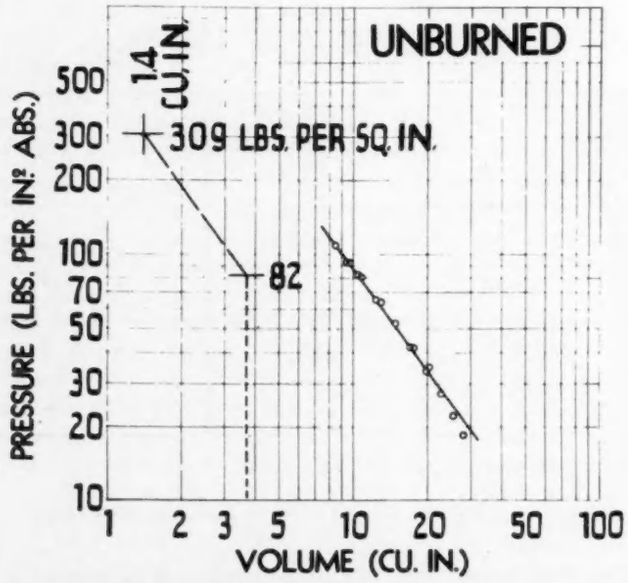


Fig. 10 – Relationship between Pressure and Volume of Non-Inflamed Charge as Determined from the Firing Compression Card ($n' = 1.36$)

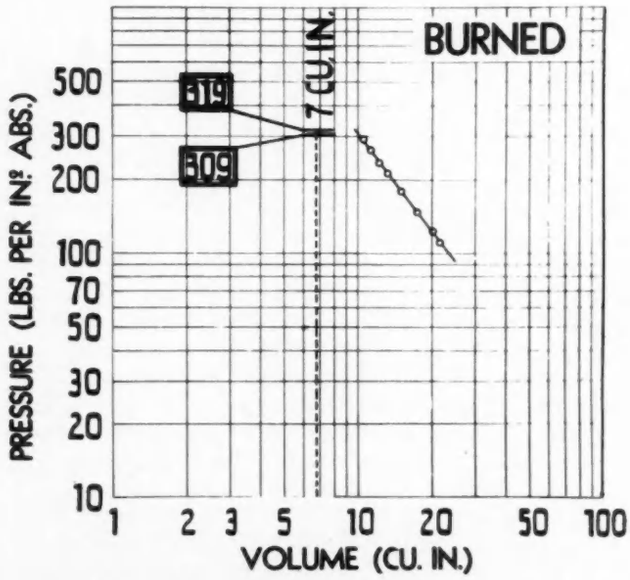


Fig. 11 – Relationship between Pressure and Volume of Inflamed Charge During Early Part of Expansion Stroke ($n'' = 1.30$)

tionship between pressure and volume which is shown graphically in Fig. 11 may be written

$$V''_{tf} = V''_t \left(\frac{P_t}{P_{tf}}\right)^{1/n''} \tag{5}$$

Dividing both sides by V_{tf}

$$\frac{V''_{tf}}{V_{tf}} = \frac{V''_t}{V_{tf}} \left(\frac{P_t}{P_{tf}}\right)^{1/n''} \tag{6}$$

or

$$\text{Fractional mass inflamed (second method)} = \frac{V''_t}{V_{tf}} \left(\frac{P_t}{P_{tf}}\right)^{1/n''} \tag{7}$$

Consider briefly the advantages and disadvantages of the second method. In favor of this method, it will be noted that, again as for the first method, the quantities involved in the calculation of mass burned are either measured directly or are computed by direct methods. On the other hand, two known sources of error appear in this second method of calculating mass burned.

(1) It is known from earlier temperature measurements[5] that, at the instant the combustion is completed, the temperature and density of the inflamed charge are not uniform throughout the entire combustion space. In non-knocking explosions the temperature of the gas which burns first is at least 500 deg. fahr. higher than the temperature of the gas which burns last. The significance of this fact is that the volume, V''_{tf}, occupied by the given portion B at the end of combustion is greater than would be the case if the temperature were uniform throughout the combustion space. In other words, errors from this source tend to give values of mass burned that are too high.

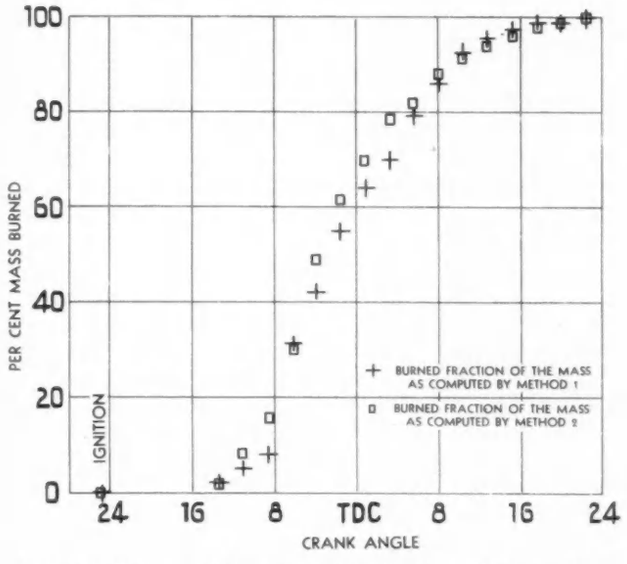


Fig. 12 – Values of Mass Burned in the Several Frames in Fig. 1 as Computed by Each of the Two Methods

(2) The second known source of error has to do with the value of the polytropic exponent. From the pressures and volumes measured during the expansion stroke in the explosion recorded in Figs. 1 and 2, a value of 1.30 was obtained for the polytropic exponent. However, under analogous engine conditions, values for the polytropic exponent have been observed as high as 1.38 and as low as 1.30, the average being about 1.33. Under these engine conditions the reversible adiabatic exponent for the reaction products is 1.25 according to the charts of Hershey, Eberhardt, and Hottel[6]. The difference between this calculated adiabatic exponent and the observed polytropic exponent is due largely to heat losses to the walls. Now, the important point to be noted is that the polytropic exponent which was determined experimentally during an expansion process is being used to determine volume changes during a compression process except for the comparatively small portion of the explosion which occurs in the interval between the time of maximum pressure and the end of combustion. In other words, a polytropic exponent smaller than the reversible adiabatic exponent should be used for most of the computations of mass burned by the second method. The use of a polytropic exponent larger than the adiabatic when an exponent smaller than the adiabatic should be used, tends to raise the results of the calculation of mass burned above the correct value.

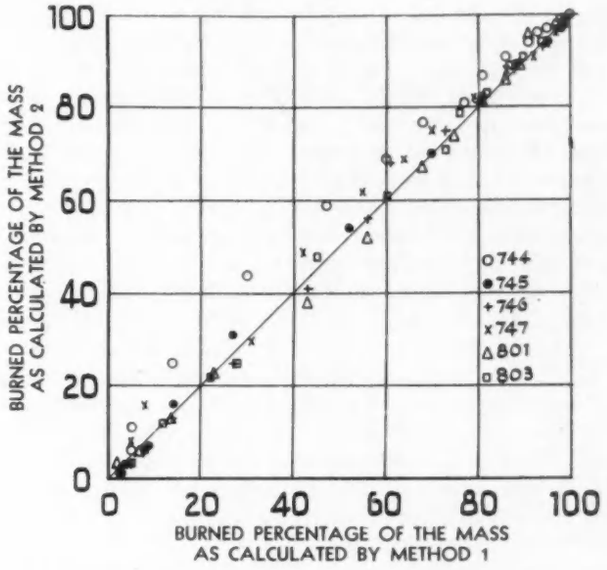


Fig. 13 – Comparison of Values of Per Cent Mass of Charge Burned As Calculated for Six Different Explosions by Two Methods

On this basis, it would appear that, if only the systematic errors in the method were considered without regard for random errors in the measurements, the values of mass burned in any given frame should be higher when calculated by the second method than when calculated by the first method. A comparison of values of mass burned as derived from observed values of volume swept with the two methods will now be made for other frames of the explosion shown in Fig. 1 and for other explosions.

Comparison of the Values of Mass Burned as Calculated by the Two Methods – Plotted against crank angle in Fig. 12 are the values of per cent mass inflamed as deduced by the two methods just described from each of the observed values of volume swept listed in Fig. 1. It will be noted that, in this explosion, the values obtained by the second method are generally, although not always, higher than the values obtained by the first method. The values obtained for the per cent mass of charge inflamed in Frame 9 (exposed at 8.8 deg. before top dead-center) show the maximum spread. This difference amounts to about 8 per cent.

It is of course unwise to draw conclusions on the basis of results from only one explosion and for this reason data from five other explosions will now be introduced. These explosions were all photographed under the same engine conditions as those maintained for Fig. 1. The original data from all

[5] See S.A.E. TRANSACTIONS, April, 1935, pp. 125-133: "Flame Temperatures Vary with Knock and Combustion-Chamber Position," by Gerald M. Rassweiler and Lloyd Withrow.

[6] See S.A.E. TRANSACTIONS, October, 1936, pp. 409-424: "Thermodynamic Properties of the Working Fluid in Internal-Combustion Engines," by R. L. Hershey, J. E. Eberhardt, and H. C. Hottel.

these explosions are summarized in the appendix to this paper.

Fig. 13 presents the values of mass burned as calculated by the two methods for all the frames photographed during the combustion periods of six different explosions. Values of per cent mass burned as derived from observed inflamed volumes by the second method are plotted against the corresponding values derived by the first, and theoretically more accurate, method. Now if the two methods always gave the same result, all the points would lie on the 45-deg. line which has been drawn without regard to the location of the points. On inspecting these data, two observations may be made:

(1) On the average, the results by Method 2 are somewhat higher than by Method 1. In other words, the mean locus of the points in Fig. 13 is slightly above the 45-deg. line. Such a result might be expected on the basis of systematic errors inherent in the methods.

(2) The random departure of the individual points or of individual explosions from the 45-deg. line is large compared to the difference between the 45-deg. line and the mean locus of the points. It has been found that arbitrary variations either in the pressure or in the inflamed volume in any given frame change the value of mass burned as calculated by one method more than the value obtained by the other. On this basis the random variations evident in Fig. 13 probably arise from random errors in the original measurements of volume and pressure.

Graphical Comparison of Per Cent Volume and Per Cent Mass Inflamed – In order to present a simple picture of the numerical relationship between the values of volume swept by flame and the mass of charge contained in the given inflamed volume at any instant during an engine explosion, the per cent mass burned data have been plotted in Fig. 14 against corresponding measurements of the per cent volume inflamed. The plotted values of mass burned are the average of the results obtained by the two different methods from the six explosions described in connection with Fig. 13. It is apparent

[7] See Mitteilungen über Forschungsarbeiten, Vol. 54, 1908, pp. 1-42: "Versuche über die Zündgeschwindigkeit Explosiber Gasgemische," by A. Nagel; see also Wiens Sitzungsberichte, Vol. 126, 1917, p. 9, by Flamm and Mache; see also S.A.E. TRANSACTIONS, Vol. 17, 1922, Part I, pp. 94-120: "Molecular Movements During Combustion in Closed Systems," by Thomas Midgley, Jr.; see also Bulletin No. 157, Engineering Experiment Station, University of Illinois, July, 1926, by C. Z. Rosencranz.

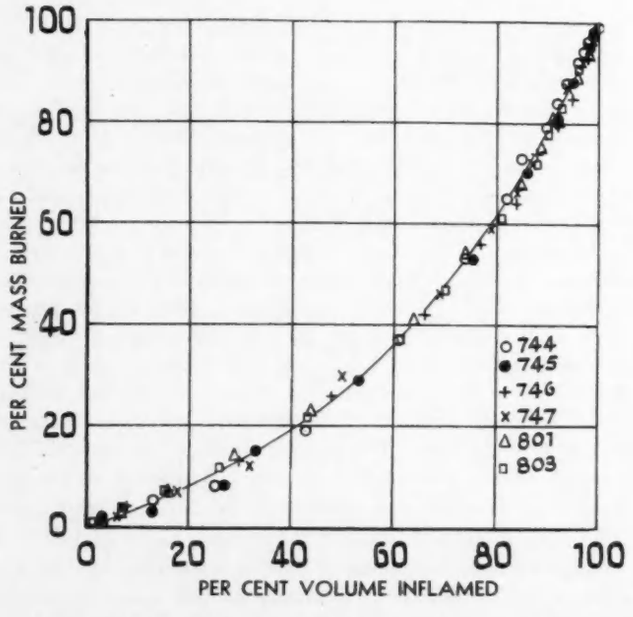


Fig. 14 – Relationship between Volume Swept and Mass Burned in Six Explosions Photographed Under Similar Engine Conditions

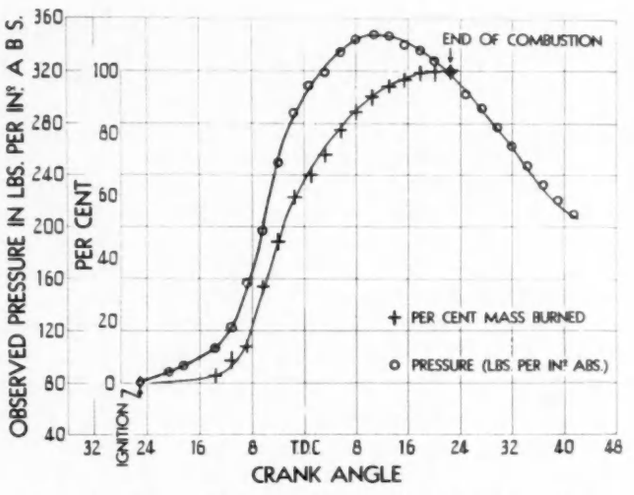


Fig. 15 – Observed Pressure Compared with Per Cent Mass Burned in Each Frame of Fig. 1

from this curve that, when the flame has swept through 25 per cent of the volume of the combustion-chamber, only about 10 per cent of the mass has been burned. When 50 per cent of the combustion-chamber is filled with flame only about 25 per cent of the mass has been burned. Again, when 75 per cent of the volume is inflamed, there is still half, by weight, of the original mixture left to burn; in other words, 50 per cent of the weight of the charge is burned in that portion of the combustion space where the flame sweeps through the last 25 per cent of charge volume.

Pressure and Mass Compared

Direct Comparison Between the Pressure Developed in the Engine and the Per Cent Mass Burned – The next step in the analysis of the high-speed motion pictures of engine flames and simultaneously recorded pressure records consists in a study of the relationship between the per cent mass of charge burned and the pressure developed in the engine cylinder. On account of the comparatively small piston motion during the combustion period in a gasoline engine, the combustion phenomenon frequently is assumed to take place at constant combustion-chamber volume. If this assumption is permissible, one might reason that much of the thermodynamic treatment of constant-volume bomb explosions can be applied directly to engine combustion. Now, in the theoretical treatment of the combustion process in the constant-volume bomb as presented by Nagel, Flamm and Mache, Midgley, Rosecrans and others[7], it appears that the fractional mass of charge burned at any instant during a given explosion is practically equal to the fractional pressure rise. Consequently, if it is permissible to neglect the changes in combustion-chamber volume during combustion in an engine, one would expect the observed pressure-time curve from Fig. 2 to be of the same shape as the mass burned-time curve from Fig. 12, particularly so, if the two curves are plotted with comparable ordinate scales.

A direct comparison of per cent mass burned and engine pressure is made in Fig. 15 for the explosion represented by Fig. 1. The values of mass burned plotted in Fig. 15 are averages of results by the two methods of calculation. The ordinates in this figure have been so scaled that the mass-burned and pressure curves are in approximate coincidence at the beginning and end of combustion. Despite this arrangement of the data, the shape of the mass-burned curve differs quite considerably from the observed pressure curve. The most obvious differences in the shapes of these curves are to be seen between top dead-center and the end of combus-

tion. In that interval, the data show the striking fact that the pressure reached a maximum in this explosion about 12 deg. of crankshaft revolution before flame propagation was completed. This observation furnishes a clue to the reason for the discrepancy between these two curves.

The motion of the piston during the interval between top dead-center and 20 deg. after top dead-center is very small, only 0.2 in. For this reason, the pressure-volume changes resulting from piston motion during the combustion process are frequently neglected in considering the engine cycle. But Fig. 15 suggests that, in the interval between top dead-center and the end of combustion, the effects of piston motion tending to decrease the pressure are greater than the effects of combustion tending to increase the pressure. The very considerable effect of such small amounts of piston motion during the combustion process results from the obvious fact that, as the pressure level is increased, the pressure change produced by a given change in total combustion-chamber volume also increases. Thus, the next logical step in finding the relationship between the mass of charge burned and the pressure developed by combustion is to sort out from the total pressure changes the changes due to combustion and the changes due to piston motion. In carrying out such a process, account must be taken of the fact that, as the charge burns and the pressure rises, the magnitudes of the pressure changes resulting from piston motion become greater and greater.

[8] See *Automotive Industries*, Mar. 2, 1935, pp. 324-329; Mar. 9, 1935, pp. 354-357; and Mar. 16, 1935, pp. 394-397: "Factors Controlling Engine Combustion," by Hector Rabezzana and Stephen Kalmar.

Use of Pressure Cards

The Use of Observed Pressure Cards to Determine the Pressure Changes Due to Combustion – Fig. 16 illustrates the system used for determining approximately the pressure changes due to combustion from the observed pressure card. The method first will be described briefly to show the general scheme used and a few of the details then will be discussed more fully. In some respects, this system resembles the one described in the appendix to the paper presented in 1935 by Rabezzana and Kalmar[8].

The time axis of the observed pressure card is first divided into a number of small, equal intervals of crank angle – the smaller, the better. The intervals chosen here are 2.4 crankshaft deg., which represent the time between the exposure of successive frames of the corresponding flame picture as indicated by the sequence of frame numbers just above the pressure curve in Fig. 16. Now consider any one of these intervals, say that between Frames 7 and 8, which is shown as an enlarged section at the bottom of the figure. During this interval, the total observed pressure change was 16 lb. per sq. in. This pressure increment may be divided into two parts, one due to piston motion and the other due to combustion. To determine the pressure change due to piston motion, imagine that combustion is momentarily stopped at the end of exposure of Frame 7 and that the piston is then allowed to move a distance corresponding to the next 2.4 deg. of crankshaft revolution. The total combustion-chamber volume decreases from 8.9 to 8.7 cu. in.; the pressure before the change was 107 lb. per sq. in. abs.; the problem then is to find the pressure at the end of this volume change. The relationship between pressure and volume changes in the burned and unburned gases has been considered in the first part of this paper (see Figs. 10 and 11). There it was found that the polytropic exponents as determined experimentally, first, for both the compression and expansion processes in the unburned charge and, again, for an expansion process in the burned charge were nearly equal to 1.3. Using this value for the polytropic exponent of the mixture of burned and unburned charge, it can be found readily that, when the combustion-chamber volume is decreased from 8.9 to 8.7 cu. in. without combustion, the pressure will rise from 107 to 110 lb. per sq. in. Thus, if combustion were stopped momentarily at the end of exposure of Frame 7, and if the piston were then moved until the combustion-chamber volume was equivalent to that at the end of exposure of Frame 8, the pressure change due to piston motion would be 3 lb. per sq. in. The remaining pressure rise of 13 lb. per sq. in. is considered to be due to combustion as indicated in Fig. 16. This process

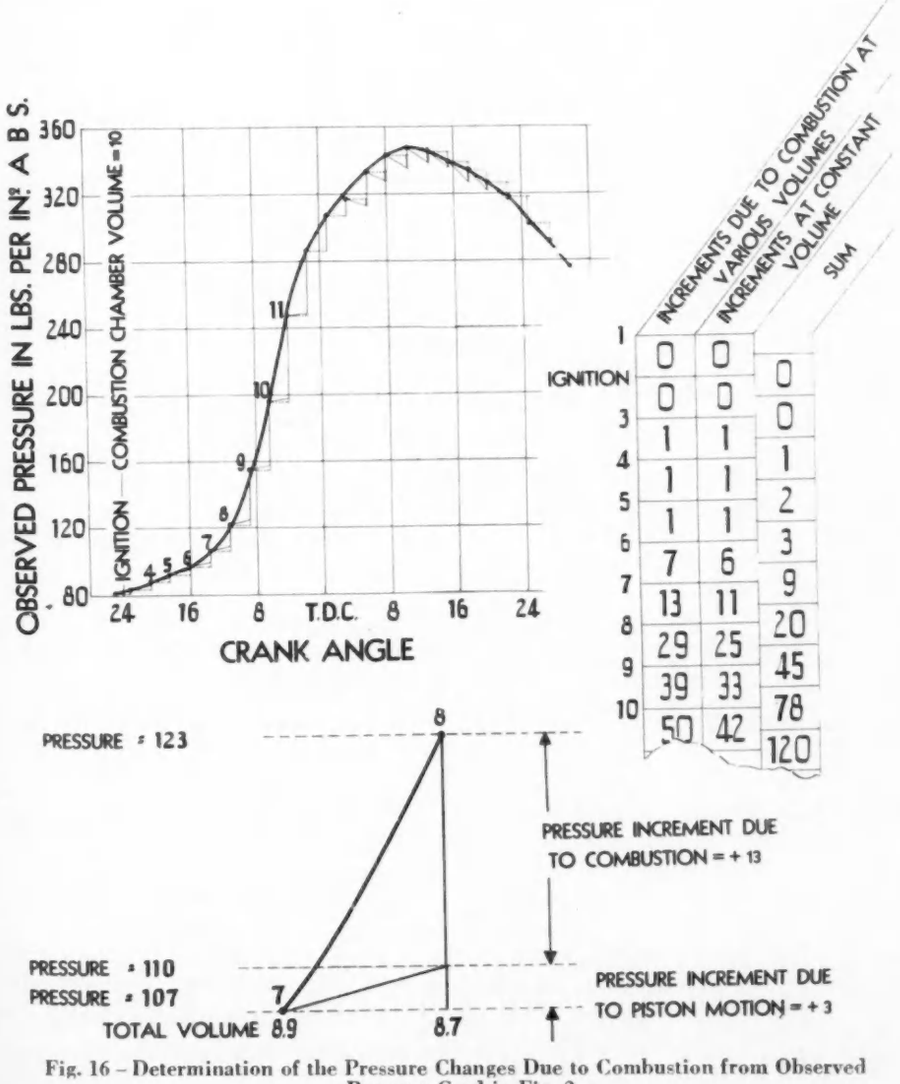


Fig. 16 – Determination of the Pressure Changes Due to Combustion from Observed Pressure Card in Fig. 2

can be repeated for each interval of 2.4 deg., as indicated on the pressure card.

Now what has been done so far? The original pressure card, instead of being considered as a smooth curve, is conceived as being composed of a series of steps. For purposes of this analysis, the pressure first is changed polytropically to account for the volume change due to piston motion; then, an increment of charge burns; the pressure is again changed polytropically; another increment of charge burns; and so on until the point is reached where the total pressure change can be accounted for by the polytropic pressure changes made to allow for the piston motion. The pressure increments due to

Table 1 – Computation of Per Cent Pressure Rise Due to Combustion from the Pressure-Time Record Reproduced in Fig. 2

1	2	3	4	5	6	7	8	9	10	11	12	13
FRAME NUMBER	CRANK-SHAFT ANGLE	TOTAL VOLUME OF COMBUSTION SPACE	FIRING PRESSURE (lbs/sq.in., abs.) FIRST OBSERVER	FIRING PRESSURE (lbs/sq.in., abs.) SECOND OBSERVER	FIRING PRESSURE (lbs/sq.in., abs.) AVERAGE	$\left(\frac{V_{t1}}{V_{t2}}\right)^{1.3} - 1$	PRESSURE INCREMENTS DUE TO PISTON MOTION $P_{t2}\left(\frac{V_{t1}}{V_{t2}}\right)^{1.3} - P_{t1}$ (lbs/sq. in.)	TOTAL PRESSURE INCREMENT $(P_{t2} - P_{t1})$ lbs/sq. in.	PRESSURE INCREMENTS DUE TO COMBUSTION AT VARIOUS VOLUMES COL. 9 – COL. 8 lbs/sq. in.	PRESSURE INCREMENTS DUE TO COMBUSTION AT CONSTANT VOLUME COL. 10 × V_t/V_{t1} lbs/sq. in.	SUMMATION OF PRESSURE INCREMENTS LISTED IN COLUMN 11 lbs/sq. in.	PER CENT PRESSURE RISE UNDER CONSTANT VOLUME CONDITIONS
Ign.	-2.0	10.15	80	84	82							
3	-23.2	9.91	85	86	86	+0.032	+ 3	+ 3	0			
4	-20.8	9.61	86	91	89	+0.041	+ 3	+ 4	1	1	1	
5	-18.4	9.34	91	95	93	+0.038	+ 3	+ 4	1	1	2	
6	-16.0	9.11	96	98	97	+0.034	+ 3	+ 4	1	1	3	1
7	-13.6	8.90	107	106	107	+0.030	+ 3	+10	7	6	9	4
8	-11.2	8.72	122	124	123	+0.024	+ 3	+16	13	11	20	9
9	- 8.8	8.58	154	156	155	+0.022	+ 3	+32	29	25	45	19
10	- 6.4	8.47	200	194	197	+0.017	+ 3	+42	39	33	78	33
11	- 4.0	8.40	252	246	249	+0.011	+ 2	+52	50	42	120	51
12	- 1.6	8.38	290	285	288	+0.006	+ 1	+39	38	31	151	64
13	+ 0.8	8.38	309	308	309	0.0	0	+21	21	17	168	71
14	+ 3.2	8.39	319	318	319	0.0	0	+10	10	8	176	75
15	+ 5.6	8.44	333	335	334	-0.01	- 3	+15	18	15	191	81
16	+ 8.0	8.54	344	344	344	-0.016	- 5	+10	15	13	204	86
17	+10.4	8.67	347	348	348	-0.020	- 7	+ 4	11	9	213	90
18	+12.8	8.84	342	349	346	-0.025	- 9	- 2	7	6	219	93
19	+15.2	9.04	338	341	340	-0.028	-10	- 6	4	4	223	95
20	+17.6	9.26	335	334	335	-0.032	-11	- 5	6	5	228	97
21	+20.0	9.52	328	325	327	-0.035	-12	- 8	4	4	232	98
22	+22.4	9.81	318	319	319	-0.038	-12	- 8	4	4	236	100
23	+24.8	10.13	302	302	302	-0.041	-13	-17	- 4			
24	+27.2	10.47	292	290	291	-0.043	-13	-11	+ 2			
25	+29.6	10.85	276	276	276	-0.046	-13	-15	- 2			
26	+32.0	11.28	262	261	262	-0.048	-13	-14	- 1			
27	+34.4	11.71	246	248	247	-0.048	-12	-15	- 2			
28	+36.8	12.17	230	234	232	-0.048	-12	-15	- 3			
29	39.2	12.64	218	222	220	-0.049	-11	-12	- 1			
30	41.6		208	212	210	-0.049	-11	-10	+ 1			

combustion as found for each of the intervals between the exposures of the first ten frames of the explosion reproduced in Fig. 16 are listed in the second column at the right of the figure.

Each of the pressure increments results from burning a small fraction of the total mass of charge. An inspection of Fig. 1 shows that each of these combustion increments occurs at a different combustion-chamber volume. For example, the total combustion-chamber volume at the end of the exposure of Frame 4 was 9.6 cu. in., whereas the total combustion chamber volume at the end of the exposure of Frame 10 was 8.5 cu. in. So, to eliminate completely the effects of volume change due to piston motion, it is desirable to determine what the magnitudes of the several pressure increments due to combustion would be if they had all occurred in the same total combustion-chamber volume. The volume of the combustion-chamber to which the various increments will be reduced is the volume at the time of ignition; namely, 10.2 cu. in. for this explosion. Now consider again the increment between Frames 7 and 8. Fig. 16 shows that the pressure rise due to combustion was 13 lb. per sq. in. Fig. 1 shows that, during this interval, 5 per cent of the mass of charge burned. Up to this point in the analysis, this 5 per cent of the mass of charge has been considered as burning at a combustion-chamber volume of 8.7 cu. in., producing a pressure rise of 13 lb. per sq. in. The problem now is to find what pressure rise would have been produced if this 5 per cent of the mass had burned at a combustion chamber volume of 10.2 cu. in. Now, it can be shown that the pressure increase produced in a constant-volume bomb by liberating a given amount of energy in a given mass of charge is inversely proportional to the total volume of the bomb.[9] So the pressure increment, 13 lb. per sq. in., is multiplied by 8.7/10.2 giving 11 lb. per sq. in. This same procedure can be applied, with appropriate volume ratios, to each of the pressure increments. The third column of the tabulation at the right of Fig. 16 shows the pressure increments that would have resulted if the various portions of the mass of charge, as listed in per cent beneath the several flame pictures in Fig. 1, had been burned with a total combustion-chamber volume of 10.2 cu. in. instead of a series of different total combustion-chamber volumes as actually was the case. Now, in order to find the pressure rise due to combustion at the end of exposure of Frame 8, for example, all of the increments of combustion occurring prior to the end of exposure of Frame 8 are summed. The results of such a summation process for the first ten frames are presented in the fourth column of the tabulation at the right of Fig. 16.

The details of the procedure for determining the pressure rise due to combustion by use of the system just described are illustrated in Table 1. The first six columns of this table present data obtained directly from the engine as indicated by the headings. The two pressure observations were made on the same card in an effort to decrease the errors introduced while laying out a time axis and reading the pressures. Column 7 presents a series of factors found by subtracting unity from the ratio, raised to the 1.3 power, of the total combustion chamber volume at the beginning (V_{t1}) and at the end (V_{t2}) of the interval between the exposure of each consecutive pair of flame pictures. Each of these factors is multiplied by the pressure observed at the beginning of the corresponding interval to obtain the data in column 8. Column 9 presents the actual pressure changes observed during each interval, and the difference between corresponding items in Columns 9 and 8 are the pressure increments due to combustion at various total combustion-chamber volumes. It will be noted on Fig. 16 and in Column 8 of Table 1 that the pressure increments due to piston motion decrease in magnitude as top dead-center is approached and change sign after passing top dead-center; but, between top dead-center and maximum pressure, the effect of combustion is greater than the effect of piston motion with the result that the observed pressure increments remain positive. After maximum pressure, Frame 17, the increments in Columns 8 and 9 are both negative showing that the effect of the downward motion of the piston on the pressure more than overcomes the effect of combustion. The angle at which the pressure increments due to combustion (Column 10) become zero or negative is considered the end of combustion (see Frame 22, Table 1). It is important to note that the angles of revolution determined for the end of combustion by breaking down the pressure card generally lie within ± 5 deg. of the end of flame propagation as observed on flame pictures of the same explosions.

Column 11 of Table 1 presents the pressure increments due to combustion at a constant volume of 10.2 cu. in., whereas Column 12 sums, at each frame, the prior pressure increments. These values of pressure rise, above the pressure at ignition, are reduced to a per cent basis in Column 13.

In judging the accuracy of the preceding determination of the per cent pressure rise due to combustion, the following considerations must be taken into account:

(1) The effects of piston motion and combustion have been considered as taking place in a stepwise manner instead of simultaneously. Small steps have been used to avoid large errors from this source.

(2) The effects of heat loss on the development of pressure are not dealt with very adequately. If accurate values were available for the polytropic exponents of the mixtures of non-inflamed and inflamed charge for each 2.4-deg. interval, then the increments in Column 8 of Table 1 would represent the combined effects of piston motion and heat loss. The value of 1.3 has been used because it represents the average value as obtained experimentally. Actually, a change from 1.25 to 1.35 in the value of this exponent has only a small effect on the results of the analysis.

(3) A gas leak in the engine may affect the results. A continual effort is made to detect and prevent such leaks.

(4) In changing the pressure increments listed in Column 10 of Table 1 from variable-volume to constant-volume conditions, it was stated that the pressure increment resulting from the combustion of a given fraction of the mass of charge is inversely proportional to the total combustion-chamber volume. From the equations developed by Flamm and Mache[7] for a constant-volume bomb, it appears that the polytropic exponents for the burned and unburned sections of the charge must be equal for such a relation to hold rigorously.

Comparison of Pressure Rise

Comparison of the Per Cent Pressure Rise Due to Combustion with the Per Cent Mass Burned – It was shown in Fig. 15 that the mass-burned curve did not match the observed pressure curve; furthermore, the data suggest that the disagreement was caused by pressure-volume changes resulting from piston motion. Now that the pressure changes produced by combustion have been separated from those due to piston motion and have been expressed upon a percentage basis, these values will be compared with corresponding values of per cent mass burned. Plotted against crank angle in Fig. 17 are the values for per cent pressure rise due to combustion at constant volume, as obtained from the observed pressure-time curve shown in Fig. 15, and the values of per cent mass burned as derived from the flame pictures of the same explo-

[9] See *Industrial and Engineering Chemistry*, Vol. 17, April, 1925, pp. 397-402: "Gaseous Explosions," by G. G. Brown, E. H. Leslie, and J. V. Hunn.

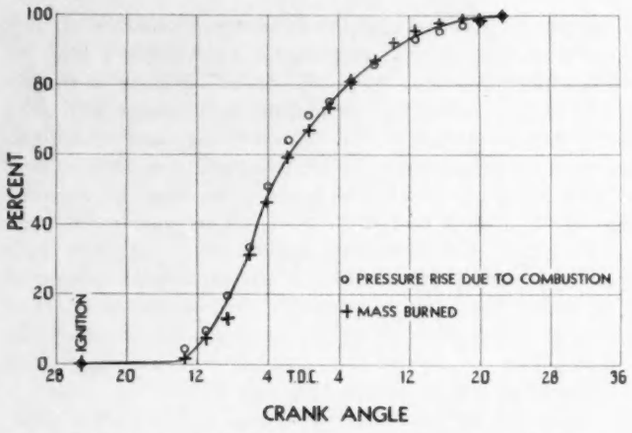


Fig. 17 – Per Cent Pressure Rise Due to Combustion Compared with Per Cent Mass Burned in Each Frame of Fig. 1

sion. It is at once apparent that curves drawn through either set of points in Fig. 17 will follow essentially the same course. Thus, it may be concluded that, throughout the entire combustion period of this explosion, the value of per cent mass burned at any given crank angle is very nearly equal to the per cent pressure rise due to combustion at that same angle. The significance of this result is that information about the fraction of mass of charge burned at any instant during this engine explosion may be obtained by finding the per cent of the pressure rise due to combustion using observed pressures and total combustion-chamber volumes as the only original data. Similar computations have been made with data from five other engine explosions, and the results of these computations now will be compared with the mass burned values.

Plotted against crank angle in Fig. 18 are the observed cylinder pressures, the values of per cent mass of charge burned and the per cent pressure rise due to combustion as determined for six different engine explosions. The flame pictures of these explosions are shown in the appendix (Fig. 22), together with observed data from the flame pictures and pressure records (Table 2). Differences of considerable magnitude between the numerical values of per cent mass burned and per cent pressure rise due to combustion appear in explosions 744 and 801, shown respectively, in the upper-left and in the lower-left corners of Fig. 18. In explosion 744 the values for the per cent pressure rise due to combustion are as much as 11 per cent higher than the corresponding mass-burned values; in explosion 801, the values for the per cent pressure rise due to combustion are as much as 12 per cent lower than the corresponding mass burned values. It will be noted, however, that a curve fitted to the points representing either per cent mass burned or per cent pressure rise due to combustion in these explosions will follow the same general course and that, during the interval in which the points show the greatest discrepancy, the pressures in the cylinder are rising very rapidly. These facts suggest the possibility that the discrepancies shown by explosions 801 and 744 in Fig. 18 may be due, partially at least, to errors in correlating the time scales for the flame pictures and corresponding pressure cards. In the four other engine explosions which were recorded under similar engine conditions, a fairly close agreement was found for the values of per cent of pressure rise due to combustion and per cent mass burned.

It follows then that, under the engine conditions used for these experiments, the per cent mass burned at any given instant during an engine explosion may be determined with a fair degree of certainty from an observed pressure card without reference to flame pictures simply by finding the per cent pressure rise due to combustion at the given instant. The accuracy of the results, of course, will depend primarily upon the accuracy of the pressure measurements.

Comparison of Inflamed Fractions

Comparison of Inflamed Fractions of Charge as Observed in the Engine with Values Calculated for Conditions of Constant Combustion-Chamber Volume – In developing relationships between the pressure and inflamed volume for bomb explosions, it is assumed frequently that the fractional pressure rise is equal to the fractional mass burned and that, prior to its combustion, the non-inflamed portion of the charge is compressed adiabatically. Since it has been shown for the engine explosions described herein that the fractional pressure rise due to combustion as reduced to constant-volume conditions is approximately equal to the burned fraction of the mass and that the polytropic exponent for the non-inflamed charge differs from the adiabatic by only a few per cent, it is natural to suppose that relationships of the type developed for bomb explosions may hold approximately for engine explosions, providing of course that pressure values from which the effects of piston motion have been eliminated are used instead of the observed pressure data. This possibility will now be investigated in the following way:

(1) On the basis of the assumption that the piston remains at rest throughout an explosion, an equation will be set up for calculating fractional volume inflamed from fractional pressure rise due to combustion as reduced to constant-volume conditions.

(2) Values of fractional volume swept by flame as calculated with this equation will then be compared with inflamed fractions of charge as observed on the flame pictures.

The symbols used in the derivation are as follows:

v'_i = specific volume of the charge at the time of ignition.
v' = specific volume of the non-inflamed portion of the charge at any given time.
V = total volume of the combustion-chamber (assumed to be constant).
V' = volume of non-inflamed portion of the charge at the given time.
V'' = volume of inflamed portion of the charge at the given time.
P_i = absolute pressure observed at the time of ignition.
P = absolute pressure (reduced to constant volume) at the given time.
P_f = final pressure (reduced to constant volume) at the end of the combustion process.
M' = mass of non-inflamed section of the charge at the given time.
M'' = mass of inflamed section of the charge at the given time.
M = total mass of charge.
n' = the polytropic exponent for the non-inflamed portion of the charge.

It will be noted that these symbols are quite similar to those shown in Fig. 9 except that the subscript "*t*" has been omitted to distinguish constant-volume quantities from the actual observed quantities.

Now the experimental data plotted in Fig. 10 show that, during the compression stroke – prior to ignition – the unburned portion of the charge obeys the polytropic law

$$P^{1/n'} v' = K \tag{8}$$

where n' = the exponent for the non-inflamed portion of the charge
K = a constant.

In this derivation it is assumed that the relationship between the pressure and specific volume of the non-inflamed

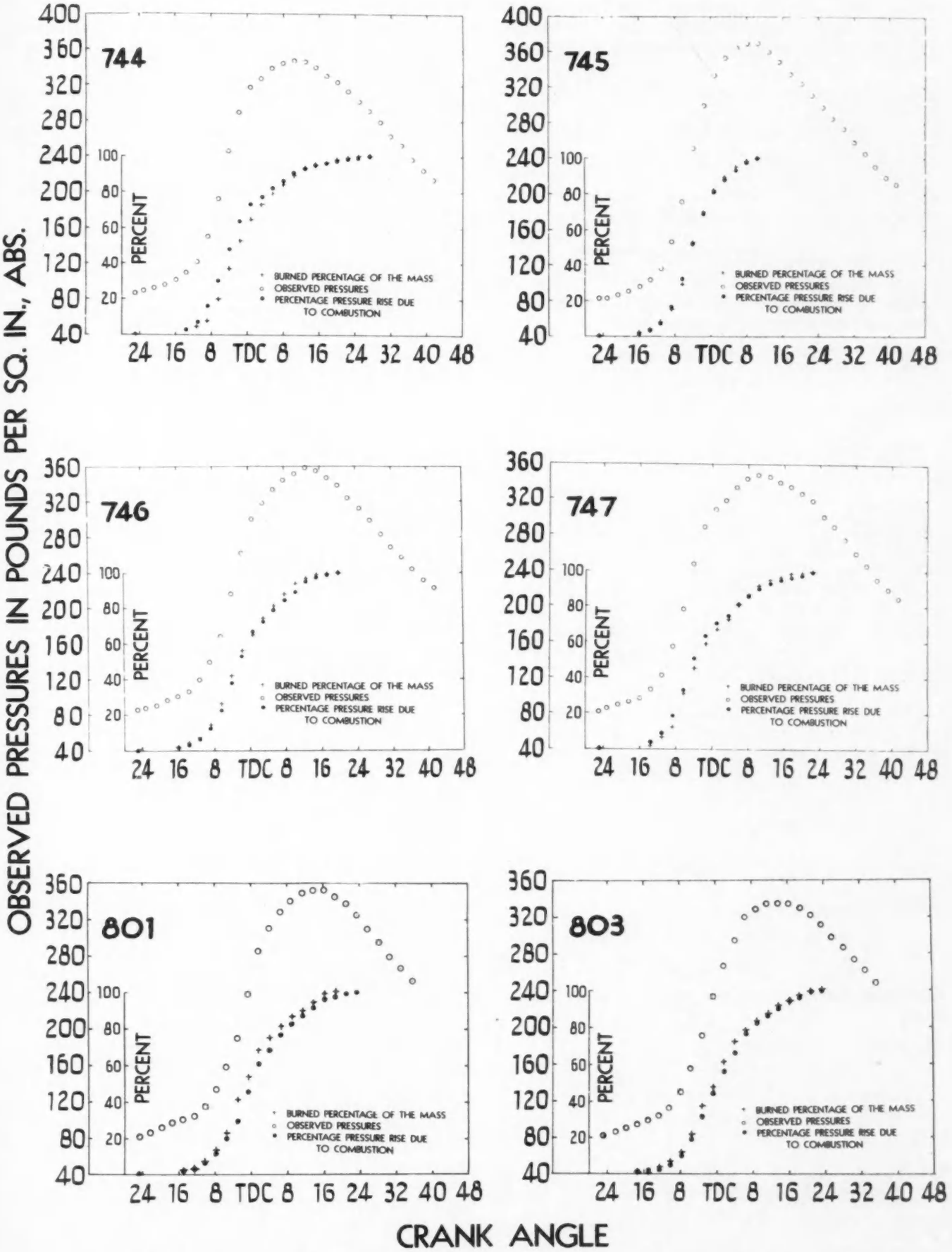


Fig. 18 – Per Cent Mass Burned Compared with Per Cent Pressure Rise Due to Combustion in Six Engine Explosions

charge as expressed by Equation (8) holds after ignition as well as before ignition. In order to express K in terms of measurable quantities, the particular values of pressure and specific volume at the time of ignition may be substituted in Equation (8) giving

$$P_i^{1/n'} v_i' = K \tag{9}$$

Substituting this value of K in Equation (8) and rearranging, we have

$$\left(\frac{P_i}{P}\right)^{1/n'} \frac{v_i'}{v'} = 1 \tag{10}$$

But the specific volume at the time of ignition,

$$v_i' = V/M = \frac{\text{Total combustion chamber volume}}{\text{Total mass of charge in the chamber}} \tag{11}$$

and the specific volume of the non-inflamed charge at any given time during the explosion is

$$v' = V'/M' = \frac{\text{Volume of charge not yet inflamed}}{\text{Mass of charge not yet inflamed}} \tag{12}$$

Substituting Equations (11) and (12) in Equation (10)

$$\left(\frac{P_i}{P}\right)^{1/n'} \left(\frac{V}{M}\right)\left(\frac{M'}{V'}\right) = 1 \tag{13}$$

or

$$V'/V = (P/P_i)^{-1/n'} \times M'/M \tag{14}$$

In Equation 14 the quantity (M'/M) is the non-inflamed fraction of mass and the quantity (V'/V) is the non-inflamed fraction of volume. Consequently, $(1 - M'/M)$, which is equal to M''/M, is the inflamed fraction of mass, whereas $(1 - V'/V)$ is the inflamed fraction of volume. Thus, if both sides of Equation 14 are subtracted from unity, and if $(1 - M''/M)$ is substituted for M'/M, there is obtained

$$(1 - V'/V) = 1 - (P/P_i)^{-1/n'} (1 - M''/M) \tag{15}$$

On the basis of the assumption that the fractional mass burned is equal to the fractional pressure rise due to combustion, we may write

$$(M''/M) = \frac{P - P_i}{P_f - P_i} \tag{16}$$

Then, writing the identity

$$\frac{P - P_i}{P_f - P_i} = \frac{P - P_i}{P_f - P_i} \tag{17}$$

and solving for P/P_i, we get

$$P/P_i = \left(\frac{P - P_i}{P_f - P_i}\right)\left(\frac{P_f}{P_i}\right) - \left(\frac{P - P_i}{P_f - P_i}\right) + 1 \tag{18}$$

Substituting Equations (16) and (18) in Equation (15), we obtain

Fractional volume inflamed =

$$\frac{V''}{V} = 1 - \left[\left(\frac{P - P_i}{P_f - P_i}\right)\left(\frac{P_f}{P_i}\right) - \left(\frac{P - P_i}{P_f - P_i}\right) + 1\right]^{-1/n'} \left[1 - \frac{P - P_i}{P_f - P_i}\right] \tag{19}$$

Equation (19) expresses the fractional volume swept by flame as a function of three quantities: (1) the fractional pressure rise due to combustion $\frac{P - P_i}{P_f - P_i}$; (2) the ratio of the final constant-volume pressure, P_f, to the pressure at ignition, P_i, and, (3) the polytropic exponent, (n'), of the non-inflamed portion of the charge. The equation is represented graphically by the family of curves in Fig. 19. Each curve represents the relationship between per cent volume swept by flame and per cent pressure rise due to combustion for a given value of the ratio, P_f/P_i, of the final constant-volume pressure to the pressure at the time of ignition. This ratio is constant for any given explosion but it may take different values under different engine conditions. The shapes of the curves are not much affected by changes in n' of the magnitudes to be expected in practice. It has been found, for example, that changing n' from 1.30 to 1.40 changes the volume swept corresponding to a given per cent pressure rise due to combustion by only 1 per cent. In preparing Fig. 19, a value of 1.35 was used for the polytropic exponent of the non-inflamed charge.

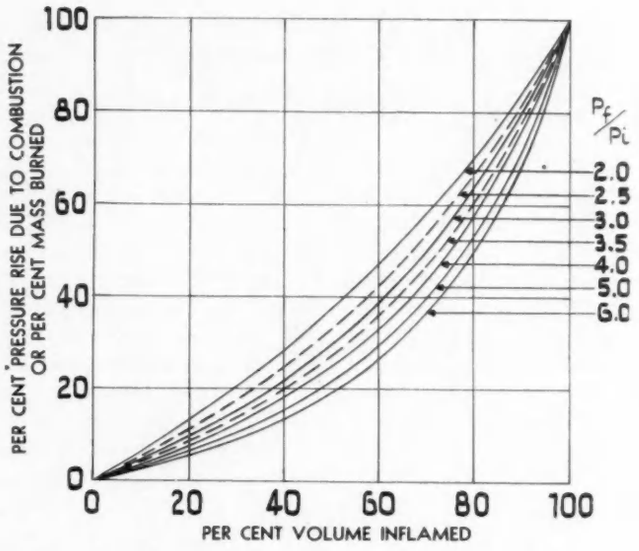


Fig. 19 – Curves Used to Derive Per Cent Volume Swept from Per Cent Pressure Rise Due to Combustion

Using the family of curves in Fig. 19, the inflamed fraction of the volume may be obtained readily at any given stage in the explosion from values of pressure rise due to combustion as deduced from observed pressure cards. However, it is apparent from the development of Equation (19) that these fractions of inflamed volume are calculated on the basis of the assumption that the piston remains at rest in the position occupied at the time of ignition. On the other hand, the quantity desired is the actual fraction of volume swept (for the same burned fraction of the mass) with the piston in motion. If the value calculated by Equation (19) is to represent the actual value, at least one condition must be met. The pressure-volume relationship in the burned portion of the charge must be identical with the relationship in the unburned portion during compression and expansion processes; that is, n' should equal n''. This same assumption was made in deducing pressure rise due to combustion from the observed pressure cards. Since this condition probably is never fulfilled in the actual case, errors from this source may produce a spread between the calculated and observed fractions of inflamed charge in the engine explosions.

Values of fractional volume swept which were calculated in the foregoing manner for the six explosions described in connection with Fig. 13 are compared in Fig. 20 with values observed on the flame pictures. These calculated values agree fairly well with the observed values of fractional volume inflamed in four of the six explosions. In the curves from explosion 744, which shows the poorest agreement, the maximum difference between the observed and calculated values is 13 per cent. It is not possible at present to proportionate the spread shown by the points in Fig. 20 between the

random experimental errors in the original measurements and the inherent errors in the various steps involved in deducing volume swept from the engine pressure measurements. But these data do furnish an illustration of the results to be expected by reducing the observed pressure data to constant-volume conditions and then by calculating from these data the fractional volume inflamed on the basis of the assumption that the total volume remains constant.

A Relationship

A Relationship Between Pressure Developed, Fractional Volume of Combustion Space Swept by Flame and Total Combustion-Chamber Volume – In the preceding discussion, a method is described for computing from a pressure card either the fractional mass or the fractional volume of inflamed charge at any time during an engine explosion. Both of these computations require the determination of the pressure rise due to combustion under constant-volume conditions which determination involves a number of time-consuming steps (see Table 1). Consequently, it seemed advisable to devise a shorter method for determining the approximate values of either the fractional mass or the fractional volume of inflamed charge from the pressure cards. A comparatively fast method of determining these quantities from total combustion-chamber volumes and corresponding pressures observed during the combustion period will now be outlined.

In developing the equation it will be assumed that the mass burned as calculated by Equation (4) (Method 1) is equal to the mass burned as calculated by Equation (7) (Method 2). That such an assumption is only an approximation is apparent both from the nature of the methods and from the data in Figs. 12 and 13. Just how good or how bad the approximation is for the present purposes perhaps can be judged best from the results which follow. The polytropic exponents ($n' = 1.36$ and $n'' = 1.30$) used in solving Equations (4) and (7) differ by only about 5 per cent; hence, they will be assumed equal. Replacing n' and n'' by n in Equations (4) and (7) and equating the two expressions for fractional mass inflamed, we obtain

$$1 - \frac{(V_t - V_t'')\, P_t^{1/n}}{V_{ti}\, P_{ti}^{1/n}} = \frac{V_t''\, P_t^{1/n}}{V_{tf}\, P_{tf}^{1/n}} \qquad (20)$$

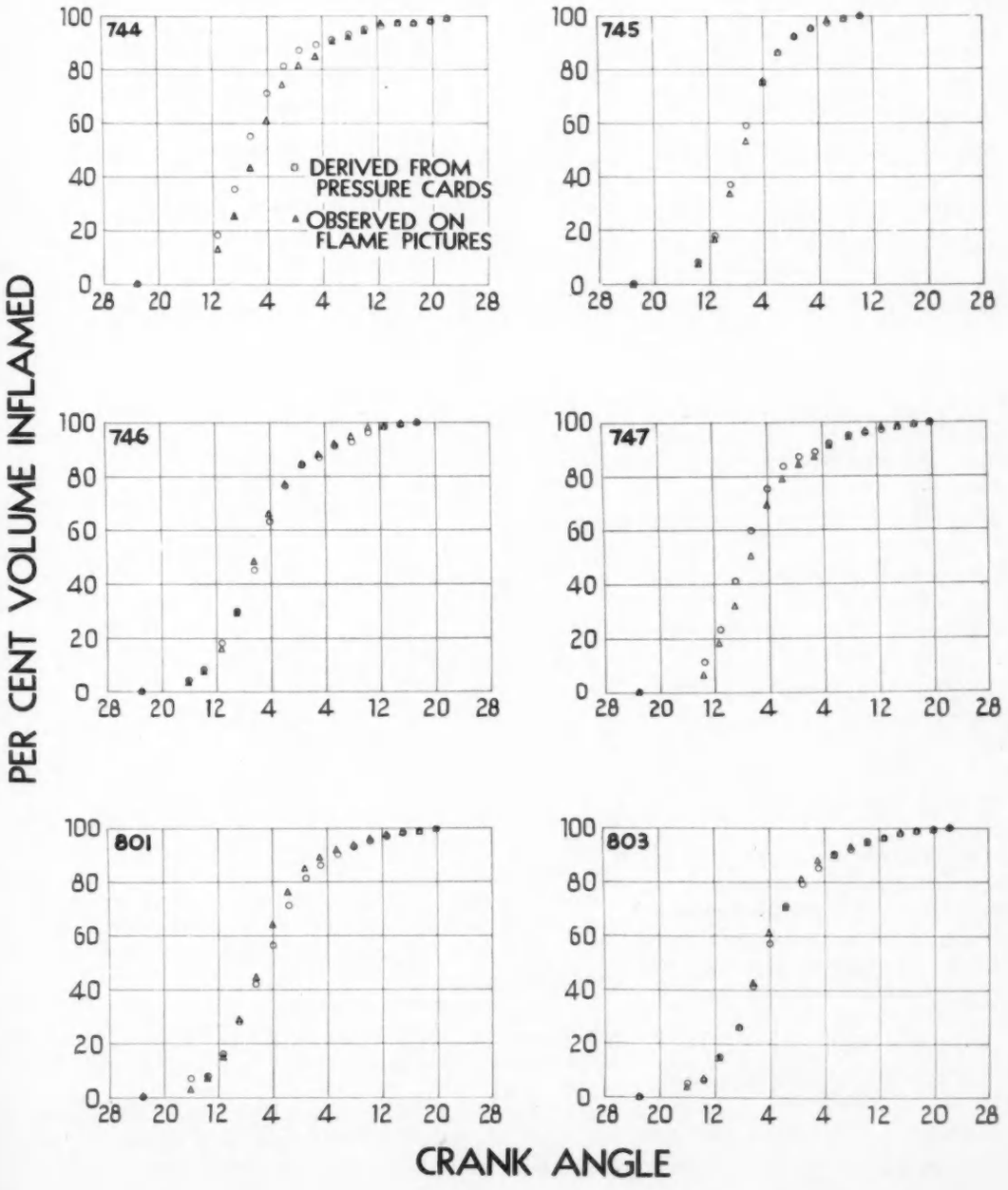


Fig. 20 – Per Cent Volume Inflamed as Observed on Flame Pictures Compared with Values Calculated from Pressure Rise Due to Combustion, Assuming Constant Combustion-Chamber Volume

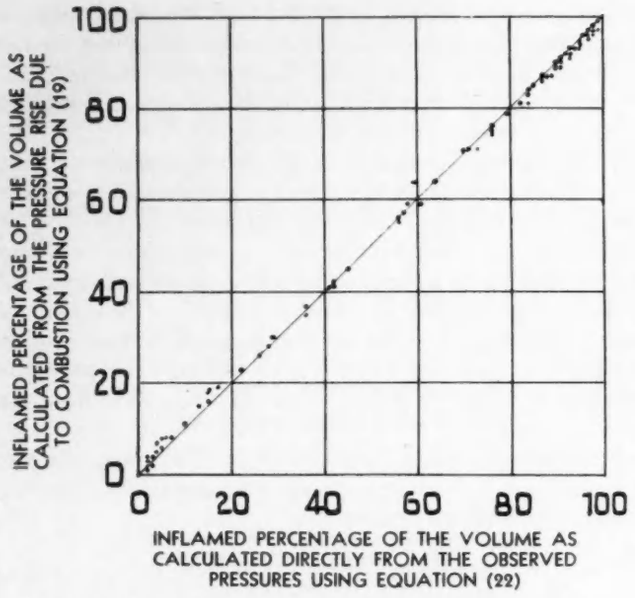


Fig. 21 – Comparison of the Direct and the Constant-Volume Methods of Calculating Inflamed Fractions of Charge from Pressure Cards

Rearranging, solving for fractional volume swept, and simplifying

$$\frac{V_t''}{V_t} = \frac{\dfrac{V_{ti}\,P_{ti}^{1/n}}{V_t\,P_t^{1/n}} - 1}{\dfrac{V_{ti}\,P_{ti}^{1/n}}{V_{tf}\,P_{tf}^{1/n}} - 1} \qquad (21)$$

This may be written in the form

$$\text{Fractional volume swept} = \frac{V_t''}{V_t} = \left(\frac{A}{V_t\,P_t^{1/n}} - 1\right) 1/C \qquad (22)$$

where A and C are constants for a given explosion and are equal to $V_{ti}\,P_{ti}^{1/n}$ and $\dfrac{V_{ti}\,P_{ti}^{1/n}}{V_{tf}\,P_{tf}^{1/n}} - 1$, respectively.

Equation (22) thus expresses fractional volume swept by flame in terms of total volumes of the combustion space and corresponding pressures observed during the combustion period.

In Fig. 21, the values of per cent volume inflamed as calculated for the six explosions with Equation (22) are compared with the values of per cent volume inflamed as calculated with Equation (19). The comparatively small deviation of the points from the 45-deg. line indicates that the inherent differences between the two methods of computing fractional volumes of inflamed charge lie well within the limits of the errors in the measurements of the observed quantities, namely, the engine pressures and corresponding volumes of the combustion space.

On the basis of the same assumptions that were used for deriving Equation (22), a similar type of equation may be developed for computing the fractions of mass inflamed from a pressure card. If the volume of inflamed charge, V''_t, is eliminated from the simultaneous Equations, (4) and (7), and the resulting equation is solved for fractional mass of inflamed charge, we obtain

$$\text{Fractional mass inflamed} = \frac{P_t^{1/n}\,V_t - P_{ti}^{1/n}\,V_{ti}}{P_{tf}^{1/n}\,V_{tf} - P_{ti}^{1/n}\,V_{ti}} \qquad (23)$$

With Equation (23) the fractional mass of inflamed charge may be estimated directly from the total volumes of the combustion space and corresponding pressures observed during the combustion period.

General Discussion

Interest in this work rests chiefly on the possibility that the results may help to clarify the interpretation of engine-pressure measurements in terms of the combustion process. A prime requisite for deducing accurate information about the progress of combustion from pressure-volume data is the recording of accurate pressure cards. But, on the basis of the present trends in indicator development, it is believed that the accuracy and dependability of engine indicating instruments will continue to improve and that their use in engine development will become more and more prevalent.

In the analysis of the data presented in this paper, relationships are set up between the four principal variables; namely, the pressure, the total volume of the combustion-chamber, the inflamed volume, and the burned fraction of the mass. Two general methods of attack are suggested:

(1) The observed quantities may be dealt with directly. Such an approach to the problem is embodied in Equations (4) and (7) and in the equations derived therefrom (Equations (22) and (23)).

(2) The problem may be reduced to conditions of constant combustion-chamber volume. As a first step in this latter plan, the effects of piston motion are eliminated from the observed pressure cards. Once this step is taken, the relationship between the four variables listed above becomes comparatively simple providing the heat losses to the walls, and the differences between the polytropic exponents for the inflamed and non-inflamed portions of the charge may be neglected.

The introduction of per cent mass burned into the discussion not only aids in understanding the relationships between the observed quantities but also furnishes a means of studying the effect of engine variables upon the relative mass rates of reaction. These latter data are of considerable importance because of the fact shown herein that the rate of pressure rise is a function of the mass rate of reaction and of the volume changes produced by piston motion. The effects of engine variables such as mixture ratio, engine speed, and spark position on rates of chemical transformation throughout engine explosions should supplement the thermodynamic treatments of the combustion portion of the engine cycle. Indeed, it is precisely at this point that the application of thermodynamics to the engine combustion process has fallen down; thermodynamics takes cognizance only of the initial and final states of the system under consideration and pays little attention to the velocity or mechanism of the chemical reactions involved. Hence, any effect of engine variables upon the rate of transformation during the combustion period is overlooked frequently in thermodynamic studies.

In describing the combustion process in this paper, the concept of flame velocity has been avoided purposely. On the other hand, many writers feel that this quantity has attractive possibilities because the velocity of flame propagation as measured in stationary flames and in quiescent mixtures in bombs appears to a function of, and thus predictable from the nature and state of, the reactants. But an effort to use the concept of flame velocities in analyzing engine data meets with two principal difficulties:

(1) An examination of the flame pictures shows that the velocity of the flame movement away from the spark-plug varies with the direction of the line along which the velocity is measured.

(2) The flame speed is influenced very markedly by engine speed possibly as a result of turbulence. Since little or no quantitative information about the effects of turbulence on the

APPENDIX

(Table 2, *below*, and Fig. 22, *following page*)

Table 2 – Pressures and Inflamed Volumes Observed in Six Different Engine Explosions

Frame Number	Crank Angle	Combustion Chamber Volume cu. in.	Explosion 747 Observed Pressure lbs. per sq. in.	Explosion 747 Per Cent Volume Inflamed	Explosion 744 Observed Pressure lbs. per sq. in.	Explosion 744 Per Cent Volume Inflamed	Explosion 745 Observed Pressure lbs. per sq. in.	Explosion 745 Per Cent Volume Inflamed	Explosion 746 Observed Pressure lbs. per sq. in.	Explosion 746 Per Cent Volume Inflamed
Ign.	-25°	10.15	82	0.0	86	0.0	83	0.0	85	0.0
3	-23.2°	9.9			89		84		87	
4	-20.8°	9.6	89		92		87		90	
5	-18.4°	9.3	93		96		91		96	
6	-16.0°	9.1	97		101		96	3.0	100	3.4
7	-13.6°	8.9	107	6	109		104	7.9	106	8.3
8	-11.2°	8.7	123	18	122	13.3	116	17.1	119	15.7
9	- 8.8°	8.6	155	32	150	25.1	146	32.6	139	29.5
10	- 6.4°	8.5	197	50	192	43.3	192	52.9	168	47.5
11	- 4.0°	8.4	249	69	245	60.9	251	74.8	216	65.7
12	- 1.6°	8.4	288	79	289	74.3	300	86.1	262	77.1
13	+ 0.8°	8.4	309	84	317	81.8	334	91.8	300	85.8
14	+ 3.2°	8.4	319	87	327	85.4	353	95.1	318	88.0
15	+ 5.6°	8.4	334	91	338	89.5	364	97.9	334	91.6
16	+ 8.0°	8.5	344	94	344	92.1	369	99.3	344	95.2
17	+10.4°	8.7	348	97	347	94.1	370	100	351	97.6
18	+12.8°	8.8	346	98	346	96.2	360	100	358	99.1
19	+15.2°	9.0	340	99	340	97.0	349		355	99.4
20	+17.6°	9.3	335	99.4	330	97.9	335		347	99.8
21	+20.0°	9.5	327	99.8	323	98.8	324		338	99.9
22	+22.4°	9.8	319	100	313	99.7	311		324	100
23	+24.8°	10.1	302	100	301	100	298		312	
24	+27.2°	10.5	291	100	291	100	286		299	

Frame Number	Crank Angle	Total Combustion Chamber Volume	Explosion 801 Observed Pressure lbs. per sq. in.	Explosion 801 Per Cent Volume Inflamed	Explosion 803 Observed Pressure lbs. per sq. in.	Explosion 803 Per Cent Volume Inflamed
Ign.	-25°	10.05	82	0.0	82	0.0
3	-22.2°	9.7	87		87	
4	-19.9°	9.4	92		91	
5	-17.4°	9.1	97		95	1.0
6	-15.0°	8.9	101	2.8	100	3.0
7	-12.6°	8.7	106	7.0	104	7.2
8	-10.2°	8.6	116	15.5	113	14.6
9	- 7.8°	8.5	134	29.1	130	26.2
10	- 5.4°	8.4	159	44.4	156	42.8
11	- 3.0°	8.3	191	64.1	192	60.6
12	- 0.6°	8.3	239	76.3	234	70.4
13	+ 1.8°	8.3	286	85.2	267	81.1
14	+ 4.2°	8.3	310	89.3	295	87.8
15	+ 6.6°	8.4	329	92.2	320	90.2
16	+ 9.0°	8.5	340	94.3	329	95.0
17	+11.4°	8.7	349	96	334	94.8
18	+13.8°	8.9	352	98	335	96.2
19	+16.2°	9.1	353	99.6	335	97.5
20	+18.6°	9.3	345	100	329	98.8
21	+21.0°	9.6	337		322	99.2
22	+23.4°	9.9	325		311	99.9
23	+25.8°	10.2	310		298	100
24	+28.2°	10.6	296		286	

flame velocities is available at present, the introduction of the concept of flame velocity into the analysis appears to be quite hopeless. Thus, it is preferable, on the basis of present information at least, to describe the progress of combustion in terms of inflamed volume or in terms of fractional mass burned.

Although neglecting the direction of, and the velocity of the flame propagation does permit the establishment of a comparatively close relationship between the burned fraction of charge and pressure developed, this procedure suffers one serious disadvantage. In studying combustion in engine explosions for which no flame pictures are available, it is not possible to determine the location of the inflamed fraction of the charge in the combustion space. Making such a determination by spherical sectioning of the combustion space is not trustworthy because of the irregular form of the flame fronts. Thus it appears that, if the location of the inflamed portion of the charge must be determined, it will be necessary to supplement the analysis of the pressure cards with direct observations. For making such observations in commercial engines, the method of ionized gaps appears to be the most promising.

It cannot be emphasized too strongly that this work is by no means completed. The data presented in this paper were all obtained with one air-fuel mixture at a comparatively low engine speed while operating with the throttle full open under non-knocking conditions. Other engine conditions are being investigated, and further efforts are being made to improve the reliability of the pressure measurements.

Acknowledgments

The writers take this opportunity to express their appreciation to C. F. Kettering and T. A. Boyd for making the publication of this work possible and to Dr. V. C. Smith for very valuable assistance in developing suitable instrumentation.

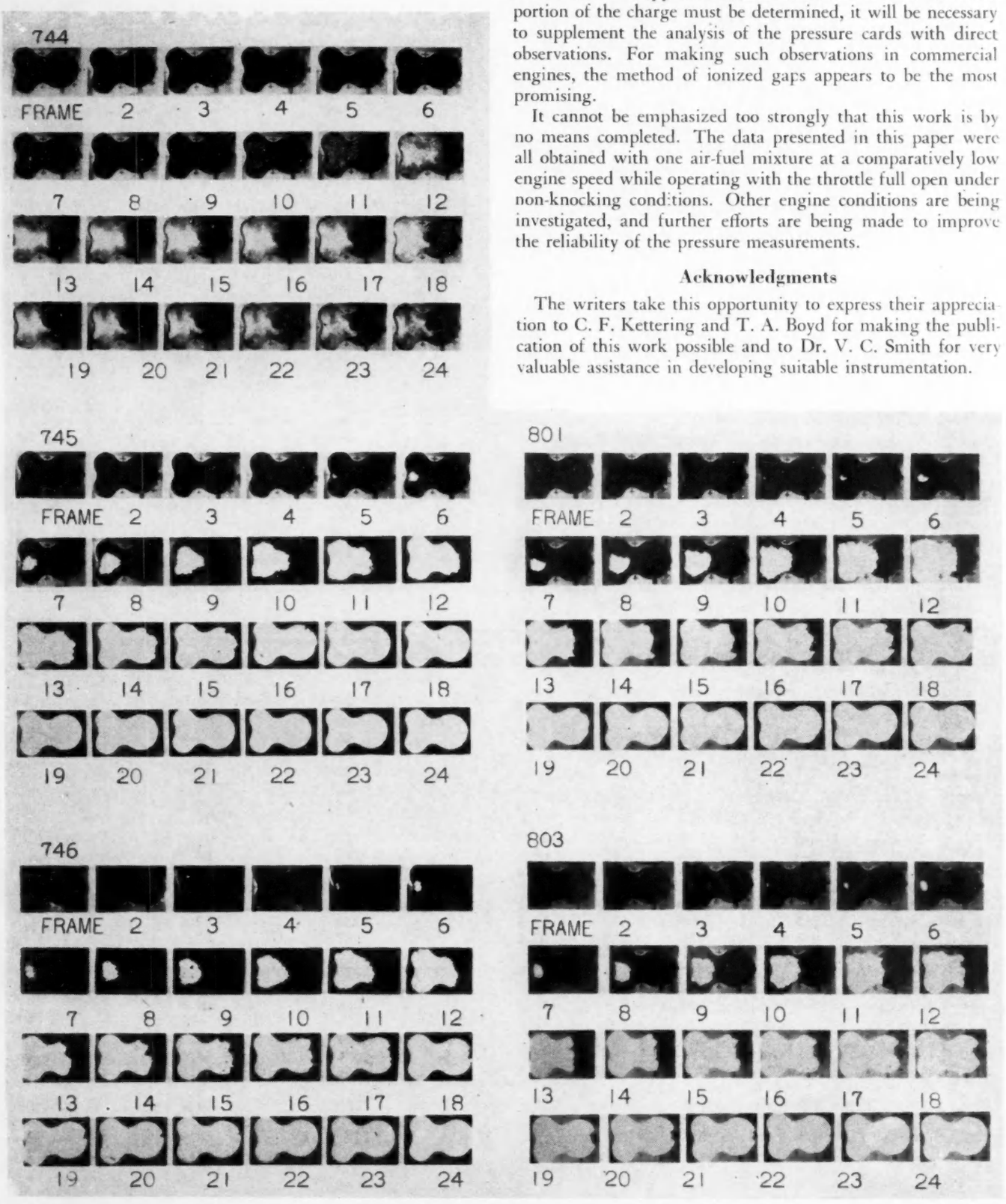


Fig. 22 (*Appendix*) – Pictures Showing the Flame Propagation in Five Non-Knocking Explosions Which Are Discussed in this Paper

The Accessory-Drive Problem of Aircraft Engines

By R. P. Lansing
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A BRIEF history of the subject is given, tracing the growth of accessory drives from their initial being as magneto drives down to the present stage which embraces such items as automatic pilot pumps, de-icer compressors, and propeller governors. This review indicates that the present groups may well be divided into two parts: one, intimately associated with the operation of the engine, the other with that of the plane. The difficulties facing the engine designer, the installation engineer, and the operator are outlined to illustrate the necessity of a new approach to this problem.

In view of the present general interest in auxiliary powerplants, a short description of such an engine is included. The paper terminates by mentioning the present tendencies in the field and classifying the various means of accessory drive with the size of plane with which it probably will be employed.

AIRCRAFT engineers have long recognized the fact that accessory drives present a serious engineering problem. To be sure, in the infancy of the industry, the problem was not particularly troublesome but, since within the past few years drives have had to develop with almost incredible speed in order to keep pace with engine improvements, the problem that they have created is now one of well nigh major proportions. The industry now has reached that stage where continued development along the lines heretofore followed is hardly possible, and it may well pause to consider other methods of meeting the accessory-drive problems and concentrate its efforts on those having the greatest promise.

Before taking up the analysis of the topic and describing the program contemplated for its solution, we will review briefly its history.

The early designers of airplanes and their engines had but little to worry about in so far as accessory drives were concerned, in comparison with the problem of the powerplant engineer of today. Probably, the first accessory-drive problem was the magneto and, later, the tachometer drive. There the matter rested until the World War which, with its stimulus to aircraft development, brought a host of new problems. Then almost overnight, as engines developed rapidly in size and the first demands of radio and armament manifested themselves, the engine designer had to find room for a starter, a generator, and machine-gun synchronizers.

At the same time, the necessity for interchangeability and quantity production made the establishment of standardized specifications for mounting flanges and accessory drives mandatory. Naturally, this work was done by the Army and Navy Engineering Divisions in conjunction with the engine and accessory manufacturers. It is interesting to note that these specifications, which took final form about 1920 with the exception of minor changes, and the gradual discarding of those of smaller sizes, will be found in our present-day standards. In the ten-year period following the war, the number and size of the accessories grew very little, probably because the industry was using, for the most part, the war surplus of engines and associated equipment, and only limited commercial flying was being done.

With the accomplishment of the early transatlantic flights and the expansion of air mail into both mail and passenger service, the picture began to change rapidly. Landing, running and cabin lights, and "two-way" radio communication required larger generators. Faster schedules and heavier loads demanded larger engines and new problems became evident. Whereas before this time, our strength problems had been mainly those of reconciling light weight with sufficient strength, vibration now changed from a minor to a major factor. Increasingly severe vibration inevitably led to the abandonment of all but the largest of our old standards for accessory drives, and even these standards remained in use only because of constant improvement in design and use of new materials.

By 1935, supercharged engines were finding extensive commercial use and supercharger and manifold-pressure regulators were developed. Wing de-icers, necessitating the use of engine-driven compressors, became standard winter equipment. The Department of Commerce, instituting regulations concerning instrument operation, brought the vacuum-pump into prominence, and the artificial-horizon and gyro-compass created a demand for vacuum that no longer could be obtained satisfactorily from venturi-tubes mounted in the wind stream. The result was yet another load for the engine.

The automatic-pilot added an oil pump, and hydraulic operation of the landing gear and flaps whenever specified made still another oil pump necessary. Finally, the governor drive of the constant-speed propeller, and a drive for an alternator for remote instrument operation added to the multiplicity of loads on the engine. The casual observer might well wonder where the power for the propeller was to be found.

Until the present the engine designer has attempted to

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meet these drive requirements by means of new rear sections on the engine crankcase. However, it has become apparent to him and to the rest of the industry that it is an impossible task to provide sufficient power take-offs of adequate strength and spacing and speed ratios to meet all the existing demands. In this connection, engine problems alone are sufficiently difficult without the complications of accessory drives. Yet, the additional drives not only complicate the crankcase but create a complicated torsional-vibration system and add a continuous power drain upon the engine. This drain, although not large, is looked upon with disfavor by the military engineers who feel that, under combat conditions, the last iota of power at a crucial moment may stand between success or failure. Some quick means of temporarily de-clutching the non-essential accessory load would take care of this condition.

We are getting along today by compromising between the equipment we would like to carry and what we can carry, and by assisting inadequately sized mounting flanges by external bracing. The space between the fire wall and engine of a modern airplane is a virtual maze of accessories with their associated pipe lines, cables, operating rods, oil separators, wires, and so on. As may be expected, the position of the powerplant engineer is far from being a happy one. He must not only find room for all of these items but must label them, ventilate some and keep others warm, or cool them as may be required. Also he is faced with the necessity of securing accessories having a wide operating range, for the same gear ratios are used in the rear sections of many engines differing in cruising speed from 1650 to 2200 r.p.m. This range has been increased downward by the constant-speed propeller which permits full throttle power at low propeller speeds. The position of the operators' personnel who must inspect and service them and be responsible for their functioning can well be imagined. The position of these people is complicated further because, under existing conditions, the accessories must be serviced when the engine is overhauled. There is no reason to believe that the operating endurance of the generator, the vacuum pump or the hydraulic power pump is identical with that of the engine and, depending upon the operating organization, they should be overhauled every 300 to 600 hr. Also, since some of the engine-driven accessories come under the jurisdiction of other departments, an overlapping of authority and interests exists, as in the case of generators and radio communications. Independence of servicing and ground-testing is much to be desired.

Analysis of Problem

To approach an analysis of the problem, let us first determine what are the principal accessories we must take into consideration. We can divide them into two groups: first, those intimately connected with the functioning of the individual engine (which, for the purpose of this paper, we will refer to as the engine accessories, or first group); second, those pertaining to the operation of the airplane as a whole (which we will refer to as the plane accessory, or second group).

They are as follows:

Engine Accessory or First Group

Two magneto drives	Lubricating oil pump	Supercharger regulator
Fuel pump	Scavenger oil pump	Tachometer drive
Starter	Propeller-governor drive	Machine-gun synchronizer

Plane Accessory or Second Group

Generator, d.c. or a.c. for lights, starting radio, cooking, heating, and so on
Generator, a.c. for remote-reading-instrument operation, and so on

Automatic pilot pump	Fuel transfer pump
De-icer compressor	Instrument vacuum pump

Hydraulic power oil pump

Inspection of the two groups brings out the following points:

(1) It is evident that the accessory drives for Group 1 are connected so closely with the operation of the individual engine that they must be located on the engine and duplicated for each engine and that no liberties may be taken with them except, possibly, the propeller-governor drive.

(2) With the exception of the magneto and starter, the bulkier and heavier items fall into the second group.

(3) Likewise, those units requiring large amounts of power are in the second group.

(4) In the case of multiengined planes, in order to assure operation in case of engine failure, the second group would need to be duplicated on two or more engines.

(5) In the second group would be found those units requiring sub-accessories that would be mounted best adjacent to the unit in question; for example, the generators next to their radio filters and voltage regulators and the de-icer air compressor next to its associated relief valves and oil separators.

The preceding conditions have suggested the idea of removing the units of the second group from the engine and, by means of hydraulic or electric motors, operating them as individual units. Power would be supplied by a pump or generator driven either by the main engines or by an auxiliary engine. This method, with modifications, already has received some consideration in certain quarters.

Another possible method of driving this second group would be to mount them on a separate accessory drive unit or "gear box," which may be driven in many ways. Some possible ways are:

(1) Flexible shaft capable of transmitting 30 hp. driven by the airplane engine.

(2) A hydraulic motor driven by a large hydraulic pump on the engine.

(3) An exhaust-driven turbine.

(4) An electric motor driven by large generator on the engine.

(5) Auxiliary gasoline engine.

Some of the advantages to be secured by removing the accessories of the second group from the engine would be to provide better accessibility forward of the firewall, providing greater freedom for engine adjustment, and to eliminate the necessity of subjecting accessory units to heat of the engine, thereby increasing their life.

Units would not be subjected to engine vibration and, conversely, the engine would not be subjected to increased vibration and wear caused by accessory units.

Accessories could be provided with desired mounting flanges and drives that now are fitted with compromised flanges in order to mount them on existing pads designed originally for other purposes. Likewise the accessories themselves may be improved, provided their designer is not restricted by the limited space on the rear section of the engine.

Freedom from the limited speeds available on the engine would permit accessory designs of greater reliability and possibly lighter weight.

The accessories may be mounted where they really can be serviced, adjusted, and removed when necessary, and the present Christmas-tree appearance of the engine can be abolished.

At this point it may be well to consider what the accessory-drive unit of the "gear-box" type may be like. Existing specifications call for total power capacity of 20 hp. with the likelihood of being superseded by a 30-hp. unit. Although various manufacturers have built gear boxes, they have not yet found widespread use.

Specifications for the 20-hp. gear box indicate that it should consist of a main section and an auxiliary section, with provisions for driving the main section alone should only a few drives be required. Each section ordinarily would handle four accessory drives, and the main section would contain a complete lubricating system. This system would consist of a pressure-feed pump supplying approximately 50 lb. per sq. in. pressure and a suitable oil tank. The lubricating system would need to supply not only the gear box but all the accessories mounted thereon, such as the vacuum pump. The main section would carry three 5-hp. accessory drives and one 2.5-hp. drive and all, with the exception of one, would drive normally at 3000 r.p.m. The remainder, a 5-hp. drive, would operate at about 2000 r.p.m. These drives would be used for generators, hydraulic pumps, vacuum instrument pumps, and de-icing pumps. The auxiliary section would have similar drives but would carry an extra fuel pump in place of the hydraulic pump.

The design of such gear boxes is not the simple problem that it may first appear. By specifications, the complete weight of the gear box is limited to 40 lb. In addition to being sufficiently rigid in order to hold its driving gears in absolute alignment and rigidly support the accessories, it should incorporate a suitable disconnect clutch between its flexible shaft and itself so that it may be operated while running. The clutch and the lubricating system should fall within the above-mentioned 40-lb. allowance.

Although most designs to date have utilized ball bearings, it is probable that a weight advantage may be obtained by reverting to plain pressure-lubricated bearings, and the weight thus saved may be used more advantageously for rigidity of the frame work. This possibility is quite feasible in view of the fact that the necessary lubricating system must be supplied in any event.

Ventilation and cooling of the gear-box unit must be given consideration for, not only will the box be called upon to radiate its own pumping and transmission losses which in the case of a 30-hp. box may be equivalent to 2.5 or 3 kw. of heating, but it must assist some of the accessories to cool as well. How much of a problem this would be in service would depend upon its location in the plane and the range of ambient temperatures encountered.

Vibration, set up within the gear box itself, must be guarded against by proper gear-tooth form, balance of rotating parts and, where possible, by dampening devices. In addition, attention must be given to minimizing noise by avoiding resonance chambers and side panels. Unfortunately, we cannot borrow the acoustical treatment which the automobile maker secures by using asphaltum compounds, as such treatment would prevent necessary heat conduction and thereby impair cooling.

To drive such a gear box, the larger engine companies have, or will provide, a power take-off suitable for 30 hp. on engines up to about 1000 hp. In the interest of minimizing vibration effects it would be well to have this drive located as close to the center of the engine as possible.

In those cases where the gear box is to be driven directly by the engine, a tubular steel shaft properly fitted with universal joints is used. This drive introduces still another problem and weight complications, as some form of torsional-vibration coupling or damper must be interposed between the gear box and the engine. If it were available, a large flexible shaft of the speedometer type probably could be used to advantage, as we believe this shaft will absorb most of the power impulses. However, a flexible shaft having reasonable weight and bulk, capable both of transmitting 30 hp. at 3000 r.p.m. and of standing the service is, so far as we know, nonexistent.

Hydraulic drive of the gear box by means of a hydraulic motor has been proposed and, at first glance, it appears ideal, particularly from the standpoint of simplicity.

Let us examine briefly a few aspects of the hydraulic drive. We are restricted, at the present stage of the art, to a maximum oil pressure of about 1500 lb. per sq. in. Although higher pressure can be developed, the weight of pipe lines, fittings and hydraulic pump and motor housings increases to a prohibitive amount. Using this pressure of 1500 lb. per sq. in. as a working maximum, we would need, for 30 hp., to supply the hydraulic motor with fluid at the rate of 35 gal. per min. This flow, if unreasonable losses were to be avoided, would require a pipe line of about 1-in. diameter, provided the gear box could be located not more than 5 ft. from the engine and that low-viscosity fluid is used. Although the design of such pumps and motors can be made without difficulty, the actual operation of such a system offers several obstacles. It is believed that the pump and motor combinations can be built to weigh under 40 lb., although such a figure compares unfavorably with the weight of a mechanical driveshaft. Another weight factor that must be taken into consideration is the weight of the oil which, incidentally, will be subjected to considerable heating so that it is definite that an oil cooler would be necessary unless a large oil reservoir is employed.

Probably the most troublesome detail would be the flexible tubing required between the flexibly mounted engine and the rigid tubing secured to the ship's structure. Flexible tubing under high pressures tends to become very rigid, and failure at connection points may be a problem.

On the other hand, the hydraulic system of drive permits a great deal of flexibility in the location of the gear box, provided tubing of adequate size is used. Thus the gear box can be located either adjacent to the flight engineer or at some point close to the center of load distribution. Also, all other factors being identical, the hydraulic system will compare very favorably from a weight and efficiency standpoint with most other methods of remote gear-box drive.

Exhaust-Driven Turbine

Considerable attention has been attracted by the suggestion of using an exhaust-driven turbine to drive a gear box. Although we are not well acquainted with what has been done on this type of drive, it will be realized that an enormous amount of energy is being lost constantly through the exhaust to the atmosphere and, if a sufficient amount can be recovered without too great a cost in weight, the device should find a wide field of application.

We previously have mentioned the use of a large generator mounted on the engine and primarily utilized to supply a motor on the gear box. Such an arrangement promises many desirable features such as constant-speed drive of gear box, ease of connections, freedom of location, and great flexibility of control. The most serious drawback to this method appears to be excess weight and space requirements on the engine and somewhat lower than normal efficiency.

The major saving in weight in motor or generator design is obtained by utilizing very high operating speeds rather than by any other single factor. This method could be used to a limited extent in the case of the motor by incorporating proper reduction gears in the gear box. However, the space and weight limitations about the engine would hardly permit "gearing-up" the generator. Either a.c. or d.c. power could be used for this purpose, and it is probable that a.c. power would lend itself best to high-speed operation.

All the methods of gear-box drive so far discussed have had one great limitation, that is, none permits ground operation of the gear box and its accessories when the airplane engines are either throttled or at rest. This operating characteristic is very desirable for operation of radio, cabin lighting and heating, fuel transfer, refueling, and many military functions.

To obtain ground operation under these conditions, power must be supplied either from an external source or by a separate prime mover. It is this point that counts heavily in favor of an auxiliary powerplant of the gasoline-engine type. This practice is by no means new, having been utilized for many years, although with limited power, by our own Navy and, more recently, by foreign operators. At the present time a French concern is offering a 10-hp. plant that, besides driving a fairly large generator, drives oil and fuel pumps, air compressors, and so on.

Many of you know from the current newspapers and newsmagazines that more than one new military or naval plane recently has been fitted with an auxiliary power system but, since they are all secret projects, nothing can be said of them in this paper. However, along somewhat different lines, some of the operating companies have pooled their interests to secure the development of a super-transport. After careful investigation and study of the various proposals made, it was decided to adopt two four-cylinder engines of about 20 hp. each for driving the gear boxes. With the thought that you may be interested in some of the features of this engine, it will be described briefly later in this paper.

Advantages of Auxiliary Engine

Returning for the moment to the broad considerations of the auxiliary engine, we note that it has the following advantages:

(1) Operation is entirely independent of main engines.

(2) It may be built to any suitable horsepower and, accordingly, the gear box is not limited to the power that may be taken from any one engine.

(3) Optimum speed for minimum weight of engine and accessories may be selected.

(4) Servicing of engine, gear box, and accessories may be done independent of main engine.

(5) Full power of main engines is always available.

On the other hand, it must be admitted that initial cost and operating expense will be higher than in most other means of gear-box drive. Fuel and oil must be carried, although they are not necessarily large items. As before stated and again stressed, the outstanding advantage of the gasoline auxiliary is operation independent of the main engine.

In all the methods so far considered for driving the second accessory group, the use of a gear box as a focus point of the auxiliaries has been assumed. In many cases it will be conceded that a greater advantage would be gained if, for example, the automatic pilot pump could be located close to the automatic pilot; the de-icing compressor, adjacent to its distributor valve, and so on. To this end it has been suggested that each auxiliary be driven independently by its own motor which would derive its power from a central generator or hydraulic pump on one or more of the main engines.

We believe that, except in the larger ships, such a system cannot be used to advantage, simply because the total weight of individual driving units, together with their mounts, gear boxes, supply lines, and so on, greatly exceeds the weight of the central gear-box system. It must be pointed out that, in at least two ships, partial use has been made of such a system, in one case, hydraulic drive being used and electrical drive in the other. Except for military reasons no advantage exists in such a system if most of the items of the accessory group can be located best at some central point.

It is evident that none of the foregoing systems may be selected on its individual merits, but must be chosen with due regard to the type and size of the plane, and with consideration for the accessories that are specified. A huge seaplane for transpacific service naturally would have accessory equipment differing widely from that of a large military bomber or say a transcontinental passenger ship, and a different type of powerplant may be best adapted to each class.

The accessory-drive problem may be expected to grow in proportion to the size of the ship, rather than to the size of the engine which, until recently, has been roughly the case. We believe that the single-engined planes powered with engines of over 600 hp. will, for the most part, be confined to military and sport uses and, with some exceptions in the former class, they will depend upon the engine crankcase for accessory drives.

It is expected that the larger twin-engined planes and the smaller four-engined planes will eventually drive their accessories from a gear box driven from the main engines either mechanically or by one of the other methods mentioned.

The larger ships of over 30,000 lb. undoubtedly will use either auxiliary powerplants for gear-box drives or operate some form of generator for independent drive of accessories. At the present time the tendency is to use a combination of gear box and remote electrical drive. This arrangement is accomplished by diverting about one-half of the available power to accessories mounted on the gear box, such as hydraulic pumps and air compressors, and the remainder is used for a large generator, likewise mounted on the gear box. This latter power is used for general lighting, radio and isolated power applications such as starters, fuel transfer pumps, small ventilating fans, and so on.

In one case, remote electrical drive of a hydraulic pump has been used in turn to drive hydraulic motors for retracting mechanisms. This method was used both to secure greater speed control than otherwise would be possible and to locate the equipment to the best advantage.

The auxiliary engine previously mentioned has been developed for a large transport now under construction. The reliability and durability characteristics of the standard aircraft engine were sought for in the design of this engine even to the extent that light weight and compactness were sacrificed temporarily. The fact that the engine was built to fill an immediate need also influenced the selection of fundamental design. It is a four-cylinder double-opposed engine, Prestone-cooled and operated continuously at 4000 r.p.m. No attempt has been made to supercharge the engine for altitude, but it purposely has been made oversize so, at altitude, sufficient power still will be available. Automatic control of carburetor mixture for altitude has been provided. The double-opposed cylinder construction was adopted as the best compromise between minimum overall length and balance. This compromise was necessary because of the restricted space available, and because specifications demanded that the engines be mounted directly behind the fire wall in the inboard-engine nacelles. The engine will develop approximately 30 hp. at sea level although it will be rated in the neighborhood of 22 hp. Aviation practice in design has been incorporated throughout.

In the interests of dependability, dual magneto ignition was specified. The engine crankcase and the gear box are practically integral, provision having been made for a total of seven accessory drives operating at speeds ranging from 2000 to 4000 r.p.m.

Because of the remote location of the engine, all functions

(*Continued on page* 224)

The Outlook on Legal Performance Requirements

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PRESENT interest in ways and means of improving the speed at which many slow-moving vehicles climb grades, springs from a real necessity and an honest desire on the part of all agencies concerned to produce a practical solution.

The United States Bureau of Public Roads recently has undertaken to develop apparatus and a procedure which may be used to secure a large amount of data on current hill-climbing practice. It is planned that this method will be applied shortly through the agencies of several of the State Highway Planning Surveys.

The apparatus and procedure referred to were used during the past summer to secure a limited amount of "trial" data. Both the tests and data are discussed as well as certain plans which are now being formulated to make a special study of both new and used trucks in the dynamometer laboratory of the Motor Transport Division of the Army, and to correlate these data with actual hill-climbing tests.

The purpose of these experiments is to establish the optimum performance which may be expected of modern trucks, to evaluate the rate of decline of grade ability, and to check certain factors used in performance formulas.

WHY do we discuss legal performance requirements? It is because we all recognize that a needlessly large number of motor vehicles travel at such slow speeds as to impede the proper tempo of traffic; it is because we all realize that something must be done about it; and it is because we are all interested in seeing that whatever is done shall represent an intelligent approach to the problem.

Something must be done about it because the presence of these slow-moving vehicles hampers the rapid transportation of people and goods and, thus, seriously interferes with the efficient use of our highway capital plant. Something must be done about it because of the frailty of the patience of the human animal which prefers to take chances with his property, his passengers, and his own skin to staying in line behind a car or truck that either cannot or will not move at a reasonable speed. See Figs. 1, 2, and 3.

To point to the problem is easy – but, to solve it, is exceedingly hard. The solution is difficult because it involves so many interests and so many factors. The manufacturer of equipment, its owner and operator, the public whose interest it serves, and the motorist and the general taxpayer, both of whom help pay for our highway system, are all affected profoundly.

What can be done about it? As we see it, there are four possible lines of attack: we may reduce gradients; we may separate fast and slow traffic either by multilane construction, or by building two separate and distinct routes; we may hold powerplants at their present level and reduce legal loads; we may retain present load restrictions and increase engine power.

Obviously none of these solutions will be adopted to the exclusion of the others. An intelligent approach demands that the economics and safety of highway transport be studied as they are influenced by the application to the problem of various combinations of these possibilities.

Performance Study Initiated

Working in close harmony with the Interstate Commerce Commission and with the automotive industry through such bodies as the Truck Sub-Committee of the Automobile Manufacturers Association, of which David C. Fenner is chairman, the Bureau of Public Roads recently undertook an investigation which, it is hoped, will furnish some of the information upon which rational decisions and intelligent regulations may be based.

It was thought that a picture of current hill-climbing practice was of fundamental importance and, accordingly, our first efforts during the past summer were directed to the development of apparatus and procedure by which we could get such a picture.

A uniform 6 per cent grade was selected on U. S. 40 near Baltimore. This hill was located 4.5 miles from a State Roads pit scale, and its length of approximately 2000 ft. was sufficient to decelerate practically all trucks to a constant climbing or crawl speed.

Procedure at Weighing Station

The procedure may be divided into two parts. That part involving the weighing station consisted of securing detailed information which might affect the vehicle's ability to negotiate the test hill. Gross weight, drive-wheel tire sizes, governed speed, total mileage traveled, and mileage since last

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Fig. 1 – Will He Make It?



Fig. 2 – Looks Like He Did

engine replacement or overhaul were tabulated, as were identification data such as license numbers, make, model, year, engine and serial number, and so on. Fig. 4 shows operations at the weighing station. Special rear-axle assemblies also were noted. When all the information was gathered, a red card was attached near the right end of the front bumper. This card, visible from a considerable distance through binoculars, served to warn the speed-recording crew on the hill of the approach of a test vehicle. This method was necessary because great care was exercised to keep truck drivers from knowing that their speed was being observed. Therefore, they could not be slowed down or stopped as they approached the hill.

Speed Recorded on Test Hill

When the vehicle reached the hill, its speed was measured by recording the instant it ran over each of twelve equidistant electric road strips laid on the road surface. These switches were located 170 ft. apart from bottom to top of hill. When the vehicle was seen to approach the hill, the dial switch (Fig. 5) was turned so as to "cock" the first road switch which was then the only one of the twelve capable of recording. When the truck wheels ran over it, they pressed together two strips of spring steel which under normal conditions were insulated from each other by a series of rubber insulators. See Figs. 6 and 7. The contact permitted current to flow from a storage battery through a wire connected to one spring element of the road switch and through the second element, back through a common return wire to energize a solenoid. See Fig. 5. The solenoid actuated a stylus of a chronograph thus marking the position of the truck on the hill upon the moving chronograph tape. The set-up and operation of the chronograph, clock, and instrument box are shown in Figs. 8 and 9. At the same time another solenoid-actuated stylus recorded the 10-sec. time beats of an electrically connected clock on the same tape. After the vehicle had passed over the first switch, the chronograph operator turned the dial switch so as to "cock" the second road switch. By repeating this operation in cadence with the truck's progress up the hill, a complete record of vehicle positions against elapsed time was secured on the tape. At the end of the day's work these chronograph records were turned over to a small office force which analyzed them for vehicle speed by scaling position and time intervals. The vehicle was identified specifically against the weigh-station record by noting its license number as it passed the chronograph station hidden to one side of the road. (Figs. 8 and 9.)

By the procedure just described an average of about 10 vehicles per hr. were observed after all the "bugs" were taken out of the apparatus and after the personnel was familiar with its task.

Desirability of Further Tests

Before discussing some of the data which were secured, specific attention is called to the fact that our job thus far has been one of development and of laying plans for the future. Therefore, the data presented are in no way conclusive, based as they are on a handful of observations which were more or less incidental to the development of method. They are presented here merely to indicate what kind of information possibly may be gathered by a procedure similar to the one used.

Fig. 10 is a frequency-distribution diagram of "crawl" speeds, that is, of minimum sustained climbing speeds to which the 6 per cent grade reduced the vehicles. From such curves we will be able to ascertain what portion of vehicles is pulled down to various speeds by the several steepnesses of test grades contemplated.

Fig. 11 shows a cumulative frequency curve of crawl speeds. Similar information for a variety of grades will help us to see what percentages of vehicles will be affected by the



(Courtesy Washington Post)

Fig. 3 – But He Didn't!

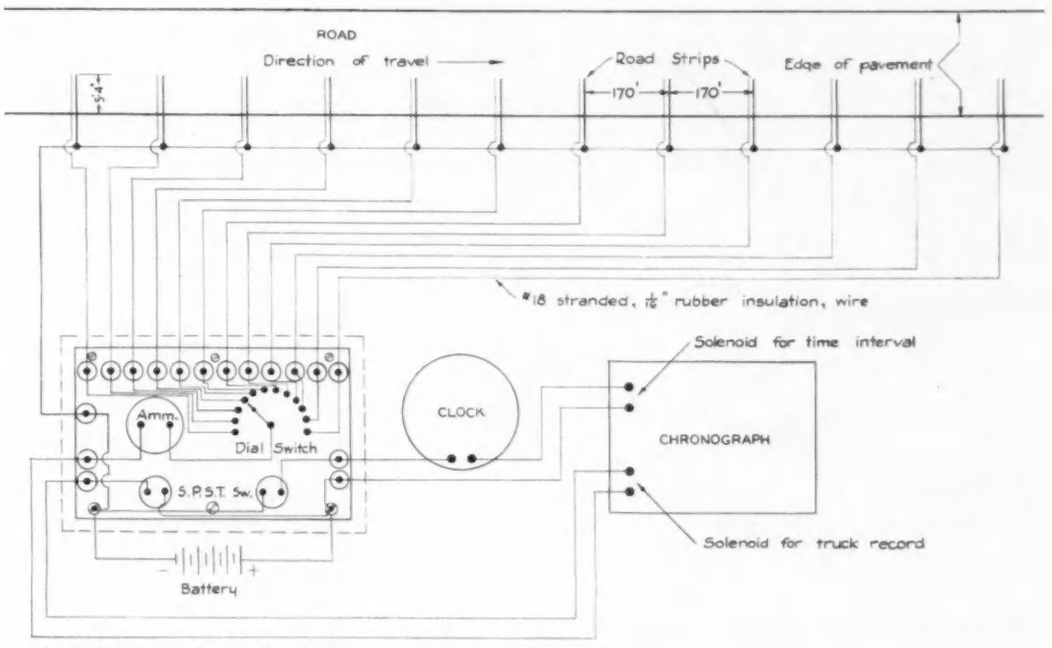


Fig. 5 – Apparatus for Measuring Truck Speeds

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several grade-speed stipulations which may be under consideration for proposed regulations.

Fig. 12 shows the general effect of gross weight upon climbing speeds on the 6 per cent grade. Since the principal variables which must be adjusted satisfactorily are speed, weight, and grade, the usefulness of such data for a number of grades is obvious.

Fig. 13 shows the relative drop in speed as trucks of various



Fig. 4 – Operations at Weighing Station

weight classes ascended the 6 per cent grade. Additional information of this kind is necessary for a variety of grades when consideration is given to grade reduction since the effect of length of hill will play an important part in deciding which categories of hills to reduce.

All of the data just referred to were secured for trucks just as they happened to come along except that we did not attempt to record very lightly loaded ones. It should be noted again that the drivers generally were unaware that their speeds were being recorded.

Special Tests of Low-Mileage Vehicles

The attempt was made to select for special study a few new (1937) trucks which had been run far enough to be broken in but not far enough to show the effect of wear. The mileage range decided upon was between 5000 and 20,000 miles. We were anxious to develop a procedure whereby new trucks could be used to demonstrate the best possible performance for given makes and models. It also was thought that possibly the observed performance might be checked against performance formulas.

It must be admitted that the study of such special trucks was not very successful and, therefore, none of the data which were secured are presented here. We adopted the following procedure:

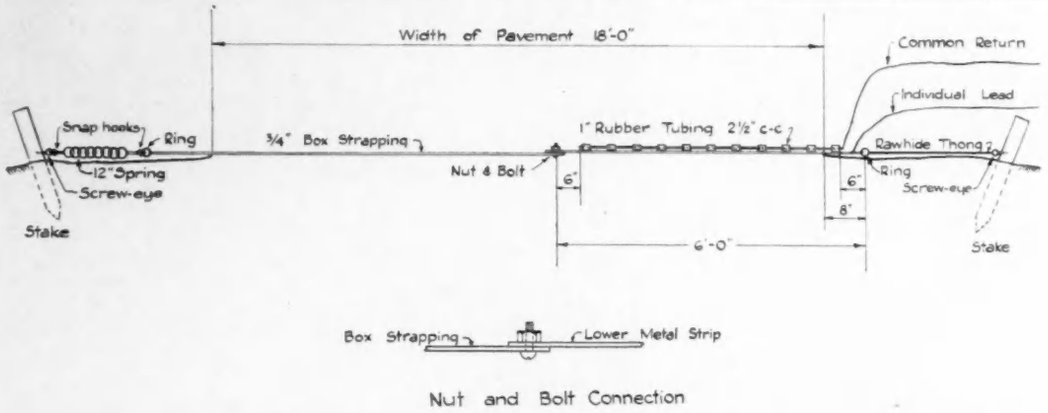
At the scale house all possible information was secured concerning items having an effect upon hill-climbing ability. In addition to the data secured for the ordinary run of trucks, we ascertained the rolling radius of all drive-wheel tires and tried to ascertain the axle ratio. We developed a method for measuring the former distance accurately as shown in Fig. 14 but, in a great majority of cases, we were unable to ascertain the latter figure even though we crawled under the truck with flashlight and wire brush to see if we could discover it on the axle or differential housing. Before the vehicle left the weighing station, it was marked with a special tag, and the driver was informed that, just before he reached the test hill, he would be stopped by a man with a red flag. After our observer had stopped the vehicle and explained to the driver that we wanted him to get over the test hill as fast as he could, he rode in the cab of the vehicle to see that his instructions were carried out. Meanwhile, the speed-recording crew on the hill was made aware of the approach of the vehicle by means of the special tag already mentioned. When the vehicle hit the No. 1 road switch, the observer in the cab started a stop watch and noted the transmission gear in which the truck was at the moment. Subsequent gear changes were timed and, in this way, we later were able to tie in the speeds recorded on the chronograph tape with the actual corresponding transmission-gear ratios.

The actual engine speed corresponding to the recorded road speed was then computed by the well-known formula[1]:

$$\text{r.p.m.} = \frac{168 \times R \times S}{r}$$

The published engine torque corresponding to the known

[1] See S.A.E. JOURNAL, March, 1933, p. 16: "A Method of Rating Commercial Trucks."



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Fig. 6 – Road Strip Tie-Down

engine speed was then inserted in the equally well-known formula[1]:

$$\text{G.V.W.} = \frac{T \times 12 \times E \times R}{r(G/100 + f)}$$

in which the coefficient of rolling resistance, f, was taken as 30 lb. per gross ton. The gross vehicle weight computed in this manner was compared with the known gross weight.

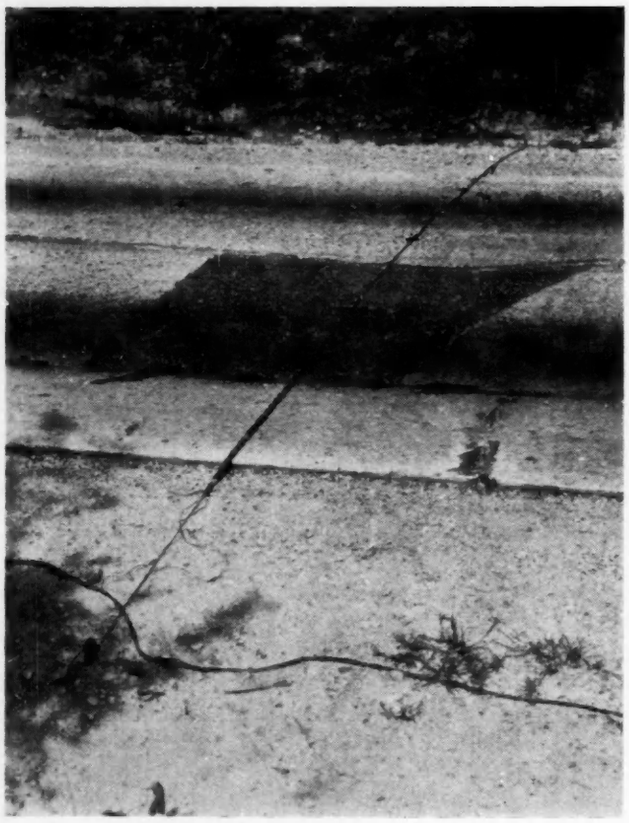


Fig. 7 – Road Switch

Grave discrepancies were apparent in all save a very few of the cases studied. In a few cases, the axle ratio was found stamped on some part of the driving axle assembly. Even in these cases observed gross weights and those predicted by the formulas failed to agree. In cases where we could not find them marked on the vehicle, we got the axle ratios from published specifications. In general, none of the optional ratios seemed to provide a check. It was noted that, in practically every case, the gross load which actually pulled the vehicle down to its crawl speed was far below that given by the formula being, on the average, about two-thirds of the formula value.

Since in the denominator of the formula, rolling radius and grade are measured quantities and the rolling resistance well established by tests, it may be suspected that excessive values were used in the numerator for torque or efficiency.

Investigation To Be Extended

The extension of the work done last summer is being planned and arranged for actively. It will be carried forward on three general fronts. In the first place we are planning to secure a large amount of data on hill-climbing practice through the agency of several of the State Highway Planning Surveys. Secondly, we have arranged with the Motor Trans-



Fig. 8 – Chronograph, Clock, and Instrument Box

port Division of the Quartermaster Corps of the Army to use its dynamometer laboratory at Camp Holabird, Baltimore, for a study of special vehicles. Thirdly, we are giving attention to the availability or development of devices which may be used in the enforcement of performance regulations when they finally are promulgated.

Study of Present Hill Speeds Proposed

It is proposed that mass data on current hill-climbing practice be secured through the use of a technique and apparatus quite similar to those reported here, and that they will be applied through the cooperation of several of the existing State Highway Planning Surveys. A wide range of grades is contemplated as well as a variety of pavement surfaces. The



Fig. 9 – Chronograph Operation

data thus secured should prove useful in a general study of the effect of grade, length of hill, and vehicle weight and type on performance.

Dynamometer Studies To Parallel Hill Tests

The purpose of the work planned for an early beginning at Camp Holabird may be stated briefly as follows:

(1) To discover what may be expected as the best possible performance of new vehicles. We may thus establish the ideal or optimum ability of current models.

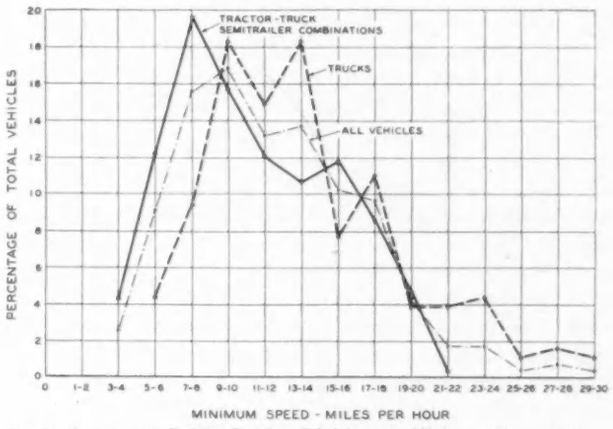
(2) To find out how ability is affected as age and mileage increase. In this way we may get an idea how to discount proposed regulations to provide for wear.

(3) To verify some of the performance formulas or, in the absence of verification, possibly to establish empirical coefficients which will pull them in line with observed facts.

These objectives will involve assembling at Camp Holabird a considerable number of both new and used trucks.

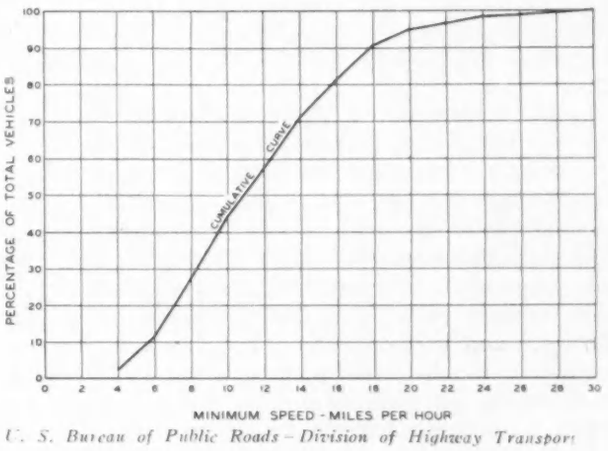
New vehicles of selected makes and models may be supplied to a limited number by both the Bureau of Public Roads and the Army, but it is hoped that the truck manufacturers also will be willing to cooperate in this phase of the undertaking. It is planned to secure used vehicles of selected makes, models, and mileages whose maintenance history is known. These vehicles may be obtained from the Bureau, the Army, and, it is hoped, from local operators.

The data to be gathered in the dynamometer laboratory at Camp Holabird will include tire pressures, rolling radii, axle ratios, engine torque-speed curves, drive-wheel torque – road speed curves with the vehicle in various transmission gear ratios, and so on. The vehicle then will be loaded to various degrees and run over nearby test grades of selected steepness. The transmission gear ratio in which the vehicle may be running at any moment on the test grades will be determined by



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Fig. 10 – Frequency Distribution of Minimum Speed



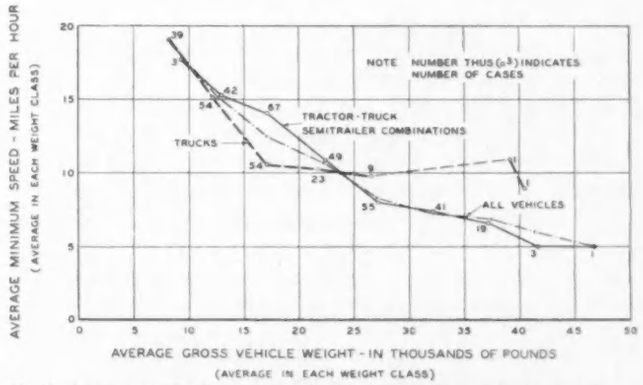
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Fig. 11 – Cumulative Frequency Distribution of Minimum Speed

an observer in the cab who also will see that the driver is getting the best performance that he can. The speed at which the truck moves may be recorded by a special instrument driven by a fifth wheel attached to the vehicle. This instrument gives a continuous plot of instantaneous speeds as the truck ascends the grade.

Enforcement of Performance Requirements

An attempt has been under way since last summer to uncover devices or schemes which may be used as enforcement tools. These devices should be capable of predicting whether a vehicle has the ability to climb a hill of given



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Fig. 12 – Variation of Minimum Speed with Gross Vehicle Weight on a 6 Per Cent Grade

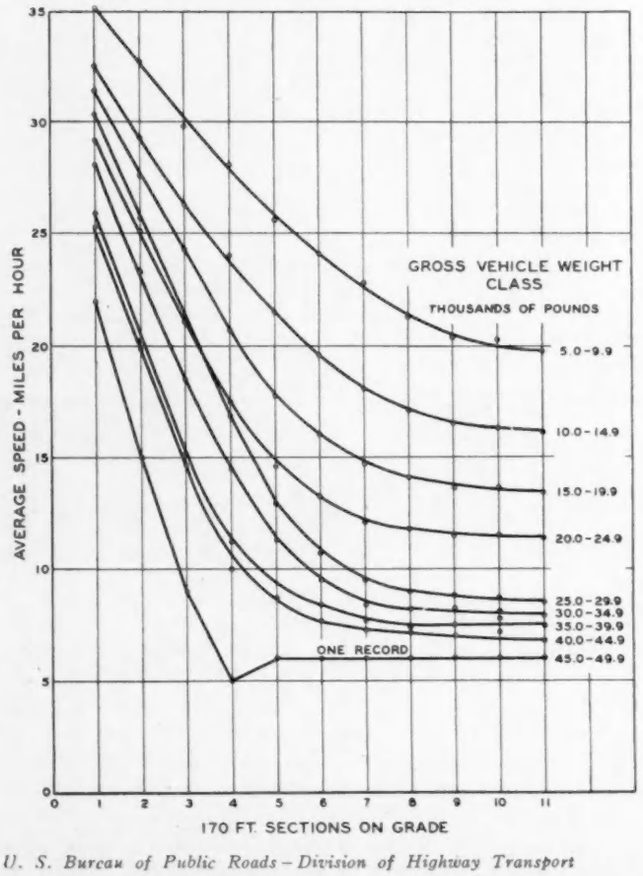
grade under stipulated speed and load conditions. Such devices will obviate the necessity of finding test grades of exactly the steepness indicated in possible future regulations. Limited experience has proved that the use of actual test grades in enforcement is not satisfactory.

Two instruments have been examined. Their producers claim that, when attached to the vehicle, one can read the accelerations which they record and make the necessary interpretations. We have not been satisfied fully that this result is possible for reasons which cannot be gone into here. Even were it possible to secure an accurate measurement of accelerations, the average enforcement officer would be at a loss to interpret them in terms of grade ability.

We also have examined a water-brake type of dynamometer which possibly could be used effectively. This device is not prohibitively expensive, and may be placed at strategic points on main routes and used to check and enforce ability requirements in much the same way as pit scales are used commonly to control overloading. Just as we have portable weighing devices, we also could have portable dynamometers, and indeed one company is now developing such a portable type.

An added advantage of the use of dynamometers for enforcement lies in their adaptability to the field of preventive maintenance. Not only could they be used at times of periodic inspection but, when vehicles suspected of lack of ability are tested on them, it could be ascertained then and there whether the apparent lack was due to some fundamental cause or to some such temporary cause as, for example, a poorly adjusted carbureter or fouled distributor points.

In conclusion, let me say that everyone connected with the investigation is fully appreciative of the complexity and seriousness of the undertaking. For this reason it is not being entered upon lightly but with a determination to proceed carefully and as slowly as need be so that the final outcome may represent a real contribution to the continued progress and development of highway transportation.



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Fig. 13 – Variation of Speed on a 6 Per Cent Grade



Fig. 14 – Measuring Rolling Radius (Level Bubble Is Mounted on the Far Side of the Cantilever Arm)

Discussion

Sight-Distance Improvement Needed

– W. F. Childs, Jr.

State Roads Commission, Maryland

WHEN considering such studies as the one just reported and their practical value in application, it is natural that I consider them in association with Maryland's problem which, after all, may not be so far disassociated from those that confront other States.

Unquestionably, in the design of new roads, information recently developed with reference to critical sight distance will be put to good usage in the improvement of both alignment and grade. The importance of these features will be increased and made more impressive by studies such as those now being carried on by Dr. Thompson.

The serious problem is the correction of these conditions on thousands of miles of existing heavily traveled highways.

Dr. Thompson suggests four lines of attack. Separately and collectively they are possessed of admirable merit, but time and cost are essential elements to be considered.

Delay to the normal flow of traffic by slow-moving vehicles and trucks is not confined to grade alone but to horizontal curvature as well when sufficient sight distance is not provided.

I submit that, on the average two-lane highways which today are carrying near-capacity traffic, impeded by the customary coterie of trucks and other slow-moving vehicles, a careful study of the situation will develop that, owing to short sight distance at vertical and horizontal curves and on steep grades, the movement of traffic is slowed up by a length of road that is but a small percentage of its total mileage.

It occurs to me that obviously the common-sense thing to do is to locate and to study these offending places. The cutting off of knolls on a half dozen summits and improvement of sight distance on as many horizontal curves may mean the elimination of obstructions to the free, normal flow of traffic by the rehabilitation of as little as 10 per cent of the mileage.

Where the nature of the terrain, or for other sufficient reasons, the sight distance cannot be improved and the grade reduced economically, then there is no law against locating points between which traffic is retarded and, from the point where such retardation begins to where normal flow is resumed, to widen the road to four lanes, reserving the extreme right lane for the use of slow-moving vehicles. In cases of retardation of flow due to horizontal curves, it will be necessary to provide only one additional lane in many places.

By this procedure relief has been provided where it is needed most, sanely and economically, with the added advantage that it lends itself to stage improvement of the road as a whole, starting with those places that give the greatest relief to free movement of traffic.

Diesel Supercharging – Its Effect on Design and Performance

By Russell Pyles
Clark Bros. Co.

METHODS of increasing engine output are discussed, and supercharging is said to involve few difficulties. The location of the blower drive as it affects frequency and gear loading is considered. Bearing-load diagrams are analyzed for a high-speed and a low-speed engine.

Tests indicate a negligible increase in piston temperatures. Fuel-consumption curves show fuel economy comparable with that of the normal engine.

ENGINE performance must be improved continually. Greater output per cubic inch of displacement, lower specific weight, and a lower specific engine volume are necessary. The means available for obtaining these results are:

(1) Increased engine speed.
(2) Increased mean effective pressure.
(3) The two-cycle principle.
(4) Supercharging or boosting.

Rotational speed in small-bore and -stroke Diesels may be limited by the injection system, the difficulty being in shortening sufficiently the injection period. With larger bores and strokes the limiting feature is piston speed as restricted by a reasonable lubricating-oil and fuel consumption.

One limitation on fuel consumption is volumetric efficiency as controlled by allowable velocities through the valves. In the present state of the art, Diesels still require considerable excess air and, neglecting smoke density, the limiting power output of an engine is defined by that point where the product of volumetric efficiency and r.p.m. ceases to increase in value. Any increase in volumetric efficiency automatically raises the r.p.m. at which the horsepower curve peaks. Any improvement in the coefficient of discharge of the valves increases volumetric efficiency. Notable research work along this line has been done by Dennison and others.[1]

The exhaust-valve passage should be designed as a diverging nozzle and the inlet-valve passage as a converging nozzle, the valve seat and valve itself being shaped geometrically to approach, as nearly as possible, the throat of a venturi. Thus friction and eddy losses through the valves are reduced to a minimum. Such treatment may result in a 15 to 20 per cent improvement in the average valve coefficient of discharge above that obtainable with conventional valves and passages. The actual application of these simple principles on a 9- x 12-in. engine at 900 r.p.m. permitted a reduction in valve lift from ⅜ to ½ in. with no sacrifice of volumetric efficiency. This reduction, notwithstanding the fact that the velocity through the inlet valves, based on clear diameter and full lift, was increased from 203 ft. per sec. to 254 ft. per sec. Streamlined valves and passages on an 8- x 10½-in. engine at 810 r.p.m. corresponding to 210 ft. per sec. through the valve, showed a volumetric efficiency of 85 per cent.

Mean effective pressures may be increased by the use of special combustion-chambers, turbulence, and proper correlation with the nozzle and pump. All such means operate to improve combustion efficiency and reduce the excess-air requirements. Increased engine speed adds to the difficulty. The excess air for 30 per cent thermal efficiency and 85 lb. per sq. in. b.m.e.p., corresponding to approximately 100 lb. per sq. in. m.i.p. (mean indicated pressure) for a multicylinder engine, is 40 per cent. The limiting m.i.p. for a naturally aspirated engine with no excess air is about 138 lb. per sq. in.

Increases in available air and higher mean effective pressure may be obtained by inertia supercharging accomplished by applying individual intake pipes of the proper length to each cylinder-head. A gain of 10 per cent is possible, but the appearance of such an installation has little to commend it.

The two-cycle principle with its power stroke every revolution always has been an attractive means of increasing engine power. The attainment at high engine speeds of sufficient valve time area with port scavenging and discharge is difficult. When valves are added to the cylinder-head for either intake or exhaust, the engine then has all the details of a four-cycle engine, with the addition of high acceleration loads in the valve gear and the greater heat-transfer problems attending two-cycle operation.

Supercharging – increasing the weight of air charge above that of normal aspiration – offers a means of increasing the specific output of Diesel engines with a minimum of disadvantages and new problems. It can be applied readily to existing conservative engine designs to raise the brake horsepower approximately 50 per cent. Scavenging of the clearance volume is obtained readily with the proper valve overlap. This method gives, in effect, increased displacement with no change in bore or stroke and, at the same time, it improves combustion efficiency. Engine speed may be increased, as the limiting velocity through the valves is raised. A slightly greater pressure drop across the valves may be tolerated at the expense of a small increase in blower horsepower.

Supercharging

In applying supercharging to a new engine or an existing design, we are confronted with the question of how much punishment this increased output is going to impose upon the

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1938.]

[1] See A.S.M.E. Transactions, Vol. 53, 1931, OGP-53-6, pp. 79-97: "Experiments on the Flow of Air through Engine Valves," by E. S. Dennison, T. C. Kuchler, and D. W. Smith.

engine. Obviously, added horsepower cannot be purchased at the expense of shortened engine life and greater maintenance.

The effect of supercharging upon some of the major design

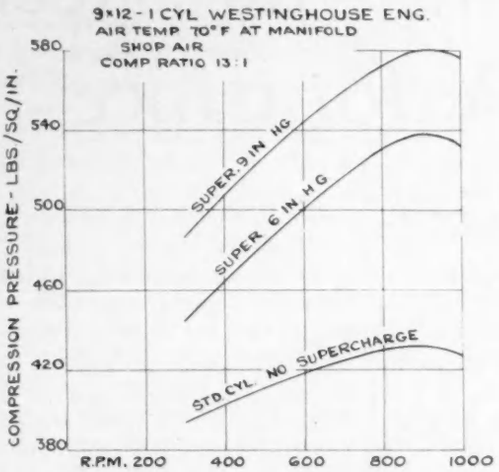


Fig. 1 – Compression Pressure Vs. R.P.M.

features has been investigated. A logical consideration of the problem starts with combustion and the indicator card. The comparative data presented herein are based upon the following general conditions:

Standard Engine, Normal Aspiration

Maximum Cylinder Pressure	= 750 lb. per sq. in. gage
Compression Pressure	= 425 lb. per sq. in. gage
Mean Indicated Pressure	= 100 lb. per sq. in.
Brake Mean Effective Pressure	= 77 lb. per sq. in.

Supercharged Engine

Manifold Pressure	= 10 in. hg.
Maximum Cylinder Pressure	= 850 lb. per sq. in. gage
Compression Pressure	= 515 lb. per sq. in. gage
Mean Indicated Pressure	= 150 lb. per sq. in.
Brake Mean Effective Pressure	= 115 lb. per sq. in.

The compression pressure and the maximum cylinder pressure for the supercharged engine are somewhat higher than that of the standard engine. If the compression ratio for the standard engine had not been lowered, the compression pressure with 10 in. hg. supercharge would have been approximately 565 lb. per sq. in. instead of 515 lb. per sq. in. For the same compression ratio the constant-volume cycle results in a higher thermal efficiency than the constant-pressure cycle, but also in excessive cylinder pressure. If, due to engine structural reasons, we set a limit on the maximum cylinder pressure, then it follows that the highest thermal efficiency will result with the lowest practicable compression pressure. This compromise permits burning the maximum amount of fuel in the region of top-center and approaches the constant-volume cycle.

Table 1 shows the effect of a similar reduction in compression pressure, based upon tests on a single-cylinder 9- x 12-in. engine at 900 r.p.m., using a dished-top piston and 10 in. hg. supercharge.

Table 1[2]

B.Hp. Single-Cylinder Engine		570 lb. per sq. in. Compression Pressure 975 lb. per sq. in. Maximum Pressure	517 lb. per sq. in. Compression Pressure 870 lb. per sq. in. Maximum Pressure	Per Cent Change
80	Lb. Fuel per B.Hp-Hr.	0.415	0.424	2.2
90		0.417	0.428	2.6
100		0.427	0.443	3.7
80	Smoke Density	0.16	0.17	6.3
90		0.23	0.25	8.7
100		0.34	0.40	17.6

Although the lower pressure cycle shows a higher fuel consumption of 2-3 per cent, this increase is negligible. The change in smoke density is too small to be noticed by the

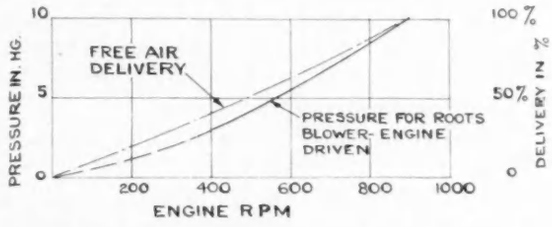


Fig. 3 – Pressure and Delivery of an Engine-Driven Roots Blower

naked eye. The smoke-density measurements were made with a smoke meter. The performance loss is very small for a reduction in maximum cylinder pressure of 105 lb. per sq. in.

So far as the ear was concerned, the presence of combustion shock was not more noticeable at 850-900 lb. per sq. in. maximum pressure when supercharged than at 750 lb. per sq. in. maximum pressure standard.

The low limit for compression pressure would seem to be determined by starting facility. Compression pressure increases with engine speed in a normally aspirated engine. This condition holds true also for the supercharged engine but to a greater degree. The blower pressure drops off rapidly with decreasing speed and, at starting speeds, the initial cylinder pressure is little more than atmospheric. This condition results in a lower compression pressure. Fig. 1 shows how compression pressure varies with initial pressure and speed.

It has been said with respect to turbo-charging that blow-through of cool air during the scavenging process was necessary for cooling of the exhaust turbine. Although this cooling is unquestionably of value in such installations, cooling alone is not required when using a direct-driven blower. However, to obtain the full advantage of supercharging, it is essential that the combustion-chamber be scavenged completely by sufficient pressure and overlap of the inlet and exhaust valves. The cooling from such blow-through is incidental, and of

[2] Westinghouse Report Library Serial No. E-873 (unpublished): "Supercharge Tests of a Dished Piston," by R. D. Wellington.

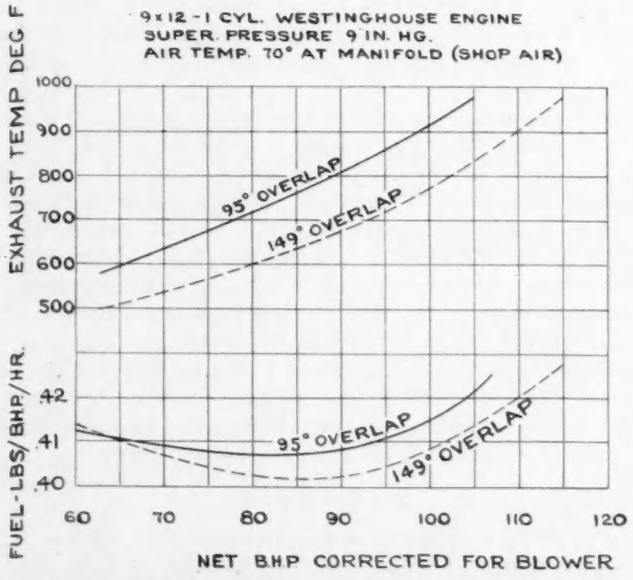


Fig. 2 – Effect of Valve Overlap on Fuel Consumption and Exhaust Temperature

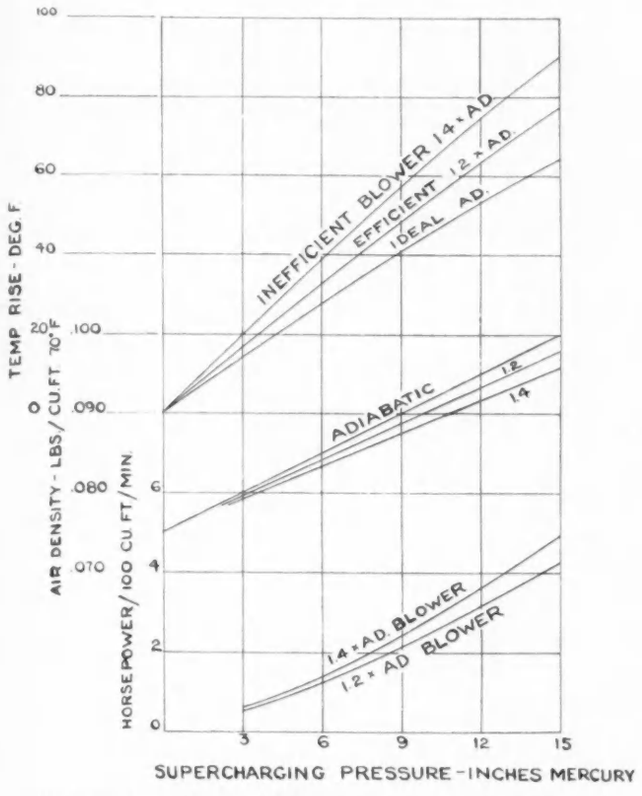


Fig. 4 – Supercharger Air-Temperature Rise, Density, and Blower Horsepower Vs. Pressure

value, but the object is scavenging. The valve overlap and scavenge pressure required are a function of the engine speed and combustion-chamber type. In general, the required valve-overlap period will increase with engine speed, the criterion here being "valve-time area." This "valve-time area" is a convenient design factor which may be used for extending valve performance data of a given engine to other sizes and speeds. In such a generally applicable form it is in units of:

$$\frac{\text{Square inch seconds}}{\text{Cubic inches per second of piston displacement}}$$

The numerator is based upon integration of the valve-lift curve on a time base, and the denominator is simply a rate of displacement.

The open type of combustion-chamber lends itself to ease of scavenging but pre-chamber types are more difficult, and complete scavenging of some types would be impossible without the aid of auxiliary valves.

Fig. 2 shows how the valve-overlap period affects fuel consumption and the ability to carry high loads with an open combustion-chamber.

Blowers

In many types of service a variable-speed engine may be called upon to carry full torque at any point in its speed range. This is a desirable characteristic for a Diesel-electric railcar or a switching locomotive, resulting in rapid acceleration of the train. The ability to carry full torque in a truck engine gives it that lugging ability for hill-climbing which minimizes gear-shifting. Rapid acceleration under load on the draw works of drilling rigs is one of the outstanding reasons for Diesel preference in this service.

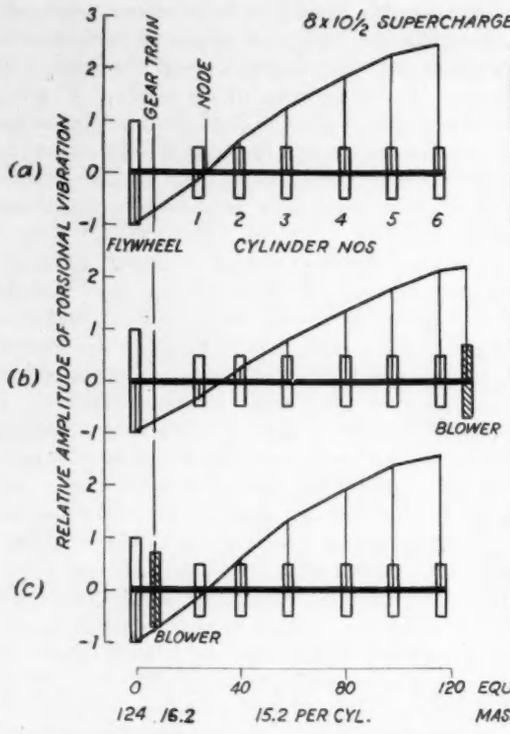
Such performance demands that the air-fuel ratio be maintained at all engine speeds. Because the pressure and delivery characteristics of a Roots blower do not fall off with speed as rapidly as they do with centrifugal blowers, the former type was chosen for this service. See Fig. 3. The efficiency of the Roots blower is reasonably good if clearances are not excessive but, if clearances are large, the slip increases with a resulting increase in the temperature rise of the air going through the blower, and a decrease in air density. The 1.2 times adiabatic line in Fig. 4 may be considered as typical for an efficient commercial high-speed blower. A 1.4 times adiabatic temperature rise may result easily from excessive clearances with reductions in air density as shown by the lower curves.

The location of the blower drive when engine-driven is extremely important because of the effect the blower mass may have on the natural frequency of the system, and also because of the effect of torsional vibrations upon the blower driving gears. In general, Diesel engine applications, other than marine installations with long driveshafts, require a consideration of only the first mode of crankshaft vibration. The elastic curve for such a single-noded vibration is shown in Fig. 5(*a*) for a six-cylinder engine with a flywheel. The natural frequency of this type of system is the highest to be encountered in any applications and is approached by the case of the engine solidly coupled to driven equipment of negligible mass. The other extreme, the largest mass to be dealt with in usual applications, is typified by the engine-generator unit shown in Fig. 6(*a*). This system in general gives the lowest frequencies. The following figures (*b*) and (*c*) in Figs. 5 and 6 show the effect on the natural frequency and the amplitudes at the blower drive gears when directly driving the blower from the free end of the engine or from a point near the node. All engines have some critical speeds within their running range. When a variable-speed engine passes through or runs on or near a critical speed, the crankshaft will vibrate torsionally in the form indicated by the elastic curves. The maximum amplitude will be at the free end of the shaft. The amplitude at the node will be zero, and this point alone on the shaft has constant angular velocity. Obviously, this is the optimum location for any gear-driven auxiliary equipment, either blower camshaft or pumps. Being driven from such a location, the gears are not subjected to the accelerations and decelerations of torsional vibration. Their maximum loads are then due only to the normal acceleration of the engine when going from idling to full speed at full torque.

An engine often must be designed to cover a wide field of applications; consequently the mass of the driven equipment and the elasticity of the shafting will vary with individual installations. As a result the nodal point is not fixed, and the practical compromise lies in locating the gear train midway between the limiting locations of the node. This compromise has been made on the engine for which the elastic curves are shown. For the engine and flywheel only, the node lies close to cylinder No. 1 whereas, for the engine-generator unit, the node lies beyond the flywheel. In the case of the engine and the flywheel only, the amplitude at the blower drive gear is three times larger for the free-end drive than for the near-

Table 2

Condition	Torque at Blower Drive Gear, in-lb.
Normal acceleration of the engine under full mean torque – 0 to 720 r.p.m. in 1.75 sec.	705
Mean driving torque of the blower, based on 25 hp. at 720 r.p.m.	2,190
Fig. 6(*c*)	
Blower driven from near node. Engine in critical. Amplitude held to ½ deg. at free end by damper	5,250
Fig. 6(*b*)	
Blower driven from free end. Engine in critical. Amplitude held to ½ deg. at free end by damper	25,000



ELASTIC CURVE FOR CRANKSHAFT WITH-:	NATURAL FREQUENCY CYCLES PER MIN.	RELATIVE AMPLITUDE AT BLOWER DRIVE GEAR
ENGINE AND FLYWHEEL ONLY	6,035	-
BLOWER DRIVEN FROM FREE END	5,400	2.195
BLOWER DRIVEN FROM BETWEEN FLYWHEEL AND No. 1 CYLINDER	6,025	0.73

Fig. 5 – Effect of Blower-Drive Location on Frequency and Amplitude of Torsional Vibration – Engine and Flywheel

flywheel drive. For the engine-generator unit, the free-end amplitude is almost five times larger than the amplitude when the drive is taken off near the node. The acceleration and torque are proportional to these amplitudes, from which it follows that tooth loadings for the gear drive are one-third to one-fifth lower when driven from near the nodal point.

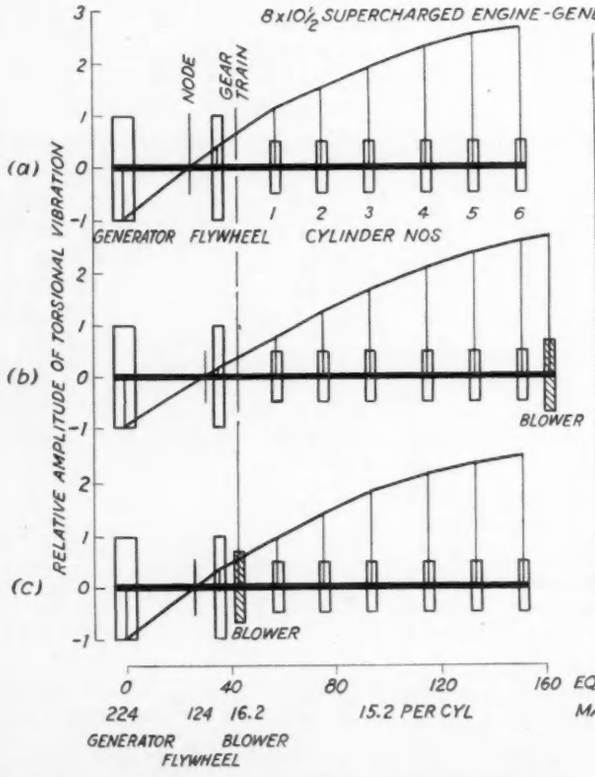
To show more clearly the effect of these drive locations on the gear loadings, the torques have been calculated for the engine-generator unit of Fig. 6(*b*) and 6(*c*). The assumption has been made that a damper is fitted on the free end (not shown), and it functions to hold the maximum amplitude in the sixth-order critical down to ½ deg. At such an amplitude the crankshaft stresses are within safe limits. Table 2 shows the resulting torques at the blower drive gear.

Even with the blower drive located near the node, it has been found desirable to incorporate a friction safety clutch between the blower and engine. In one form this clutch consists of a friction disc on each side of the blower gear, loaded by adjustable compression springs. The breakaway torque has been set at approximately three times the normal driving torque of the blower. With such a setting the clutch is likely to slip while the engine goes through major criticals, acting to keep the gear loading within allowable limits. See Figs. 7, 8, and 9.

When the blower drive is near the node, its effect on the natural frequency of the system is small. When driven from the free end of the engine, frequency is reduced appreciably. This is a serious disadvantage.

Effective muffling of the blower inlet is necessary. In addition to mufflers, the simple expedient of rounding the inlet ports with a generous radius reduces noise. On one blower at 12 in. hg. and 3000 r.p.m., the total noise was reduced from 93 to 86 db. This is a reduction in sound energy to less than one-fourth.

Some difficulty was experienced initially with lubrication of the blower timing gears in the blower of Fig. 10. To provide ample oil, it was fed to the center of the approaching faces of the gears through a 1/32-in. diameter hole at 30 lb. per sq. in. pressure. Fig. 11 shows the result and is a good lesson on how not to lubricate such gears. The oil-feed hole was then moved to feed the receding faces of the gears with satisfactory results.



ELASTIC CURVE FOR CRANKSHAFT WITH-:	NATURAL FREQUENCY CYCLES PER MIN.	RELATIVE AMPLITUDE AT BLOWER DRIVE GEAR
ENGINE, FLYWHEEL, AND GENERATOR ONLY. NO BLOWER	4,400	—
BLOWER DRIVEN FROM FREE END	4,010	2.66
BLOWER DRIVEN FROM BETWEEN FLYWHEEL AND No. 1 CYLINDER	4,390	0.56

Fig. 6 – Effect of Blower-Drive Location on Frequency and Amplitude of Torsional Vibration – Engine-Generator and Flywheel

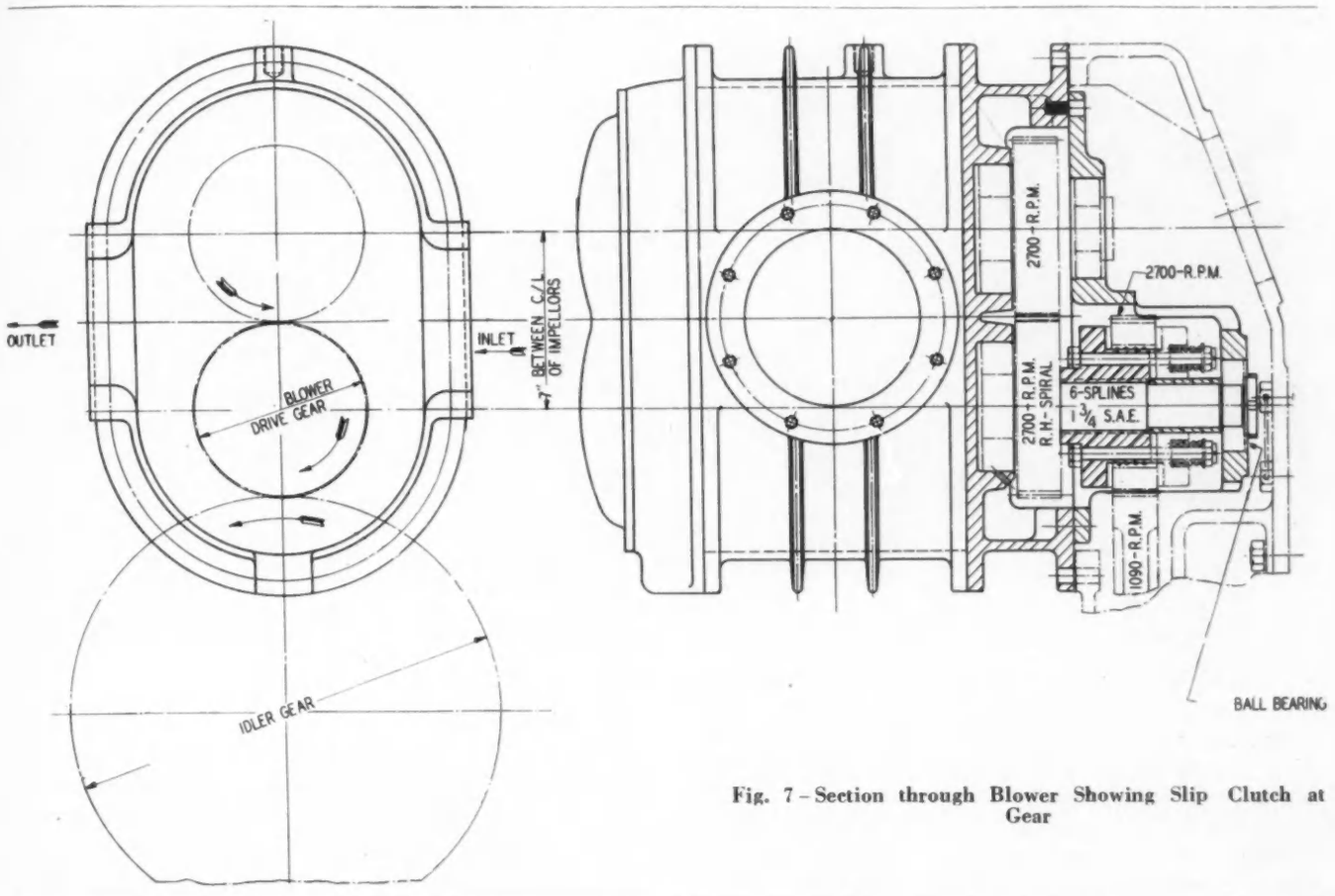


Fig. 7 – Section through Blower Showing Slip Clutch at Gear

Effect of Supercharging on Bearings

The effect of supercharging on bearing loads was investigated for a high-speed and a low-speed engine. The high-speed engine is a Westinghouse 9-in. bore, 12-in. stroke railway engine with aluminum pistons and a piston speed of 1800 ft. per min. at 900 r.p.m. The low-speed engine is a Chicago Pneumatic stationary engine, 14-in. bore by 22-in. stroke, with cast-iron pistons and a piston speed of 1100 ft. per min. at 300 r.p.m. It will be noted that these two cases cover a wide range of engine types.

The connecting-rod bearing is the first to feel the effects of high cylinder pressures. Figs. 12 and 13 show how supercharging altered the rod bearing load diagrams, the increase in b.m.e.p. being 50 per cent in both cases. Referring to Fig. 12 for the low-speed engine, notice that the change in maximum load was 18 per cent while the change in mean load was 16 per cent. In a slow-speed engine the connecting-rod bearing loading is dominated by the gas pressure. The major influence on mean bearing load is mean indicated pressure and, similarly, the maximum bearing load is influenced by the maximum gas pressure. Refer now to Fig. 13, the connecting-rod load diagram for the high-speed engine. The increase in maximum loading is 20 per cent, substantially the same increase as in the slow-speed engine. However, the mean load has increased only 4 per cent instead of 16 per cent. As engine speeds increase, the inertia force of the reciprocating weights and the rotating weights of the connecting-rod become the primary factors in determining connecting-rod bearing loads. We can visualize engine speed being carried to the point where the influence of gas pressure will be negligible.

The influence of speed and gas pressure on the maximum connecting-rod bearing load is shown in Fig. 14 for an 8- x 10½-in. Diesel using cast-iron pistons. The full line shows the maximum rod bearing load versus speed for the normal engine. Below 900 r.p.m. the maximum load occurs on the power stroke, the maximum cylinder pressure being the dominating factor. At 900 r.p.m. the maximum load occurs on the exhaust stroke, and inertia plus centrifugal forces dominate.

The dotted line shows the load when the maximum cylinder pressure is raised to 850 lb. per sq. in. Below 960 r.p.m. the maximum load is increased and still occurs on the power stroke. However, beyond this speed, the maximum load occurs on the exhaust stroke and follows along the same line as when unsupercharged. Now the normal rated speed of this engine is 720 r.p.m. but it will be seen from the curve that so far as maximum loads are concerned, the engine could be supercharged and the speed increased to 860 or 1050 r.p.m. without any increase in maximum connecting-rod bearing load. If increased to 1050 r.p.m., the point of maximum loading simply is shifted from the power stroke to the exhaust stroke. Such an increase in speed does increase the mean load on the bearing – but few Diesel-engine connecting-rod bearings ever fail due to mean loads. A bearing may be made to carry a higher mean load by circulation of a larger quantity of oil. Furthermore, an increase in r.p.m. of the journal automatically increases its load capacity if the rate of oil circulation is increased correspondingly to maintain the ZN/P ratio.

Figs. 15 and 16 are the center main bearing load diagrams for the low- and high-speed engines. Here again the low-speed engine shows the greater increase because gas pressure plays the leading role, but the per cent increase in load is small. On the high-speed engine the change due to supercharging is negligible. The effect of the higher supercharged cylinder pressure is submerged by the great influence of the forces due to reciprocating and rotating weights.

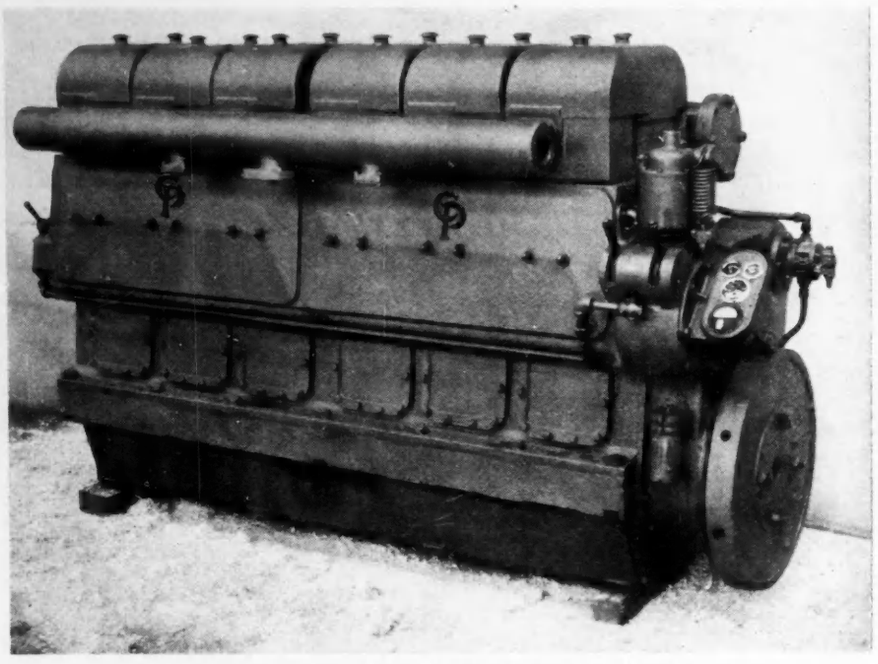


Fig. 8 – Engine Designed for Simple Application of a Blower

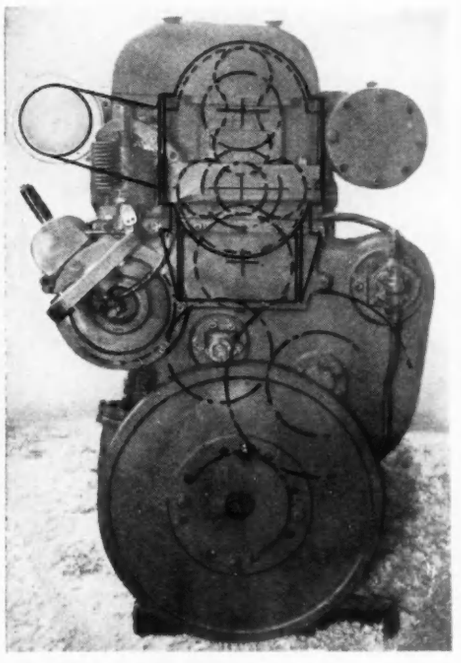


Fig. 9 – Method of Mounting Blower on Engine

The bearing load diagrams indicate that a power increase of 50 per cent in brake horsepower has the following effects:

(1) The greatest change is in the maximum load on the connecting-rod. This load may still be within safe limits on a conservatively designed engine. If the increased load approaches the danger line, the remedies are either an increase in reciprocating weight, an increase in engine speed, or a change in bearing material.

(2) The increased load on the center main bearing is of such small magnitude that its effect can be neglected, or its effect can be cancelled readily by an increase in the rate of oil circulation.

(3) The supercharged engine shows what might be termed a higher economic bearing efficiency in that the mean bearing load in pounds per brake horsepower output is smaller than on a normally aspirated engine.

(4) The small increases in bearing loads predicted by the load diagrams were confirmed in test.[3] The heat loss to the

[3] See *Mechanical Engineering*, Vol. 57, September, 1935, pp. 547-552: "The Diesel Engine for Rail Transportation," by John Dickson.

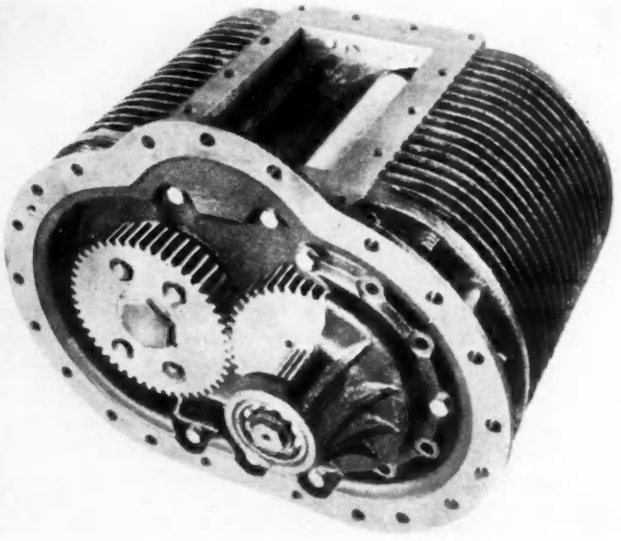


Fig. 10 – Westinghouse-DePalma Blower

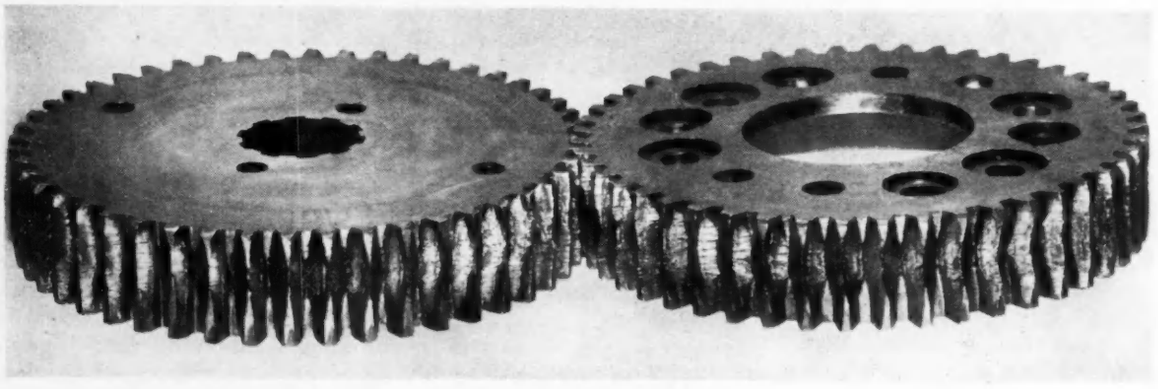


Fig. 11 – Effect of Incorrect Lubrication of Timing Gears

Blower Timing Gears

25 hp. at 3000 r.p.m., pitch-line velocity 4700 ft. per min.

Material, S.A.E. 2340
Tensile, 110,000 lb. per sq. in. approximate
Brinell, 225 – 260

Material, S.A.E. 3140
Tensile, 150,000 lb. per sq. in.
Brinell, 300 – 330

Oil fed to the center of the approaching faces of the gears at 30 lb. per sq. in. pressure through a 1/32-in. diameter hole.

lubricating oil increased only 5 per cent for an increase of 50 per cent in the b.m.e.p.

Effect on Crankshaft and Flywheel

The increased horsepower due to supercharging is reflected in an increase in the maximum indicated torque. Fig. 17, for a six-cylinder engine, shows an increase of approximately 40 per cent in this maximum value. This fact, however, is of little importance from a design standpoint as crankshaft sizes must be based upon considerations of bearing area and stresses in critical speeds.

Flywheel requirements, in the past, have been considered as influenced by the ratio of the $\frac{\text{maximum torque}}{\text{mean torque}}$. This value is substantially the same for a given engine when supercharged or unsupercharged.[3] The flywheel requirement, therefore, would appear to be unaltered by supercharging. However, a rational determination of the flywheel size requires that the engine, flywheel, and driven equipment be considered as an elastic system under forced vibration. The problem is similar to that of a critical-speed analysis and has been treated in detail by Andriola.[4] On this basis, the forced vibration energy is a function of one of the harmonic coefficients, depending upon the number of engine cylinders. These harmonic coefficients are derived in the usual manner by harmonic analysis of the torque curve. Unfortunately, the harmonic coefficients for the supercharged engine torque curve are not available for a wide range of mean indicated pressures.

However, from the author's experience, an increase in the value of these harmonics is to be expected when the mean indicated pressure is increased by any means. This statement must not be construed, however, as demanding a larger flywheel for supercharged Diesel generator sets. Speed regulation at the generator depends upon the location of the generator with respect to the node of the elastic system, as well as upon the magnitude of the forced vibration energy.

From a natural frequency standpoint, it is shown clearly in Figs. 5 and 6 that an engine-driven blower may be added with practically no reduction in the frequency of the system.

[4] See Proceedings of the National Conference, Oil and Gas Power Division of the A.S.M.E., State College, Pa., August, 1937, pp. 90-109: "The Problem of Flywheel Effect and Speed Regulation for Diesel-Engine Driven Machinery," by A. D. Andriola.

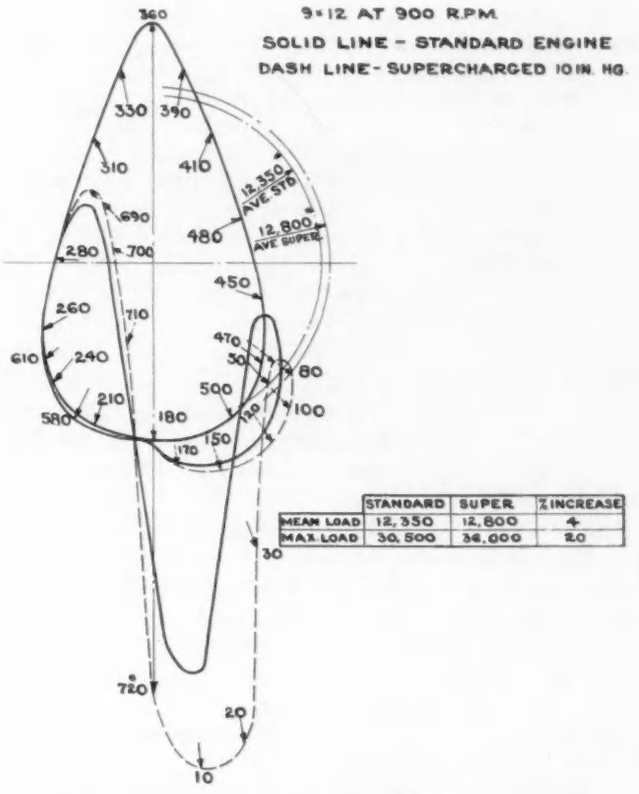


Fig. 13 – Connecting-Rod Bearing Load Diagram – 9- x 12-In. Diesel Engine

The further question arises as to what influence supercharging has upon the amplitudes and crankshaft stresses at critical speeds. The author's experience indicates that supercharging, of itself, does not affect these amplitudes, but that they are influenced greatly by the mean indicated pressure.

The energy input at critical speeds is given by:

$M_o = C_k \times \Sigma\beta \times R \times A$

M_o = Total harmonic torque, in-lb.

C_k = Harmonic torque coefficient, in-lb. per sq. in. per in.

R = Crank radius, in.

A = Piston area, sq. in.

$\Sigma\beta$ = Vector sum of the elastic curve ordinates.

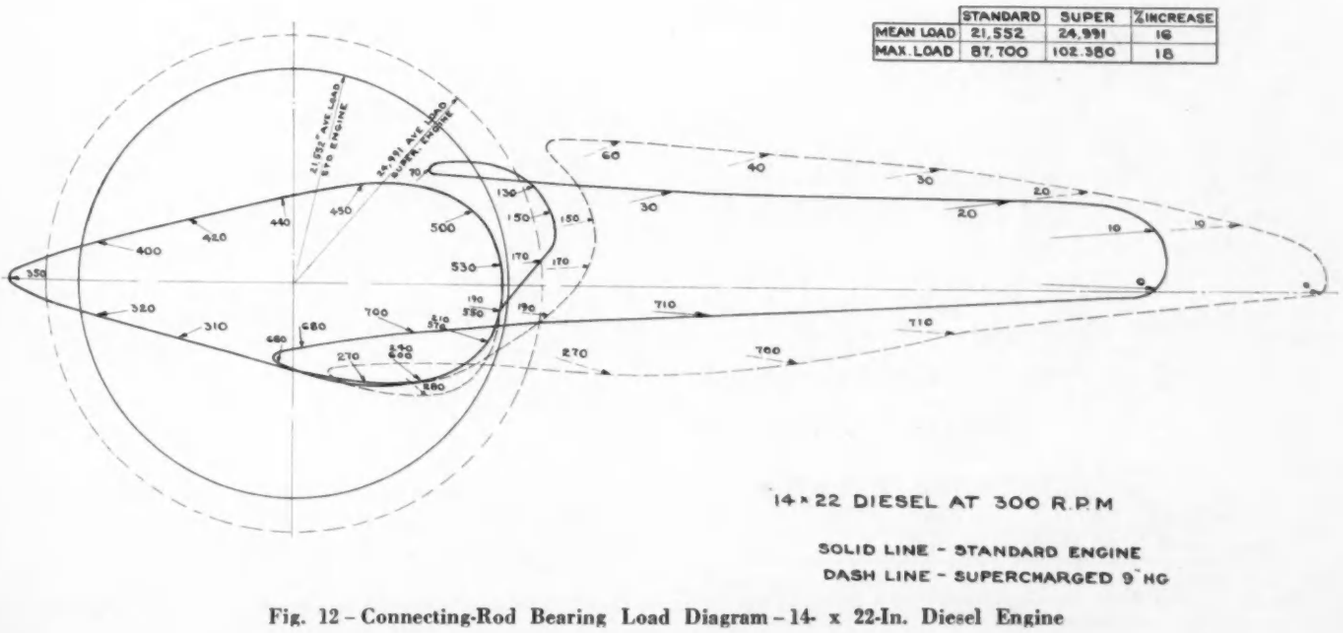


Fig. 12 – Connecting-Rod Bearing Load Diagram – 14- x 22-In. Diesel Engine

For a given engine, then M_0 varies directly with C_k. Harmonic torque coefficients have been calculated by Lewis[5] and Andriola[4] for peak pressure cards of 550 and 750 lb. per sq. in. respectively. Their curves for the sixth harmonic only are shown on Fig. 18. The upper curve has been calculated from actual amplitudes measured on the engines. A Standard and a supercharged engine are represented and two sizes of dampers. In each case the amplitude used for calculating the harmonic coefficient was the mean amplitude measured at the best damper setting and the method of calculation was based upon the damper theory of Hartog and Ormondroyd.[6] Lewis' values are for a slow-speed air-injection engine with a maximum pressure of 550 lb. per sq. in. Andriola's values are based on indicator cards of 750 lb. per sq. in. maximum pressure for a direct-injection engine. The test curve shown is for a high-speed direct-injection engine with maximum pressures up to 850 lb. per sq. in., using a combustion-chamber shaped to give pronounced turbulence.

Both the theoretical and the test data for the direct-injection engine indicate that increases in indicated pressures will result in greater amplitudes of torsional vibration at the sixth-order critical speed. Then, with a 50 per cent increase in mean indicated pressure from 100 to 150 lb. per sq. in., the energy input in the sixth-order critical may be expected to increase 15 to 20 per cent. It may be necessary to make provision for this increase in some supercharging applications by a change in crankshaft diameter or by the addition of more damping.

In the absence of the complete harmonic analysis for a high-turbulence, high-pressure card, it would appear reasonably safe to use the coefficients for other harmonics given by Andriola,[4] making some additional allowance for orders below the sixth.

[5] See Transactions of the Society of Naval Architects and Marine Engineers, Vol. 33, 1925, pp. 109-140: "Torsional Vibration in the Diesel Engine," by F. M. Lewis.

[6] See A.S.M.E. Transactions, Vol. 52, 1931, APM-52-13, pp. 133-152: "Torsional-Vibration Dampers," by J. P. Den Hartog and J. Ormondroyd.

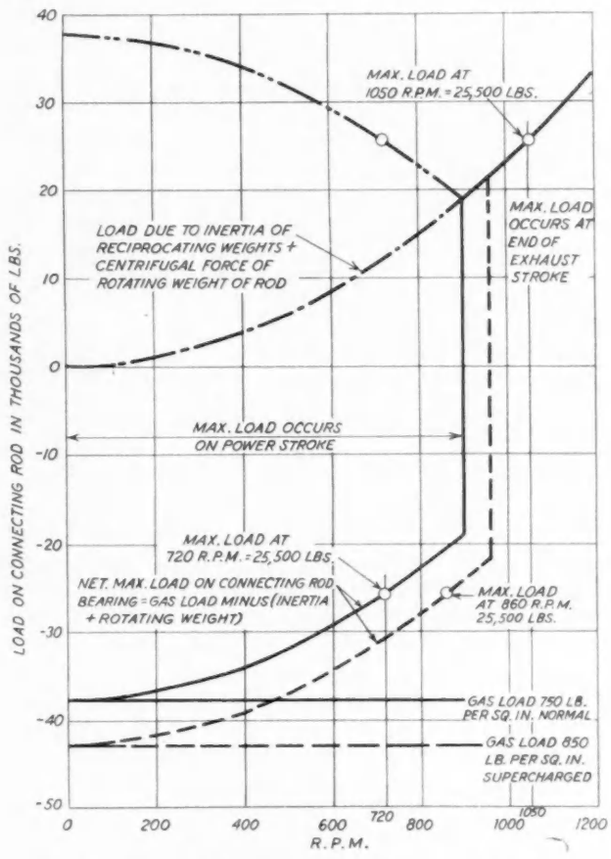
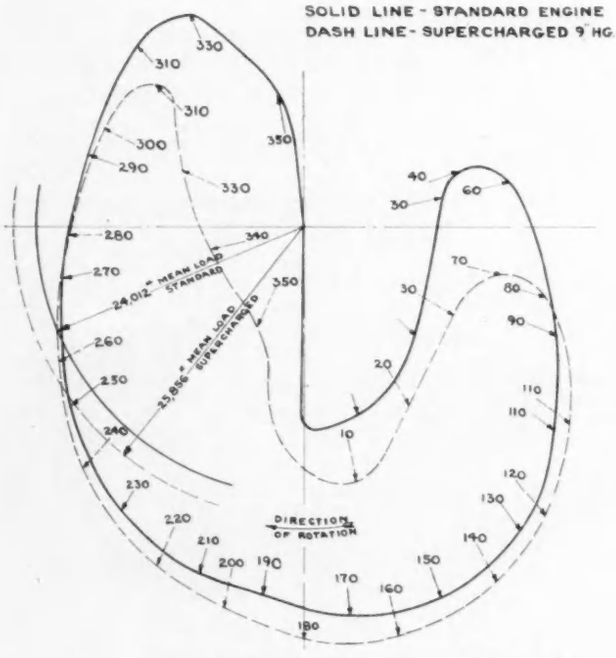
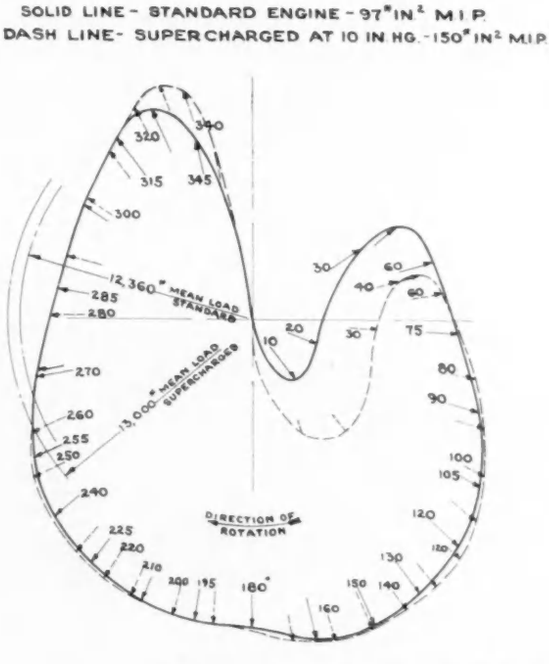


Fig. 14 – Effect of Speed and Gas Pressure on Maximum Connecting-Rod Bearing Loads



	STANDARD	SUPER	% INCREASE
MEAN LOAD	24,012"	25,856"	8
MAX. LOAD	35,300"	37,100"	5

Fig. 15 – Center Main Bearing Loads – 14 x 22-In. Diesel Engine at 300 R.P.M.



	STANDARD	SUPER	% INCREASE
MEAN LOAD	12,360	13,000	5
MAX LOAD	17,550	17,700	1

Fig. 16 – Center Main Bearing Loads – 9 x 12-In. Diesel Engine at 900 R.P.M.

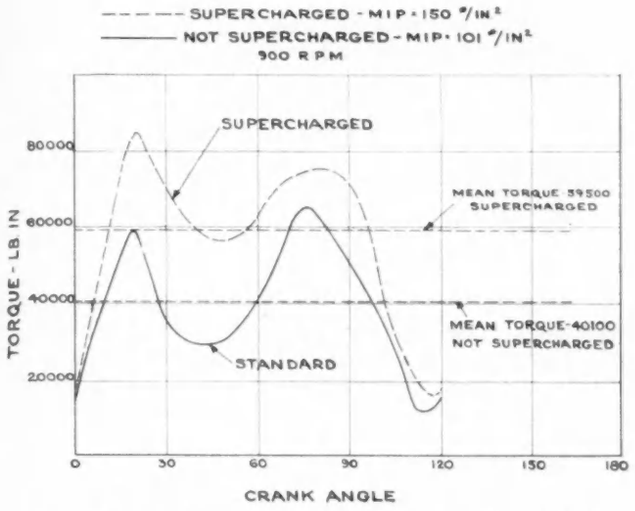


Fig. 17 – Torque Curve – Six-Cylinder Engine (9 x 12-In.)

Effect of Pistons and Fuel Consumption

The increased maximum cylinder pressures required to obtain good performance when supercharging result in about 15 per cent additional load on the piston and pin. Most designs will have a sufficient factor of safety so that no special allowance need be made for this addition.

The increased brake horsepower also might be expected to result in much higher piston temperatures. Aluminum pistons fitted with fusible-alloy plugs were run under supercharge at 50 per cent above normal rating. The temperatures were approximately 40 deg. fahr. higher at the edge of the piston crown and 25 deg. fahr. higher on the skirt than corresponding temperatures for normal rating. This skirt temperature difference might call for a 0.003-in. increase in skirt clearance on a 9-in. piston; however, pistons were run in supercharged service with the same clearances as on the standard engine.

Fig. 19 shows a comparison of the fuel consumption of an engine supercharged and unsupercharged. At the rated speed of 900 r.p.m. and 50 per cent increased brake horsepower, the fuel consumption when supercharged was practically the same as the standard engine. A point of particular importance on the fuel curve for the supercharged engine is the fact that it maintains a low consumption over an unusually wide load range. This flat type of a fuel curve commends the supercharged engine for services where the load varies greatly, such as in generator drives.

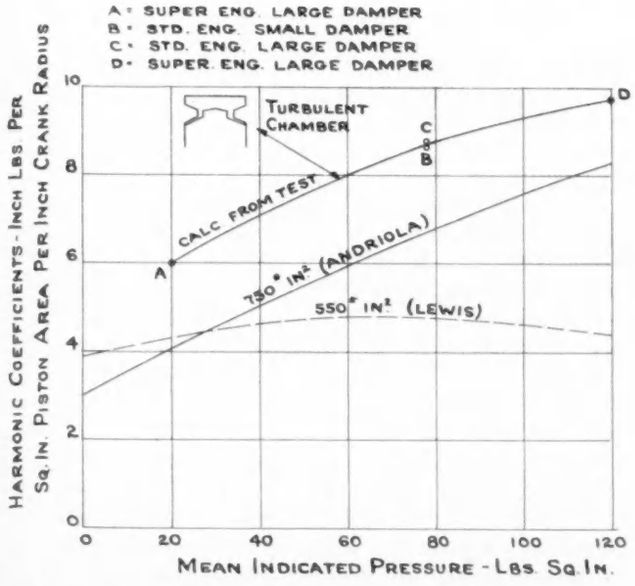


Fig. 18 – Resultant Harmonic Torque Coefficients – 12th Harmonic, Sixth Order, Four-Cycle Engine

Conclusion

The investigation indicates that brake mean effective pressures may be increased 50 per cent by supercharging without the introduction of factors requiring major design changes. The particular features demanding consideration are:

(1) The compression ratio which requires lowering.

(2) The clearance volume should be scavenged completely.

(3) Blower drive location should be such as to have a minimum effect on frequency.

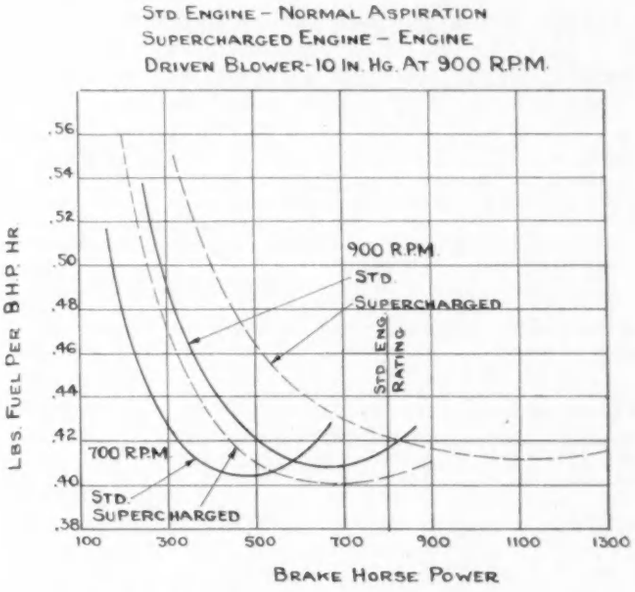


Fig. 19 – Specific Fuel Consumption – Westinghouse 9 x 12-In.; Twelve-Cylinder Engine Supercharged and Unsupercharged

(4) The connecting-rod bearing – provision being made for the effect of increased gas pressure.

(5) Stresses in critical speeds, due to the increase in harmonic torque.

The simplicity of supercharging with its resulting reduction in engine weight promises a wider future application. Furthermore, it appears economically sound to include supercharging as an essential part of the initial engine design as the cost per horsepower is reduced correspondingly.

The author is indebted to the Westinghouse Electric and Manufacturing Co. and the Chicago Pneumatic Tool Co. for permission to use data pertaining to their engines, and also to E. U. Seewer and J. G. Williams for their work in connection with the bearing load studies.

Discussion

Additional Benefits of Supercharging

– Charles S. Moore

Associate Mechanical Engineer, National Advisory Committee for Aeronautics

THE author's reasoning is not understood concerning the effects of combustion cycles on thermal efficiency. As stated, for the same compression ratio, the constant-volume cycle results in a higher thermal efficiency than the constant-pressure cycle. However, for the same maxi-

mum cylinder pressures, the higher thermal efficiency will be obtained by using constant-pressure combustion at the highest possible compression ratio. The 2 to 3 per cent increase in fuel consumption shown in Table 1 is probably a result of operating at the lower compression ratio with correspondingly lower thermal efficiency. If high thermal efficiencies are to be obtained either with or without supercharging, compression ratios must not be reduced. If maximum cylinder pressures are to be limited, the constant-pressure combustion of the Diesel cycle should be developed.

Tests made by the N.A.C.A.[a] have shown that, at a given compression ratio, supercharging decreases the ignition lag and consequently the rate of pressure rise. These decreases are owing to both increased temperature and density of the air charge.[b] Supercharging engines of 5-in. bore at speeds of 1500 to 2500 r.p.m. acts to soften or even eliminate combustion shock. It is possible that with the decreased shock of supercharged operation somewhat higher cylinder pressures may be used for the same degree of engine wear. Also, with the shortening of ignition lag caused by supercharging, fuels of slightly inferior ignition quality may be used.

Author's Closure Explains Difference with Theory

– Russell Pyles
Clark Bros. Co.

IT is true that entropy diagrams based upon simplified theoretical indicator cards will show, for the same maximum cylinder pressure, a slightly higher thermal efficiency with constant-pressure combustion than with constant-volume combustion.

Such is not the case in actual practice, as confirmed by tests on the Westinghouse 9- by 12-in. single-cylinder test engine, Fig. *A*. The curves are for a constant maximum cylinder pressure. Not only do the lower compression pressures show higher thermal efficiencies, but they also result in lower smoke density.

Why this discrepancy between theory and actual performance? The probable explanation lies in the fact that the simple theory does not account for certain important factors. It is based upon an expansion and compression stroke of the same length and, in practice, the closing of the inlet valve is timed to utilize the inertia of the inlet air column, giving an equivalent longer compression stroke. More important, the lower compression pressure permits advanced injection, and more efficient combustion completed earlier in the stroke. A reduction of ignition lag as obtained with higher compression is not always desirable, as it militates against combustion efficiency, necessitating a retarding of injection for the same maximum cylinder pressure.

[a] See Fig. 9, N.A.C.A. Technical Note No. 569, May, 1936 "Boosted Performance of a Compression-Ignition Engine with a Displacer Piston," by Charles S. Moore and Hampton H. Foster.

[b] See Fig. 20, N.A.C.A. Technical Note No. 619, November, 1937, "Compression-Ignition Engine Performance at Altitudes and at Various Air Pressures and Temperatures," by Charles S. Moore and John H. Collins, Jr.

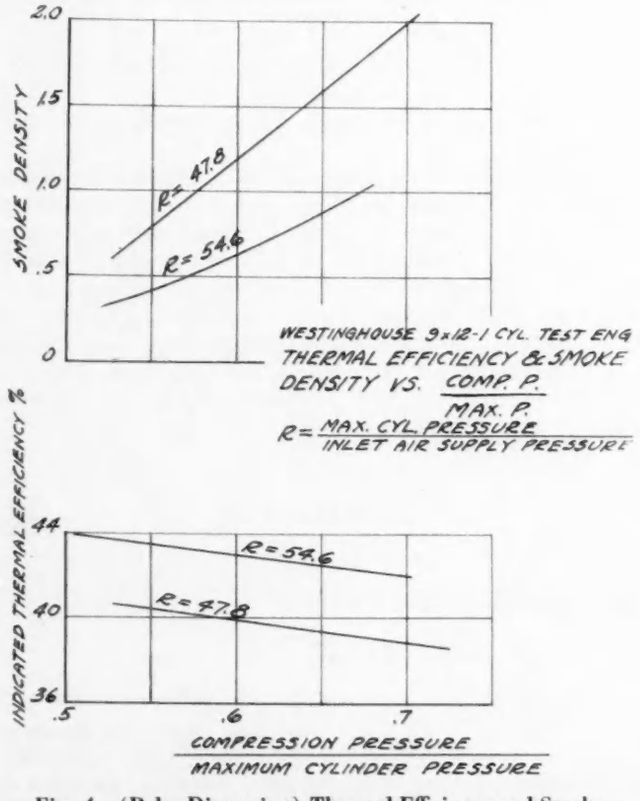


Fig. *A* – (Pyles Discussion) Thermal Efficiency and Smoke Density – Westinghouse 9- by 12-In. Single-Cylinder Test Engine

Accessory-Drive Problem of Aircraft Engines

(*Continued from page* 208)

concerned with the operation of the engine (such as starting, spark advance, carburetor heat control, mixture, choke, low-speed operation during "warm-up" period) were required to be fully automatic. The engine is maintained at constant speed within close limits by means of a sensitive hydraulic governor.

The development of such auxiliary powerplants is going ahead constantly and, naturally, it demands additional projects in order to retain the merits of the preliminary designs with substantial reductions in weight and size. Toward this end, air- rather than liquid-cooling, vertical crankshafts in place of horizontal, complete electrical rather than a combination of electrical and mechanical drives, and other variations are being built and tested.

We can briefly summarize the accessory-drive situation in the following few statements:

With few exceptions the engine designer has had to provide for the driving of all accessories by overloading the rear section of the crankcase; yet this practice has worked against the interests of both engine and accessory manufacturers as well as the plane builder and operator. To some extent this practice must be continued for special types of ships. In this case it would be very desirable from our viewpoint if the accessory drives remaining on the engine be located, as far as practical, close to the center of gravity of the engine. The use of flexible engine mounts results in the accessories being subjected to considerable whip which, being vibratory in nature, introduces severe strains in the accessory mounting flanges and has been responsible for several failures.

Starter mounting flanges should be redesigned to take care of the starter which has grown in weight as the size of the engine has advanced. Means of high-frequency vibration isolation may well be provided. The present starter mounting was standardized when 400 hp. was considered a very large engine and it cannot be expected rationally to be adequate for engines of over 2500 cu. in displacement.

For the larger ships, some form of central driving mechanism or gear box is essential. This mechanism must be of adequate size to take care of all group-two accessories, although it is our belief that it should be no larger than 20 or 30 hp., so as to limit the horsepower drain on the main engines should it be so driven. In those instances where more power is required, it would be safer and more economical to duplicate the gear box and drive from the other engines.

The largest ships, without doubt, will carry auxiliary engines which will drive either a large electrical or hydraulic source of power for individual drive of the accessories, or some form of gear box or combination of both. The larger the ship, the greater will be the tendency to individual drive.

Speaking as an accessory manufacturer, it would seem logical that the accessory engine be standardized at some unit power such as 15 or 20 hp. In those cases where more power must be provided, the engines should be installed in groups of two and even three. Not only would this arrangement provide greater reliability, but we believe that it is the only way of solving the economical problem associated with the development of highly specialized and expensive equipment.

Engine Installation and Related Problems in Large Aircraft

By Ivar L. Shogran

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THE first problem that comes to mind in this paper exists in the title itself: "Engine Installation and Related Problems in Large Aircraft." How large? So much development is going on in this country at the present time along these lines that one hesitates to consider any airplane large. To do so seems only to find a larger one in prospect so, to qualify the problems discussed in this paper, we will consider several that have arisen during the development of the Douglas DC-4, a four-engined transport landplane of 60,000 lb. gross weight, accommodating 42 passengers and a crew of 5 members. It is approximately twice the size of any regularly operated transport landplane in the United States at present.

Now this plane is particularly interesting in that it represents design features reflecting the aggregate experience of five of the country's leading airlines as well as that of the manufacturer. It is powered by four Pratt & Whitney R-2180 Twin Hornet engines incorporating automatic mixture control carburetors, and propulsion is through four Hamilton Standard hydraulic controllable constant r.p.m. full-feathering propellers. In general appearance the airplane somewhat resembles an enlargement of the Douglas DC-3 with the exception of four engine nacelles, triple vertical tail surfaces, and tricycle landing gear.

In this paper we will consider as problems related to engine installation several features essential to or affecting the operation of the engine throughout the plane from the cockpit to the engine as well as the engine installation itself. Our present records indicate that this represents from 17 to 25 per cent of the complete engineering labor entering the design of an airplane, depending upon the type.

Nacelles

One of the early problems in designing this plane was the arrangement of the engine mounts and nacelles. Each inboard nacelle contains, of course, one main engine supported on a steel tubular mount. The main engine mounts are conventional steel tubular structures. Attached to the front end are four Lord rubber bushings in 45 deg. radial positions. These Lord bushings, in turn, are attached to a tubular ring, bolting directly to and supporting the engine. The degree of vibration absorption required is gained by varying the hardness and quantity of rubber incorporated in the bushing. Aft of the main mount and ahead of the front spar of the wing in these inboard nacelles are additional mount structures containing auxiliary engines to supply power for all accessories not directly associated with, or necessary for, the operation of the main engines. These mounts, which in turn support the main engine mount, attach directly to the front spar. No batteries are used in the plane, and the two auxiliary engines each drive an alternator furnishing 800-cycle, 110- to 115-volt electrical supply. They each drive also one vacuum pump for instrument operating vacuum, and pressure for de-icer, one hydraulic pump for the autopilot, and one hydraulic pump supplying the pressure for the main hydraulic system to operate landing gear, wing flaps, and various other minor hydraulic units.

The tubular sections of the main engine mounts in the outboard engine sections are supported by monocoque structures attached to the leading edge of the wing. All four engine sections ahead of the fire wall, or in other words, the point at which the main engine sections terminate, are identical for complete interchangeability among all nacelles. The inboard nacelles, including both the main engine section and the auxiliary sections, naturally are longer than the outboard units. The position of the inboard propeller ahead of the leading edge is approximately 35 per cent of the chord. This dimension is decided by the physical dimensions required to install the auxiliary powerplant ahead of the front spar plus the main engine installation requisites. The outboard nacelles being shorter, together with a natural sweepback of the leading edge of the wing brings the outboard propellers well back of the inboard units giving excellent protection between the two propellers on each side from throwing propeller ice into each other. However, if the axes of both nacelles were maintained parallel to each other and to that of the fuselage, valuable space in the fuselage would be lost by the extent that the outboard propellers were located aft of the inboard propellers. This condition is brought about by the Department of Commerce requirement that no passengers, members of the crew, or main controls shall be placed in the fuselage within an angle of 5 deg. fore or aft of the plane of any propeller, measured from the axis of the propeller.

Adjustment of the nacelles seemed advisable, therefore, to reduce the extent of this prohibited area in the fuselage or to keep it to a minimum. The outboard nacelles were first inclined 6 deg. outward, as viewed in plan, in order that the area intersected by their propeller angles would be brought slightly forward for more advantageous arrangement in the fuselage. The inboard nacelles were then inclined outward 2½ deg. so that the intersection between the rear leg of their 5 deg. propeller angles and the fuselage approximately coincided with the corresponding point from the outboard propellers. This design allowed the prohibited zone from the inboard propellers to fall entirely within that of the outboard propellers, thereby preserving much valuable fuselage space. The slipstream converges to a single point aft to assist two- or three-engine operation. Wind-tunnel tests have indicated a negligible increase in drag due to the foregoing placement of the nacelles.

Cockpit

Next, how about the size of operating crew necessary to satisfy safety requirements and the majority of airline operators? This is a subject for a paper in itself. Some feel that,

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in an airplane of this size operating over well-organized airways, the pilot and copilot are sufficient to handle all flying as well as engine-operating functions, provided reasonable simplification of control arrangement in the cockpit, especially for the engines, is obtained. They also feel that, in landplane operation, it is important that complete control of the plane be within reach of the pilot in emergencies. Others feel that a flight engineer is absolutely essential to relieve the flying personnel of all problems pertaining to the engine operation during cruising flight. In answer to the foregoing we have attempted a compromise. All engine controls are directly available to both pilot and copilot. These controls are mounted on a pedestal between the two previously mentioned members of the crew. The two primary controls, r.p.m. or propeller controls, and the throttles are at the top of the pedestal. The throttles are duplicated so that it will not be necessary for either man to reach across the pedestal for these units. The r.p.m. controls are arranged in the center. Other essential engine controls are placed in convenient locations on the face and top of the pedestal for access to both men. Aft of the pilot and copilot a single seat is provided that may be rotated out from the left side of the cockpit to a location midway between and slightly aft of these two members. This seat may be occupied by a third man in the position of flight engineer who, in the center position, will have access to all engine controls and instruments and may thereby relieve the flight crew completely of engine-operating problems. We, therefore, have an arrangement that is suitable for either pilot-copilot engine control or the flight engineer. If it is felt that the flight engineer is not essential, the third member may, in immediate proximity to the other two, take care of all radio and navigational problems. In fact many combinations of functions are possible among the three men, allowing quite a flexible arrangement.

In order to design the cockpit so that it would be satisfactory under the preceding plan, it naturally was necessary to simplify the operation as much as possible and, consequently, these simplifications are reflected in the actual engine installation as well as other units pertaining to powerplant operation.

Oil System

To illustrate this simplification we will turn to the oil system. In the past our experience has shown that, with the necessity for larger oil coolers, warm-up in extremely cold weather has been made increasingly difficult or, in some cases, virtually impossible, even though the oil is bypassed around the core of the cooler through the muff as on the conventional type used. Apparently, under these conditions, enough cooling still takes place through the muff, lines, and tank to account for this difficulty of quick warm-up. An automatic oil-temperature regulating valve has been developed by one of the engine manufacturers which allows the complete cooler to be bypassed. However, this arrangement also allows the core to become congealed in cold weather. We have found, therefore, in conjunction with the preceding device, which in itself has been quite satisfactory for its own purpose of regulation, the only positive remedy to prevent congealing seems to be shutters controlling the airflow through the cooler. These controls have been provided in our other airplanes and have proved entirely satisfactory in preventing congealing and also for thawing out the core should it become congealed by inadvertently forgetting to close the shutters. Congealing has been experienced in flight on a 10-in. diameter cooler at temperatures as high as 15 deg. fahr. even with the integral cooler bypass valve operating, allowing the oil to pass through the muff only and, although the oil inlet temperature reached 185 deg. fahr., no tendency for the cooler to thaw out by the muff was experienced. The manually operated shutters, however, present a confusing condition to the pilots. The first evidence that the core is congealed in flight in the cold temperature is a rise of oil temperature. Under normal circumstances the last thing that would, of course, seem reasonable to the pilot to reduce oil temperature would be closing the shutters. However, this is the exact procedure necessary so that the core may be thawed out allowing it to be again effective as a cooler. So, to simplify the pilot's problems, we have designed an automatic shutter control operated by the tendency of oil to congeal in the core. When the oil becomes more viscous due to a decrease in air temperature, the resistance to flow through the cooler naturally increases the back pressure on the engine oil scavenge pump. This increase in pressure is employed to operate the shutters through a piston, to a closed position against a spring. The core again heats up, decreasing the resistance to flow and the shutters consequently are allowed to open. The preceding operation is to take place through a suitable range depending on spring pressure and inherent resistance to the flow of oil through the cooler. The shutters are located at the air exit of the cooler to prevent freezing under icing conditions. This method should eliminate the necessity for operating oil-temperature controls in the cockpit, either for warm-up regulation or for the prevention of congealing.

Fuel System

Now to touch on the fuel system of this plane. To maintain the duplication of safety in four engines it was felt necessary to provide the same degree of independence in the fuel system itself. Therefore, we have incorporated in this plane basically four single-engined fuel systems, each supplying one engine only. For each engine one 100-gal. tank and one 300-gal. tank is provided. The 100-gal. tanks are for take-off fuel, and the 300-gal. tanks are for cruising fuel. To continue the effort for simplicity in the cockpit, the selection between these tanks and their individual engines is accomplished by only one lever. It has been duplicated on each side of the pedestal for operation by either pilot or copilot. In the forward position of the single lever all four take-off tanks are connected to their individual engines. In the intermediate position all tanks are shut off and, in the rearward position, all four cruising tanks are connected to their individual engines.

This arrangement means that, under normal flight conditions where the entire fuel is not used during the flight, only three simple maneuvers of the gang lever are required for the entire trip. First, the lever is placed in the forward position before take-off. After take-off and when sufficient altitude has been obtained for cruising, cruising climb, or level flight, the lever is moved to the rearward position supplying all engines with cruising fuel. Then, before landing, the lever is again placed in the forward position connecting the remaining take-off fuel to all engines for any emergency that may arise during the landing maneuver. Supplementing and attached to the gang lever and mounted on the rear apron of the pedestal are four individual releasable levers allowing, under abnormal circumstances, variation of selection between either of the two tanks of each system to its respective engine. These levers operate in the same manner as the gang lever so that it is merely necessary to observe the position of the levers to know exactly what selection is in effect.

Further, in addition to the foregoing are four similar levers to furnish cross-feed selection through the suction side from one system to another. Four hand pumps are furnished, one for each system, and ganged together to one operating handle in the cockpit. Should it be desired through the cross-feed system to deplete one tank more than the others due to an engine failure or unequal consumption of one engine and to

supply its fuel to one or more other engines, it would only be necessary to move the cross-feed levers for the engines to be supplied forward and the individual lever on the tank to be held in reserve to the "off" position. Visual inspection of the lever positions again would show the actual selection. The preceding cross-feed system, or strictly speaking better termed a fuel transfer system, is somewhat unconventional but was selected over the type now in common use for several reasons:

First, the usual cross-feed system where the pressure side of all engine driven fuel pumps are connected together with valves open to protect against a pump failure offers the possibility of failure of all engines simultaneously should the common line of fittings in this line fail when in the "on" position. This failure would be disastrous, especially during take-off. Obviously such a system destroys the individuality of the four engines in so far as positive fuel supply is concerned.

Second, it was felt through the operating experience of the airlines that pump failures are sufficiently remote to consider them a secondary hazard. The engine-driven fuel pump also can be replaced immediately by the operation of the hand pump should a failure of that type occur. Third, more instances have occurred in the failure of the hand-pump bypass relief valves in the systems than in the fuel pumps themselves. This failure allows the fuel to be diverted back through that system rather than to the engines. In the preceding case, of course, pressure to all four engines would be lost. Fourth, with the individual systems employed it would be necessary to hand-pump the fuel to other engines in case of one engine failure. As the probability of engine failure seemed greater than fuel-pump failure through present experience, the suction side was favored. Fifth, it was felt desirable to reduce to a minimum the length of line in the ship carrying fuel under pressure to eliminate a possible fire hazard. Sixth, it was decided that circumstances under which operation of fuel transfer is required would be of such a nature that undue burden would not be placed upon the crew.

To relieve the crew of as much effort as possible, the larger the airplane, the easier must the controls operate due to the inherent complexity of the system. Therefore, it was deemed necessary to design our own fuel valves to satisfy this condition. These are special fuel valves of the poppet type, each valve being lifted by a cam to reduce the loads. Preliminary tests have indicated that the force required for manipulation is especially low and that they seem to give complete satisfaction as valves. Their ease of control has made it possible to operate four units by a gang lever without unreasonable effort which is not possible with a conventional rotating-seat valve, especially in the large sizes required. Also, the deflection of the fuel-valve control system between the cockpit and the units ceases to be a factor of importance. This point has been a serious handicap in many fuel systems on large airplanes requiring remote drives for fuel valves. Large operating loads, even with relatively small deflection of the control systems, made it impossible to be sure that the valve was in the exact position as indicated by the handle or the lever in the cockpit. This difficulty we feel has been eliminated. Two of the poppet valves are grouped together in each engine system for selection between the two tanks of each system, one being open when the other is closed. In the intermediate position, both valves are closed.

Due to the large amount of fuel carried, a refueling system is provided to fill all fuel tanks at the rate of 100-gal. per minute from the ground. Beneath the wing on each side of the plane is a receptacle with a coverplate to which the individual refueling lines from the four fuel tanks on each side terminate. These lines have quick detachable fittings for the refueling hose. They lead up from the receptacle, entering the tanks at the top so that no defect in the refueling system can result in loss of fuel during flight. Fuel gages are located in the bottom surface of the wing adjacent to the refueling attachments, as well as in the cockpit, so that the attendants will, at all times, be aware of the amount of fuel in the tanks. Fuel caps are furnished on the top of the tanks flush with the wing for normal filling if desired and are provided with safety valves to prevent damage to the tanks should they be subjected to excessive pressure during refueling. Expansion space is provided in each tank to accommodate a 25 deg. fahr. rise in the temperature of the fuel to guard against overflowing through the vents between the time the tanks are filled and take-off or engine run.

A very complete mockup of the fuel system was made and tested with various combinations of tank and cross-feed selections. The pressure drop through the lines was recorded during all tests with quantities of fuel passing through the system commensurate to that actually used by the engine, as well as with increased flows, to prove that ample margin was inherent in the system. Checks were made with both engine-driven pumps and hand pumps. Among other special conditions checked was the effect of rearward acceleration, during take-off, on the fluid in the fore and aft lines. This action, in some systems designed with close margins, has caused a drop in fuel pressure during take-off, a most disturbing condition.

Engine Controls

The arrangement of the general engine control system becomes quite a problem in a large airplane in order that close adjustment may be maintained between the cockpit and engine over relatively long runs. The distance from the cockpit to the outboard engines is approximately 70 ft. The control system selected to reduce deflection and play is a combination of push-pull rods with bell-cranks and cables. From the hand-operated levers on the pedestal down to a point beneath the cockpit floor, push-pull rods are used to accomplish compactness with convenient adjustment in this rather complicated unit. From this point back to the front spar and out the front side of this member to a position aft of each nacelle, cables and pulleys are employed. Our experience indicates that cables present many advantages over push-pull rods for long leads. They are lighter, not subject to variation of adjustment caused by deflections of the general structure due to air loads, and they eliminate play from accumulated tolerances and wear in bearings, pins and so on. Our only difficulties in the past with cables have been in cases that lacked proper alignment requiring heavy fairleading. This point we feel has been covered by careful arrangement of the cable system. A complete mockup was made of the cable controls to work out this problem satisfactorily. From the front spar aft of each nacelle push-pull rods lead forward to the engines in order that the engines may be removed or replaced without disturbing the rigging in the balance of the system.

In several cases the engine controls are combined, one lever in the cockpit operating the units on all four engines simultaneously. In conjunction with these controls, individual controls also are provided in such a manner that each unit can be operated individually if so desired, should circumstances on one engine require different adjustment from the rest. Also, in most instances where automatic controls are used, manual override controls from the cockpit to the engines also are provided for emergencies.

Exhaust System

Considerable study was given in design work to the collector ring and exhaust system. Several types were laid out

before the final selection was made. The major points for consideration were: position of the outlets for noise reduction in the cabin, concealing the view of the exhaust flame from the passengers, avoiding the proximity of the outlets to the gasoline-tank compartments, and the adaptability to the cabin-heating system. Also, as another point, our previous experiences have shown that with the larger engines the single tail-pipe becomes increasingly difficult to make serviceable. As the diameter increases, of course, the ratio of material thickness to diameter becomes very small unless much heavier material is employed than commonly used at present, with considerable increase in weight. Also, the ratio of external cooling surface to cross-sectional area or volume of exhaust carried decreases rapidly as the diameter is increased, causing further difficulties due to excessive temperatures unless the tail-pipe is well exposed to the slipstream. This difficulty is accentuated by the increasing tendency toward leaner mixtures.

With the preceding limitations in mind, a double-outlet system was decided upon with the collector ring in two halves, reducing the size of all units. The exhaust on the inboard side of each inboard nacelle passes aft through tail-pipes located in the lower portion of the nacelles each containing a steam boiler for cabin heating. These tail-pipes are circular in cross-section and are contained in shrouded stainless-steel ducts which have a cross-section similar to the letter D, the flat section being formed by a removable cowl sheet heavily perforated with louvers to promote circulation of cooling air around the pipes. The exhaust outlets from these sections are beneath and slightly aft of the leading edge of the wing. The outboard outlets of the inboard nacelles, as well as both outlets on the outboard nacelles, are short elbows and are located just aft of the collector-ring section itself, thereby practically eliminating tail-pipes at these points. The two halves of the collector rings are supported to the engine mount structure in a manner that will allow suitably for deformation from expansion and contraction. All the short sections running forward to the cylinders have double ball-and-socket joints to allow the engine complete flexibility. Full consideration toward the support of the tail-pipes also was given to prevent the transmission of loads to the balance of the structure by expansion or contraction of these units. One end is supported rigidly, whereas the remainder of the supports are links allowing the pipe to extend axially with freedom. The exhaust from the auxiliary powerplants terminate in the main-engine tail-pipes. While furnishing heat to the boiler on the ground with the main engines inoperative, some muffling also will be gained for these small engines.

Cowling

A rather conventional cowling was selected for this plane. The anti-drag ring is in three sections with fixed nose, removable sheets over the cylinders, and cowl flaps hydraulically operated forming the rear section. The engine section cowling is in four sections for each nacelle attached by Dzus fasteners with no fore and aft formers. Sufficient accessibility is provided in the main-engine sections so that, by removal of the bottom cowl sheets, a man may stand upright in the section for such adjustments as may be required on the rear of the engine itself.

Induction System

The induction system in this airplane runs from the nose and is attached to the ring cowl over the top cylinders back through a preheat valve and then down to a carburetor adapter containing the pressure doors of the automatic carburetor. The cold air is taken through an opening located in an "eyelid" projecting down from the normal shape of the cowl nose. From our previous experience the "eyelid" has helped considerably to prevent the spillage of heated air out from the front of the cowl entering the carburetor air inlet. The probability of the cold-air inlet becoming filled with ice or soft snow always has been a point of danger and, to guard against this danger, we have provided an emergency door at the top of the air duct on the surface of the cowl. This door may be opened at will by the pilot, taking in air with no ram but free of packing snow. The preheater valve is operated by an automatic servo control and, in order that the loads be kept to a minimum, a rotary balanced-barrel-type valve was chosen. The servo unit is actuated by a thermostatic element in the induction system to maintain a constant carburetor air inlet temperature of 90 deg. fahr., and is connected to the cockpit through an overriding control. The preheat is taken from around the collector ring through a duct formed by the inner collector well and extensions of the outer cowl sheets over a portion of the collector ring itself.

Auxiliary Engines

In large airplanes the installation of the auxiliary engines becomes a problem in many respects even more involved than the main engines. All operating functions are automatic and the engine must cool through its own inherent cooling system on the ground with the plane stationary, when it is supplying electric current, as well as in flight. As all airplane accessories are driven by these units, the majority of line installation problems are resident in their sections.

Aerodynamics

In so far as the powerplant items of this design are concerned, we have made every effort to maintain aerodynamic cleanliness on the top of the nacelle and wings and, to date, projecting external scoops have been avoided. The inlets for the oil coolers are in the leading edge of the wing. The coolers are mounted parallel to the leading edge, the air passing through a right-angle bend in the inlet duct, then through the cooler into the nacelle aft of the fire wall. The exit ducts or gills are in the bottom of the nacelles. In tests on other planes we have found the outlet from the cooler on the upper surface of the wing to be very detrimental as might be expected. On a single-engine airplane tested it was found that the outlet on the upper surface of the wing destroyed the flow completely from the fuselage out to the inboard end of the ailerons, causing early stalling of this wing upon landing.

In general, as a résumé, it may be said that the installation of the engine itself in the nacelle does not differ greatly from that followed in a plane of considerably smaller size. The majority of the problems exist in the airplane itself to furnish simplified control of the engines and to avoid undue complexity which may be brought on by additional equipment.

After all, the same pilot must operate the large plane as the small one, and every effort should be made to keep his problems within reasonable bounds and, for obvious economic reasons, in commercial airplanes the crew must be kept to a safe minimum. No attempt has been made to discuss or make predictions relative to extremely large airplanes that may require entirely unconventional methods of engine mounting. Much of the information presented here, however, may be extended to considerably larger planes with the possible addition of a permanent flight-engineer station.

It should be borne in mind that the problems discussed in the preceding paragraphs are by no means all those faced in the powerplant design of an airplane the size of the DC-4, which incidentally is still in the development stages and, as anyone having worked with powerplant installations realizes, they are subject to minor modifications dictated by actual flight experience.

Motor-Car Engines in England

By Alex Taub
Vauxhall Motors, Ltd.

VARIATION in engineering practice between European and American motor cars is to be expected. Many of these differences are brought about by local conditions and must be accepted. However, there are practices that vary from the American that do not justify themselves by result or local conditions. The two outstanding are bore wear and carburetion.

This paper deals only with the high spots of these two differences.

WE feel that we could do no better in this paper than to lay before you problems current in England on engines, representing work and discussion going on. Obviously any engineer going into a new field in an old endeavor will find many differences and disagreements to the practice to which he has been accustomed.

It would indeed be foolish to attempt to change the group practice to suit the newcomer's opinions. He must, of course, adapt himself and carry on in the current trend. However, if the newcomer is charged with the responsibility of getting results in keeping with the present American practice in a different country, he may have very little choice as to whether he should "buck the tide" or not.

We find many things in which there is a difference in practice; however, just as long as the functioning is good or even fair, no change is warranted. There are unfortunately differences in practice combined with results that are below par.

The general impression in America regarding the British is that they are slow to change. This opinion is definitely not so – all engineering practice and thought in England are in a state of flux – and changes are being made as fast as they can be absorbed.

English motor cars start badly, lack flexibility, and wear poorly as compared with American cars. These are facts and represent the bulk of our attack on English practice. In spite of this condition the British heavy-duty engines are superb. They are heavy and costly – but fine powerplants, perhaps among the best in the world. But let us get down to motor cars and their problems:

Up to and including 1936, the lubricating oil used in British motor cars varied upwards in viscosity from S.A.E. 40. Perhaps the biggest seller was S.A.E. 60. Need more be said on bad starting and poor bore lubrication? To break this oil down to reasonable viscosities required finesse and some fancy shadow boxing. Almost any lubricating technologist admitted that the oils used were too heavy, but technology and sales are two different considerations, and grades were not being used generally. It was impossible to specify grades when only brands existed, and there are over 50 brands marketed.

At Vauxhall, in conjunction with Mason, our chemist, we established specifications for engine oils based on what data we could lay our hands on that we felt would represent a good oil. Naturally these data were based on flat viscosity oils. We introduced a factor called viscosity ratio to lead the oil companies into line. This viscosity ratio was determined between the viscosity at 70 deg. fahr. and that at 200 deg. fahr. This work gave a picture with which, if necessary, we could have gone to the public. However, it was unnecessary and we were happy to have the oil companies agree to mark their containers with the numbers 20-30-40. Thus, in 1937, English engine oil is graded and, in the main, it is of flat viscosity. Starting is better and I believe that everyone is satisfied.

Light Oils Opposed

Opposition to oil light when cold came from strange places, papers having actually been written to prove that, when the light oil was used in an engine kept cold by circulating cold water continuously through it, greater wear ensues – provided that the amount of light oil was held down to the same as the heavy oil. No engine need operate cold, and certainly we must be ingenious to keep the light oil supply down to the viscous throw off of heavy oil.

The general opinion in England is that the small engines that are used, because of their size, are subject to more wear than the American larger engines. This opinion is certainly not so. We have seen too much of engines of all sizes with badly worn cylinders at short mileages. The small engine in England does have trouble, but not due to size. It is a matter of misunderstanding and economics, as we shall show.

Small engines are used, not alone to meet tax conditions, but because of the selling price. A 10-hp. tax rating car – 75 cu. in. engine approximately in a 2000-lb. car – sells for $800 to $1200. A car of the Chevrolet, Plymouth, Ford size sells for $1500 to $1800. With English incomes about one half of the American, it is plain why the small car is favored.

The tax is a nuisance, not from the size standpoint, but from a design standpoint – making it necessary to use an unsatisfactory bore-stroke ratio.

Getting back to bore wear, from our observations we are satisfied that bores generally were under-lubricated and that the piston-rings available could not cope with adequate lubrication and would not control blowby.

English rings are still cast centrifugally into pots, sliced off and heat-treated. If better rings are needed, they use higher priced alloys. Ring manufacturers have told me that, if they wanted to make a low-quality ring, they would follow the American practice. Platitudes, however, never work in

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engines and we needed rings, good ones, and now we are getting them.

Corrosion as the only factor of bore wear was the accepted belief by most of the engineers in England. No one quarrels with corrosion as an important factor. However, we believe that the American practice was resulting in better bore conditions than those enjoyed by European cars, this practice being:

(1) Thermostatically controlled cooling for the water, not less than 145 deg. fahr. and 165 deg. fahr. preferred.

(2) Crankcase ventilation to dissipate gas blowby and to eliminate contamination of the lubricant.

(3) Adequate lubrication for cylinder bores, meaning a copious supply.

(4) Complete control of lubricant at the piston-rings, not at the source of supply.

(5) Control of gas blowby at the piston-rings.

Considerable work has been done by the I.A.E. Laboratory and others on corrosion due to continuous cold operation. However, it is not necessary to have continuously cold operation unless an engine is not permitted to warm up at all. Engine wear after an engine is warmed up is a much stiffer problem, and there are many engines with a wear of 2500 miles per 0.001 in. that are started in a warm garage and only given one start per 50 to 100 miles. This is the type of wear that the user meets every day and needs consideration, and it is this type of wear in which we are interested. The smaller English engines do not warm up quickly and tend to overcooling. The effect of oil as a deterrent for wear from any cause except dirt and abrasion has been established by all investigators and is accepted by all. There is some difference of opinion in England as to whether or not oil can be used in desired quantities because of the belief that piston-rings made to British standards will not give the desired control. There is substance in this belief, and for this reason the American practice in piston-rings rather than the British was recommended by us.

Fig. 1 gives an interesting comparison of English and American rings. Time will eliminate this difference.

We have been a party to, and have observed considerable development in the American piston-ring. Again, since their use has been highly satisfactory, we see no reason for not carrying on and adopting that practice in England. However, before going into an analysis of American ring practice as compared with British, it would be well to outline further facts on the general problem of bore wear, as we have contacted it in England.

Experience has taught the industry that the factors affecting bore wear are complex and, although an engine may include all that is known as a protection against bore wear, still wear occurs, and under conditions considered favorable. These engines may be warmed fully and with a minimum of starts operating on a fast schedule or mixed schedule. It is this fact that inspires me to argue against the overemphasis of corrosion.

Fig. 2 is a graph indicating cylinder wear rate for a most interesting case. This engine was run for 20,000 miles under favorable conditions. The wear varies in cylinders, the maximum being at the front, the minimum at the rear.

The amount of oil on each bore is the same – as close as it is possible to get it. Further, the amount of oil on the bores is far in excess of any pressure-fed engine, being designed to equal the amount of oil thrown by splash lubrication. The engine incorporates a water thermostat with a terminal temperature of 160 deg. fahr.; the blowby is down to 0.5 cu. ft. per min. up to 4000 r.p.m. The crankcase ventilator maintains

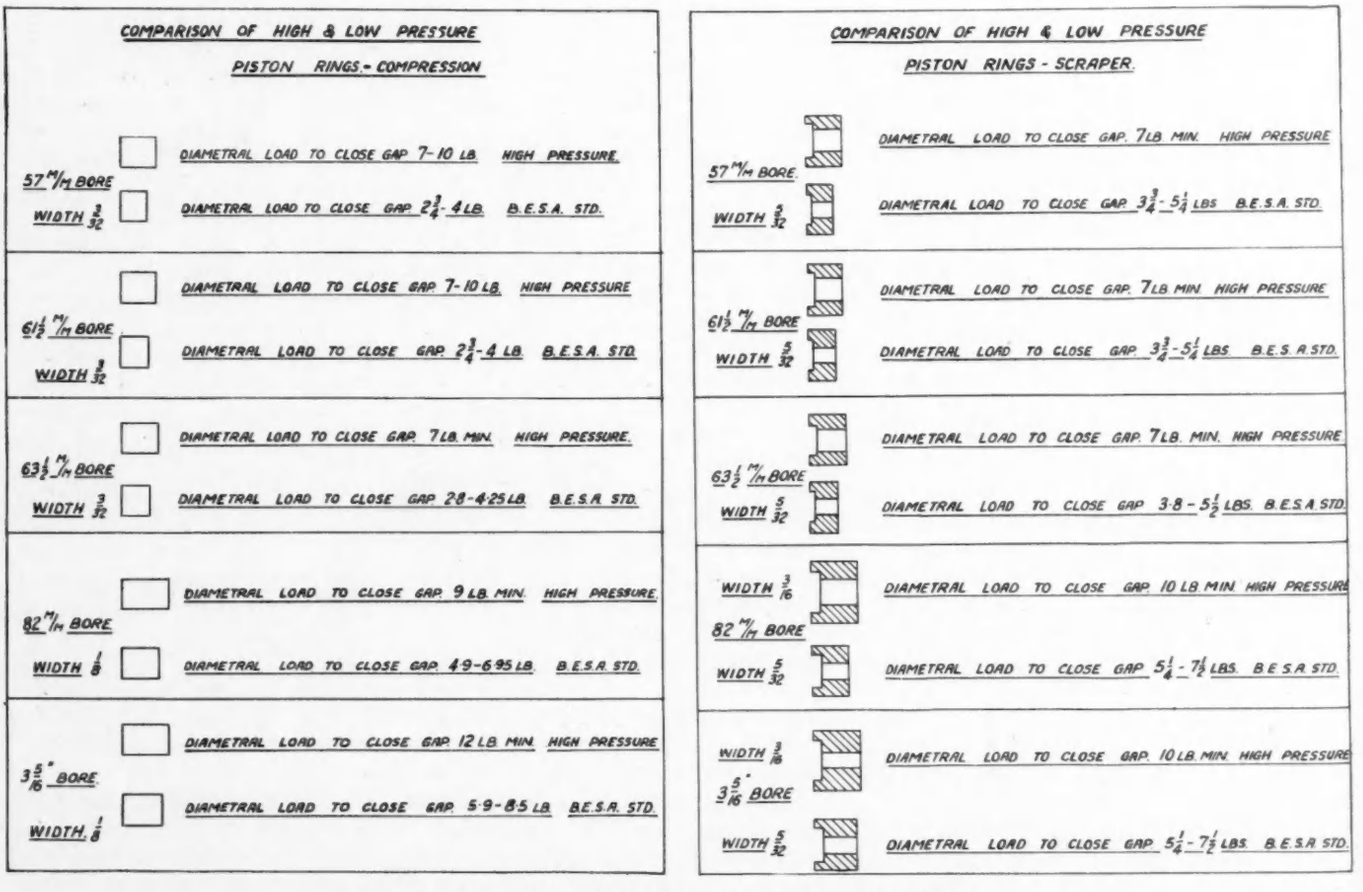


Fig. 1 – Comparison of English and American Piston-Rings

a vacuum of ½ in. of water in the oil sump and crankcase, thus the contaminating gases are reduced to a minimum.

What are the defeating variables, and particularly what are the variables that can exist in one engine?

A year ago when discussing bore wear with W. L. Fisher of the *Automobile Engineer,* he made the statement that carburetion was just as important a factor in bore wear as any other. C. G. Williams of the I.A.E. (Institution of Automobile Engineers) in discussing my last paper on this subject, stated that he was suspicious that the transatlantic manifold heaters may have something to do with the difference between American and English bore results. A. W. Pope of Waukesha Motor Co., wrote us in the same vein. I innocently stated that I did not believe that English carburetion was that bad. These gentlemen saw further than I did, because today we must face the fact that a difference between 12:1 mixture ratio and a 14:1 mixture ratio, although in the part-throttle range, may effect bore wear at the startling rate of 3 to 7 times. This statement is according to the findings of Gardiner of G.M. Research. It is said, "may effect," because the results, although carefully obtained, are on one type of engine, and other conditions of variation may mitigate this effect and, further, this test engine was an under-oiled engine. Lean mixtures favor low wear. This statement is upsetting because one would expect the oxidizing atmosphere of lean mixtures to affect bores adversely. However, in spite of this condition, the lean mixtures show not just favorably, but outstandingly so. In what way does a lean, and therefore, dryer mixture help, and incidentally why as much as 7:1 on an under-oiled engine? We believe that the oil film is weakened by dilution of a rich mixture. This oil film being over-thin in this case, is more sensitive to attack, and thus a difference in mixture richness of 16 per cent may have a major effect. The foregoing mixture ratio wear tests also were carried on with a minimum of adverse conditions. The car was kept in a heated garage and mileage run off with long intervals between starts.

Examining Fig. 2 again one sees a variation of unsatisfactoriness at the front – to fair at the rear. Does this condition tie up with mixture ratio? Well, here is an important fact. Most English carburetors that I have checked over during the past year show as much as 30 per cent mixture variation between front and back cylinders.

Today at Vauxhall distribution from the carburetor is given special consideration and is corrected to avoid this error, and no calibration is started until this part of the work is cleaned up. The carburetor used on tests as shown on Fig. 2 was not corrected and incorporates this distribution error. The drop in the curve at No. 5 cylinder is most interesting because it is known that, in this particular job, No. 5 is richer than No. 6, due to manifolding. I do not insist that this is the reason for the relative difference between Nos. 5 and 6, but it is an intriguing difference.

Examine Fig. 2 again. This engine, like many others, has its water pump in front and the coldest water in the system strikes the cylinders from front to rear. Now, is the water-temperature variation responsible for more wear in front than in the rear, or is it both mixture and wall temperature? Wall temperature does affect wear, but what can be affected by wall temperature after an engine has reached a stabilized temperature, and is relatively warm? Does the wall temperature under normal conditions still effect an incipient corrosion and a small temperature difference become important at the border line, or does the higher temperature mean more clearance and more oil? The safe conclusion is to say that high wall temperature uniform throughout should be added to the list of "musts."

Very few engines have rings pinned. The high-tension

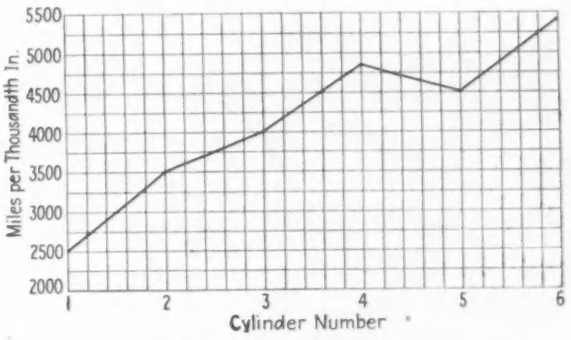


Fig. 2 – Cylinder-Bore Wear

ring lends itself to this practice and unquestionably, where high distortion exists, the rings should be pinned against rotation to lower blowby, which can destroy all other protections against wear. We do know that this practice can turn a bad job into a good one.

Pressure behind No. 1 or the top ring also may have a bad influence on wear. We have seen sharp depressions in the bore 0.030 in. deep a little wider than the ring – opposite the top ring. The pressure behind the rings is affected greatly by the fit of the top land in the bore. The length of this land is important and is a matter of design. The fit, however, may be a matter of manufacturing variation. Temperature variation which affects the diameter of the land, also affects its clearance. The temperature of the piston top may be affected by several conditions. The outstanding one may be wet mixture, the evaporation of which lowers the piston-head temperatures, and thus increases clearance between it and the bore.

Another outstanding possibility is spark timing, which materially affects the relative phasing of the maximum pressure point and the relative position of the ring in the groove. Maximum pressure normally comes just after dead top-center. If the spark is over-advanced, the pressures are higher and earlier.

Can spark advance affect bore wear? Investigators claim as much as 1½:1 to 2:1 increase in wear rate with 5 deg. over advance.

Since there may be as much as 7 deg. variation in the spark advance due to manufacturing variation in the distributor, the 5 deg. is not very much and may represent a part of the reason that the bore-wear problem as a target is constantly shifting. The reasoning that we have given as to why spark advance affects wear may not be correct; it may be some complex out-of-phasing of the burn with the piston movement increasing the general blowby. Whatever the reason, it is just one more thing to think about, and the ignition variations must be dealt with.

Another very important factor is the fuel itself. Unquestionably much work will be done in the near future to rate fuels experimentally for their effect on the bore wear.

Fig. 3 is a wear chart of a German engine showing comparative wear with two kinds of fuel; one, the bad wear picture, shows German national alcoholized fuel which is of a general low order, but is commonly used in Germany. The other is a non-alcohol fuel of aviation quality. Here is something worth investigating. What are the facts? These engines have not been treated for quick warm up of fuel or water, and German rings are not particularly effective for blowby and oil control. The oil on the bore is not abundant and what oil film exists may be weakened by blowby.

Any difference in wetness of the fuel in the combustion-chamber would certainly have the same effect as we have indicated for the rich mixtures. There is, of course, a great

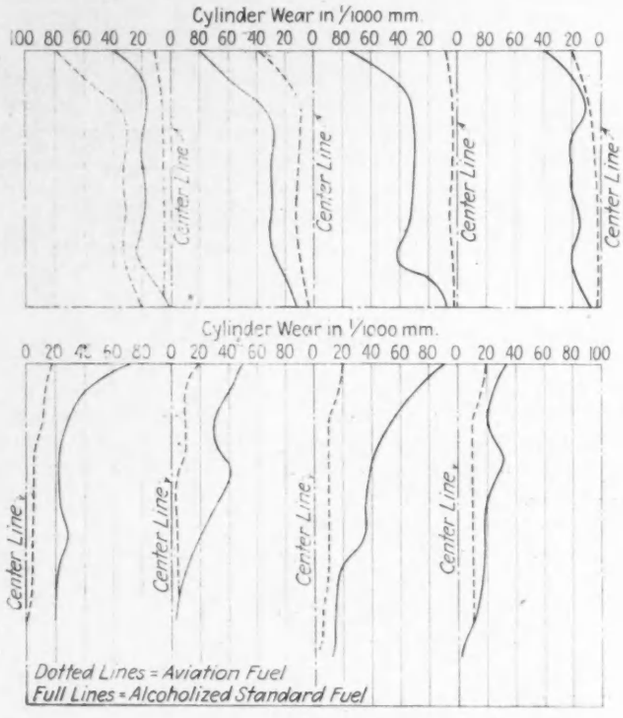


Fig. 3 – Comparative Cylinder-Bore Wear with Two Kinds of Fuel

deal of difference in the self-vaporizing qualities of a low-grade alcohol fuel and a fuel of aviation quality. It may be the temperature difference due to latent heat that affects the bore wear, or it may be a chemical reaction due to the admixture of alcohol. I expect to hear deep rumblings of fuel technicians, in protest to my unwarranted suspicions. Perhaps they will soon tell me whether fuels are to blame or whether the fault lies with the engine design in leaving something undone. One fact is well known, alcohol has the least lubricating value of any material – even less than water.

Before leaving the question of fuel responsibility for bore wear – at least that part of wear for which it may be responsible – it would be as well to consider the prevailing mixture ratios during the maximum-wear operating conditions which are starting from cold and warm up from cold. The mixture ratio for cold starting is about 1:1; the mixture ratio for warm up is from 8:1 to 9:1. Now, if the difference between 12:1 and 14:1 is from 3:1 to 7:1 in bore wear, what must be the result when 1:1 is used at full choke and 8:1 or 9:1 for long warm-up periods?

The importance of careful investigation of starting is obvious as a few doses of this rich mixture would go a long way in destructive effect. Once the engine is started, an endeavor should be made to warm up on the leanest possible mixture for the shortest possible time. Good cold distribution is essential with all the possible heat on the mixture during its travel from the carburetor to the combustion-chamber. To what extent corrosion has been credited or discredited with wear that rightly should have been pinned on carburetion, I do not know. However, I am certain that, regardless of anything else, destruction of the oil film by wet fuel should be given more consideration. Perhaps this consideration points out a real advantage for the automatic choke.

At this stage another summary of conditions required to offset bore wear can be made:

(1) Thermostatically controlled cooling for water, minimum 145 deg. fahr. with 160 deg. fahr. preferred.

(2) Crankcase ventilator to eliminate effects of blowby in oil sump.

(3) Adequate oil supply to cylinder bores – meaning a copious supply.

(4) Complete control of lubricant at the piston-rings, not at the source of supply.

(5) Control of gas blowby at the rings.

(6) Leanest possible mixture ratio for general operating conditions.

(7) Particular detail attention to the type of fuel used to be sure it is vaporized properly. Some fuel may require special considerations.

(8) Metal temperature must be uniform and as hot as possible. This item may require special consideration of the water circulation system.

(9) Spark-advance mechanism should be void of errors among cylinders of one engine, and advance or retard should be readily adjusted by competent service people only, not the customer.

Throughout all considerations the oil film appears to be the most important item because, when present it protects, and because every engine function attacks this film either to weaken it or to destroy it entirely.

Ring development over a period of years has been a most interesting study, and considerable progress has been made in the last 17 years in America. From 1920 to 1927 American rings were of a uniform radial pressure and hammered to this pressure. Rings of this period on an average had a diametral tension of 7 lb. to 9 lb. The radial thickness to diameter ratio was 28:1. Piston-rings were finished with sharp corners and there were several methods of casting them; the only specification limitation was that the product must be free from shrinkage strains. From 1927 to 1932 rings were cast individually for longer life and less blowby. This was the period of development of crankcase ventilators. The average tension of these rings was 9 to 11 lb., the radial thickness to diameter ratio was 24:1, and hammering for shape was eliminated.

The more consistent characteristics of the individually cast ring easily took the place of the haphazard hammering.

From 1932 to 1937, tension has increased to 16 to 18 lb., individually cast with the radial pressure pattern controlled to give the maximum effect on a given engine.

During this period blowby had become a serious factor and with maximum pressure, temperatures, and speeds up, the piston-ring was being asked to do a great deal. Modifications were made to meet these conditions, and the American rings represent these modifications. Perhaps the most important development in American piston-rings is the recognition and development of pressure shape. This development came about through an endeavor to shift the "break" in the blowby curve.

Fig. 4 is a typical blowby curve. The curve is fairly high through the range from 1 to 1¼ cu. ft. per min. but, at 3400 r.p.m. the break occurs and at 4000 r.p.m. the blowby is at 3½ cu. ft. per min. Now experience has indicated that 0.5 to 0.75 cu. ft. per min. throughout the useful range is ideal, being a quantity that our normal ventilators can handle and draw out of the crankcase.

A common sequence of excessive blowby is fracturing the ends into several pieces due to the hammering of these ends on each other. It is needless to discuss the destruction that follows on fractured rings.

Considerable credit is due to the American piston-ring makers for, in spite of the difficulties of distortion, they have worked out a combination that succeeded in making very good jobs out of some bad ones.

We have always advocated an excess of oil, that is excess as compared with the average pressure-fed engines here and abroad. Sufficient oil is needed to establish a fresh film between all metal surfaces for every revolution. With a copious supply of oil and good control, a constant change is going on in the oil film.

Control, although definitely at the rings, is aided by piston design. A weak piston may interfere with the operation of the rings; a piston must be structurally rigid; its shape must be such as to meter the oil on the bores for the piston-rings in order that a given combination may not have more to do than it was designed for. A good piston will lay a new film of oil like butter on bread, and the rings will cut the film to final smoothness in the process. The surfaces are protected, and the oil film is refreshed constantly.

It is quite easy to see the necessity of high-tension rings for a function of this kind – one that must go on and on with a minimum of variation despite adverse conditions. High-tension rings, properly designed, will do this job. Low-tension rings may not.

To illustrate the value of the American type of ring we offer Fig. 5 which again shows the wear diagram discussed in Fig. 2. Curve *B* shows the oil consumption over 20,000 miles for the same test run. With 0.0065 in. wear in one cylinder to 0.0035 in. in the best, it is found that the oil mileage starting at 1100 miles per imperial gal. at 1000 miles, increasing at 20,000 miles and showing over 6000 miles per imperial gal. at this mileage. Surely this is an interesting picture and, from the standpoint of the user, a very good one. Such a result could only be obtained when high-tension rings or their equivalent are used. In this particular engine the amount of oil on the bores is considerably in excess of normal; yet these high-tensioned rings backed up, and this point is very important, by a decent sort of piston, gave these outstanding oil-control results.

Compare this picture with a typical one where low-tension rings and low oil supply are used. 20 hr. to consume the first pint of oil, 2 hr. for the second pint, and twelve pints in the next 4 hr.

Piston-rings function as oil control by virtue of the sharpness of their scraping edges; yet it is an English practice to break the corners. Pistons will rock, deflect, and twist and,

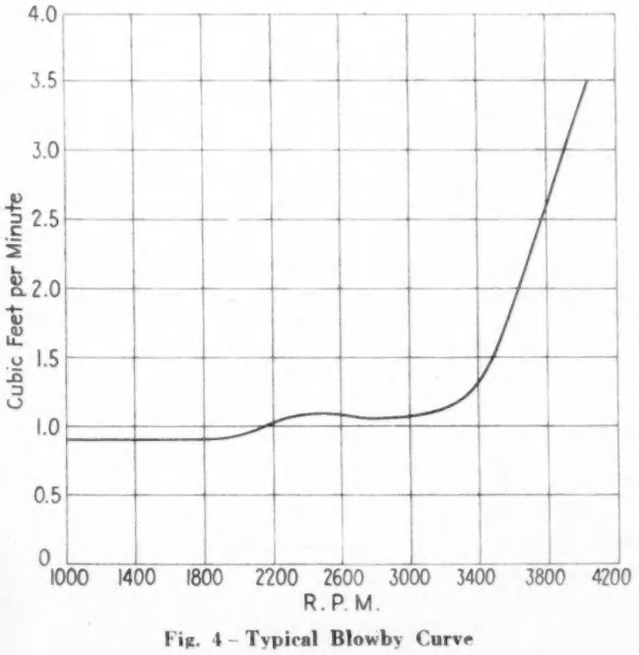


Fig. 4 – Typical Blowby Curve

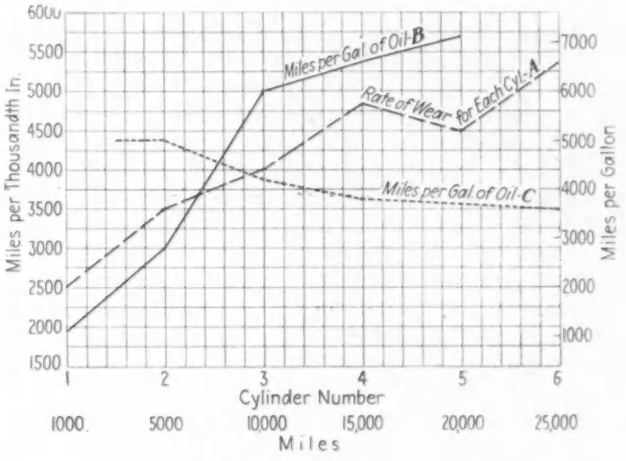


Fig. 5 – Wear Rate and Miles per Gal. of Oil

if these movements are such as to dull the edges by crowning or some other form of destruction, the rings will fail to give satisfaction. It is well known that skilful movement of a knife or tool on an oil stone will create a fine edge. Likewise, we know that, if the wrong movement is added to the stroking, the edge is taken off. This analogy applies to piston-rings. Oil control is a delicate balance. It means a great deal more than just keeping the oil out of the combustion-chamber. For instance, a combination of rings could be used consisting of all compression rings with a high tapered face. For a while these rings would do well, but not for over a long period of hard miles.

These rings would control by pushing back the oil on the down stroke that they helped to drag up during the up stroke. Too large a percentage of this oil would be the same oil, and it would carbon up and soon start abrasive action and ring-sticking.

For this reason the drain type of oil control is preferred because this type of ring "circulates oil;" thus there is a very large percentage of new oil that moves up to lubricate the upper rings and the top of bore. For this reason the capacity of the oil ring and piston to pass or circulate oil is important. Structure or strength is the limiting factor to drainage in both the piston and the ring. Slots in the ring are limited by the breakability of the oil ring and slots in the piston are limited at the ring-groove because of collapse of the lands, pinching the oil ring and making it useless. Carefully placed holes in the piston have replaced the old long slot for this reason. Holes eventually may replace slots in the rings.

The outlook on cylinder-block iron has not been reasonable in England. Because cylinder wear has been prevalent, an iron mixture is used that, although it adds nothing to bore life from its hardness, machines with difficulty. To offset wear by hardness alone requires the hardness of nitrided steels or their equivalent. Materials that, of themselves, will resist wear do not lend themselves to modern cylinder-block castings.

The present material used in cylinder-blocks in England not only shortens the life of tools but causes distortion of barrels while they are being processed. Honing is made difficult because this type of material picks up in the hone, scratching the bores rather deeply.

Compatibility of piston-ring iron with cylinder-bore iron is due for more study in England. There is an undetermined difference between English and American piston-ring iron. The English iron gives a higher modulus value than the American. However, the American iron is tough and apparently is more compatible with the cylinder iron with which it

is used, than is the English. This is evidenced by the sensitivity to cold scuffing of English high-tension rings.

To date this trouble has been overcome by swilling the piston and rings in lard oil before assembly and by lowering the break-in loads.

With rings proportioned to American standards, high modulus values mean nothing. Toughness is to be preferred. Incipient damage done by cold scuffing may result in serious hot scuffing.

That rings actually flutter we are certain, although the exact reason behind the flutter may be obscure. The question of the irregularity of the bore has been suggested as the exciter of the movement. This suggestion I do not believe because I have seen too much evidence of ring flutter in an engine that was notoriously straight and round. I still believe that the pressure behind the rings which rises and falls with the piston movement at the top of the pressure stroke, acts as the pick on the banjo string, starting the ring vibrating unless it is damped adequately.

Fig. 6 gives some interesting curves of blowby. Curves *A*, *B*, and *C* are three different 3½ by ⅛-in. rings, made by three different manufacturers. Note how close the characteristics are. The average break occurs at about 3600 r.p.m.

Curve *D* is 3½ by 3/32 in. with the same radial thickness and the tension is in proportion to its width. The radial pressure is substantially the same. This ring is a very satisfactory ring from the blowby standpoint as can be seen by the data.

These curves might indicate that the natural frequency of the rings is important. It is a pat theory and one in which I once believed. However, today I view it with much suspicion since the natural frequency of piston-rings is somewhere between 10,000 and 12,000 cycles per min. I believe the fact that rings will vibrate is important, but the actual period is not important.

I have no intention of writing for posterity and assume the privilege of denying my own opinions from time to time as new evidence is brought forward and offer Fig. 7 as an interesting bit of evidence and as a warning to investigators to make certain that their blowby data are taken after rings are broken in properly. In this figure are shown three different blowby breaking points for the same rings, after varying degrees of run-in. Had I not been acquainted with this ring, I might have assumed that the first curve represented its true characteristics. After further running, two other readings were made and improvement followed up to stabilization.

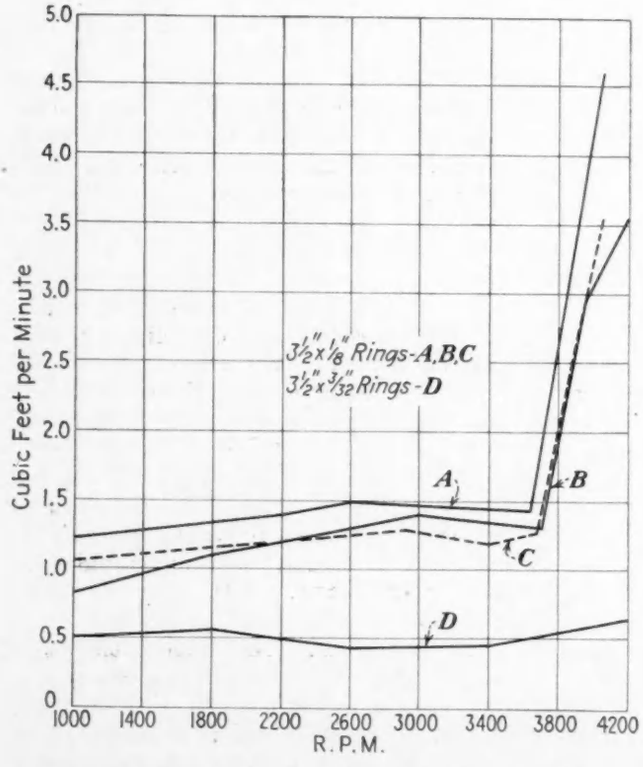


Fig. 6 – Blowby Curves for Four Different Piston-Rings

Obviously, if the natural frequency of the ring were controlling, this condition hardly could exist. It does indicate that the degree of bedding-in influences the speed at which the rings can vibrate freely. Any factor that frees the ring of the damping influence of contact with the cylinder walls at the ring points will start the rings vibrating. Thus the greatest factors in anti-flutter become high tension, high points, and quick and perfect bedding-in.

Factors that serve to make good rings out of cast iron should be applied to rings of other materials. Cast iron is used in the largest percentage of the world's engines. Individually-cast-to-form rings have been found to hold their characteristics over the longest period of miles, the important characteristic being a predetermined radial pressure pattern. The fact that a ring may be austenitic does not mean that it can take liberties with blowby. Any engineer or ring maker who overlooks or ignores this point will be rudely awakened.

Advances in material should be a forward step, not backward, and engine development can only move forward by adding to what already has been accomplished.

The British industry has been offered austenitic sleeves and rings as the cure for bore wear. However, satisfactory results are obtained in America without these very expensive austenitic products and, therefore, we are justified in insisting that 50,000-mile bores are available with cast-iron rings and sleeveless bores. This is the theme of our position on bore wear.

The foregoing at least gives you an idea of the state of discussion on this subject.

Part II

Our second major problem has been and is carburetion. This has been a particularly tough battle because so much that is associated with carburetion is in hands outside our own plant, and each change in practice has had to push through a maze of passive resistance.

Our target has been miles per gallon; our greatest obstacle has been the carburetor itself. To get good miles per gallon we have required a different breed of carburetor than was available.

Our campaign for better mileage is many years old, and much that is herein written has been said before. Nothing that we know or offer is of itself new, except perhaps the point of view. The problem is old, and the answers are likewise old. Experience, after all, is exposure to the work of others and in this way the author is indebted to many, but more particularly E. H. Shepard of the Chevrolet Motor Co. Whether I have been a good pupil, Shepard will know – but a pupil I have been.

The automotive industry has been for years at the mercy of the carburetor men and, if their reign of terror has been filled with bad mileage and flat spots, it is because too often we have demanded that the carburetor calibration cover up engine weakness or errors, that it could not do without sacrifice of carburetor performance in some way.

For twenty years, carburetor men have drilled and plugged holes to eliminate one type of bad spot and brought in two others. Each carburetor man has some pet bleed or gadget and, as a given carburetor has passed from one hand to another, it has become a mass of holes, corners, and fittings –

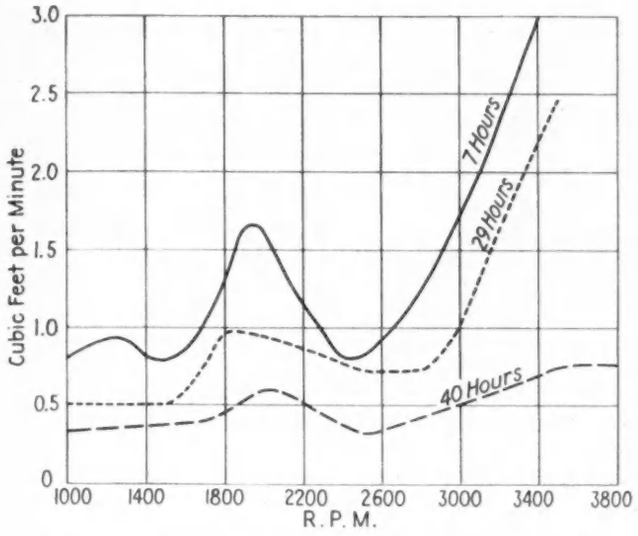


Fig. 7 – Comparative Blowby during Running-In Time

each sworn to as vital in function, but the ensemble defying the laws of nature. Our only hope today is to begin plugging holes and throwing bits away, and we are happy to report that things are getting better.

Our ability to burn lean mixtures is stymied until our ability to supply uniform lean mixtures is established definitely. Our ability to burn lean mixtures controls quite definitely our hope of better economy.

In the past our knowledge of handling lean mixtures was utilized to cover flat spots or lean spots, which may have existed. This procedure exhausted our opportunities.

Good carburetor men have known for years that ignition tuning was necessary for good carburetion. This tuning was a simple method of improving the ability to burn lean mixtures. Lean mixtures and low voltage at the spark-plugs in combination are about the worst possible condition for operation or economy. This sad situation applies particularly to full-throttle low-speed operation, although the part throttle is also dependent upon the voltage available. In the past missing, unless an obvious break was found in the electrical system, has been charged to the carburetor because enriching the mixture usually cured it.

We are beginning to look upon the carburetor for what it is – a good or bad metering device capable of creating its own variation in metering. It is quite unable to overcome variations from other sources.

Facing this fact we realize that any element – be it design or a condition that serves to interfere with uniformity of the mixture in the combustion-chamber, or the voltage to ignite this mixture – must be corrected by the engine man, not the carburetor man.

The engine designer can correct through design which may lower the cost, whereas the carburetor man, though his hand be quicker than our eye, just turns on the gas.

There are enough inherent problems in the carburetor in the way of disturbance to a uniformly correct mixture, to tax the ingenuity of the carburetor engineers.

To meet the requirements of the average motor car six-cylinder engine, a carburetor must meter, or should meter, a fairly constant mixture ratio for full throttle, varying the amount of fuel with the air consumed, a varying mixture ratio for part throttle or road loads varying with the ability of the engine to burn lean mixtures. It must meter an idling mixture ratio, that is relatively rich, capable of turning over a tight engine, and not too rich for the engine when it is loosened up. We must also have a rich mixture for accelerating at part throttle, a richer mixture for full-throttle acceleration and last, but certainly not least, a definitely calibrated starting mixture ratio that will represent the ignitible ratio for the coldest prevailing air temperature.

Good tank mileage depends upon the type of operation to which the powerplant is subjected. Obviously, except under extraordinary conditions, by far the maximum use will be at part throttle or road loads.

Road loads therefore represent the only type of operation worth attacking. With full operation we may succeed in reducing the pt. per b.hp-hr. down to 0.5 or slightly less. However, this operation seldom is reflected in tank mileage. In fact lean mixture at full throttle may be costly, not only due to lack of flexibility but to lack of durability.

To burn lean mixtures at part throttle is the objective – to burn these mixtures without the usual raggedness that is associated with lean mixtures. In 1934 in the United States, carburetor men maintained that mixture ratios already were too lean for the same reason that today in England many engines are operating and stumbling with leanness but do not give good mileage. Mixtures were getting lean without the necessary provisions for burning them. As events transpired in the United States in 1934, mixture ratios were not too lean and since that time we have operated with mixture ratios two ratios leaner. Full-throttle operation is different – here we get into trouble if we attempt lean mixtures.

The business of utilizing part-throttle lean mixtures for improving fuel consumption requires a study of the business of burning lean mixtures.

The limiting factor is not the mixture that leaves the carburetor, but the mixtures within the combustion-chamber. The exhaust gas present in the chamber displaces what might have been air. Were it not for the presence of this gas we probably could utilize ratios that today would be considered fantastic.

Exhaust gas poisons or reduces the burnability of the mixture, and a slight variation of ⅛ in. of hg. upward in back pressure will give discouragingly bad effects, even at low speeds and small throttle openings. We know from experience that valve timing makes major differences in the amount of exhaust gas trapped in the chamber. For years carburetor men have complained of excessive overlap of exhaust and intake valves. This overlap made idling uneven and difficult to handle. After all what does this condition mean except that the ability to burn lean mixtures was being challenged? The high vacuum in the intake inhales a tremendous amount of exhaust gas which, added to that which is trapped, gives an impossible mixture to burn. The old trick was to open up valve clearance and open up spark-plug gaps and surprisingly good slow running was obtained. It may have brought in noisy tappets and missing on acceleration, for which the carburetor man and his ever-ready drill was at our disposal whether we knew it or not.

In order to progress with fuel consumption, we must improve the burnability of the mixtures in the chamber, and we must improve our ability to burn lean mixtures. Under burnability we must consider the fuel. Its volatility and perhaps its ignitibility are important. Investigators have determined that ignitibility is a variable with temperature, and I am certain that no two fuels are exactly alike in this variable in a cold engine. I, therefore, believe that fuels will vary in ignitibility at the warmer temperature that exists for a given operating condition in an engine. With this point in mind, heat at part throttle is important and heat loss to the combustion walls should be minimized for the area around or near the spark-plug. Water temperatures are very important. Were it not for our inability to change the water temperature

quickly, we should like the water temperature to vary inversely with the throttle opening. We compromise on the hottest thermostatically controlled water that we can get away with, which is about 170 deg. fahr. Higher temperature than this would require a good deal less lag in the control than exists today. The hot-spot in the manifold likewise plays a dual role if designed properly. Not only does it aid in division of fuel but, in raising the temperature of the mixture, it improves the ignitibility. Improvement in the leanest mixture that will burn without missing continues with increase to part-throttle mixture temperature, long after no gain is found in distribution. To make further gains in ignitibility through temperature, we must have recourse to a thermostatically controlled and heated intake silencer. This device, however, must be hooked up only to heat at part throttle or it may breed detonation and volumetric loss.

A more important element in the improvement in burnability is the reduction of the dilution by exhaust gas. Here

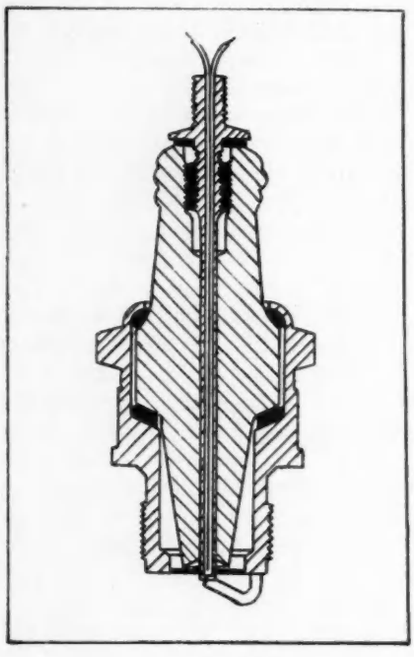


Fig. 8 – Spark Plug with Thermocouple in Central Electrode

we must reduce the amount trapped in every possible way. Shape of combustion-chamber, shape of exhaust ports and pipes, type of exhaust silencer, cam timing, and the profile of the cam must come in for consideration.

The shape of the combustion-chamber always has been credited with a good deal more than it deserves for improvement in combustion. It is true that it can be manipulated for anti-detonation and shock, but very little can be done with it to improve ignitibility except that we must be certain that it provides no catch basins for exhaust gas near the point of ignition and that it does not over-provide for heat conductivity.

Of course, the greater the anti-shock characteristics and anti-detonating characteristics, the greater the improvement in compression ratios. This factor not only improves the fuel consumption thermally, but also cuts down the amount of exhaust gas trapped by providing a smaller clearance volume and, as previously stated, a little exhaust gas does much damage.

Exhaust ports may influence the results if their shape is such as to promote eddies and if badly manifolded. Ports that are siamesed must be given particular consideration to be sure that the junction does not interfere with the flow from either of the joined ports.

The entire exhaust system from the exhaust pipe to the final tail pipe must be considered from the standpoint of easy flow and minimum back pressure. Quite often, when in the depths of silencing problems, we are tempted to reduce the pipe sizes and allow the back pressure to climb a "little." The argument is that it really has very little effect on road performance. This reasoning, of course, is correct. However, it may have a major effect on part-throttle tank mileage. This evil is so potent that it not only affects the leanest mixture that we can get away with, but it will lower the effectiveness of an existing mixture ratio. We have observed on small engines 4 miles per gal. difference between two acceptable exhaust silencers without any change in carburetor or ignition.

Timing is the most potential offender against burnability, and overlap of the intake and exhaust is the most poisonous. We must remember that all good cam shafts have quieting ramps wherein 0.001 in. may affect valve-opening time 8 deg. If normal, theoretical timing includes an overlap, the additional overlap due to ramp will, and does in many cases, give around 80 deg. actual overlap. Such conditions are difficult to deal with from the standpoint of burnability, and usually smooth, slow running and high tank mileage are out of the question. Automatic valve-clearance adjusters, allowing zero clearance for tappets and minimum overlap, always give a stable slow running and an improvement in economy. We predict that automatic tappets will become popular here and in England for carburetion reasons.

Before we finish with the burnability of the mixture in our combustion-chambers, we must return to fuel. In Europe, due to availability and government policy, we are forced to consider the use of blends which must include some homemade fluids. The ignitibility of these fuels must be studied. The volatility, of course, is important, and we believe that this varying volatility may influence ignitibility within the chamber. This factor requires varying degrees of temperature to make fuels alike from the standpoint of ignition. This point can be seen easily from the cold box when establishing ignitible mixture ratios at 0 deg. fahr.

We believe that a study of the particular germs affecting ignitibility under cold conditions would lead to the isolation of the germ or particle that, when concentrated, will give greater ignitibility under operating conditions and would lead to a fuel that might some day have a chance of living up to present billboard claims for improved fuel consumption for special fuels.

We must now consider our ability to burn lean mixtures. That high voltage is a necessary part of our ability to burn has been known for many years, and has been used for many years as a first aid for jobs hard to carburate. A good carburetor man always has been able to take one job and do comparative wonders with it. He "tunes" it by making certain that the battery is right up, all connections are tight or good, the high-tension cables are new and as short as possible, new spark-plugs with gaps opened up, a coil selected for its accidental high potential (some carburetor men carry a pet coil with them). The valves are set, not to give quiet operation, but with a vacuum gage to give the highest manifold vacuum under slow running conditions and, last but not least, he manages to cover up part of the radiator to keep things warm under the hood or bonnet. Yet for production they just turn on the gas, knowing that they can expect very little consideration from the engineer toward making these fuel-saving adjustments a regular part of production equipment. At least this condition prevailed up to 1936 in the United States, and the ice is just being broken in England.

To improve our "ability to burn" we must recognize that, in the combustion-chamber, we are shooting at a small target with a rifle and it must be a perfect shot, well directed or we

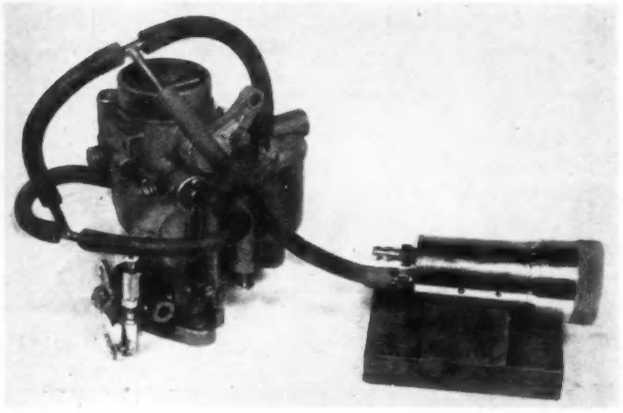


Fig. 9 – Sensitive Needle Valve with Micrometer Adjustment Attached by Rubber Tubing to the Float Bowl of the Carburetor

miss. We must exchange our rifle for a shotgun in order to be sure that we cannot miss. It is regrettable that our authorities on ignition have not given this phase of ignition more consideration. We are certain that their written works should not be so flat-footed about narrow gaps. The only justification for narrow gaps is inadequate coils and plugs. Just as soon as the coils are brought up in potential, wide spark gaps can and do operate well. This is known to many in America and, in England, it has been proved by the Lucas Co., and W. O. Kennington of Delco Remy & Hyatt, Ltd.

One may ask why wide gaps at the spark-plugs are desirable. It looks at first glance that the target we are shooting at becomes larger and easier to hit. This probably is so but, much more important than this reason, is the fact that wide gaps and a suitable coil give a multiple discharge, and thus the rifle becomes a shotgun which, together with the larger target, makes hitting relatively easy. This point is so easy to demonstrate that we fail to understand how the professional historians of the ignition industry could write as they have on spark-plug gaps. This is the danger of writing for posterity. No one can hope to have an opinion stand for long. Two or three years is a long time for a theory to stand in this industry.

Mixture distribution is, of course, a tremendous factor in our ability to burn lean mixtures. Our limiting factor is misfiring and, if this condition occurs in one cylinder only, it is just as much a limiting factor as if it occurred in all cylinders at one time, which it never does.

Unfortunately, full-throttle mixture distribution may limit our part-throttle activity. If the super coils are not available, lean mixtures do not hit well with wide gaps at full throttle pulling between 10 and 20 m.p.h. This missing may be, and usually is, due to poor distribution among cylinders, and to low voltage at the plugs. Perhaps low wattage is a better description.

A simple but stable method for measuring distribution is essential. There are many methods, some good, some bad. Judging from some of the results that we have checked in England, there must be a good number of poor methods or, perhaps, no method at all.

We have used the Rabezzana and Kalmar method as illustrated in *Automotive Industries,* Mar. 19, and 26, 1932. This system, which we refer to as the R.K. system, makes use of a simple fundamental of engine operation. Mixture distribution is determined by recording of spark-plug point temperatures.

Rabezzana and Kalmar of AC Spark Plug Co., had developed a special spark-plug for determining the operating temperature of the spark-plug, thus enabling them to recommend intelligently the proper plug for a given engine. Fig. 8 shows a section of this plug.

We have adapted our practice to the R.K. methods and, if one is satisfied to use this method for measuring distribution, very quick and stable results can be obtained.

Many engineers believe that the temperature rises with leanness. In actual operation this condition is so because few engines, if any, operate at the maximum economy point which is at the peak of the temperature. Thus, when we lean out the mixtures, the combustion products apparently get hotter. Actually, when working with each cylinder separately, we find that, past the maximum economy point, the temperature will drop. This method gives us a stable mechanism for measuring small variations in distribution down to 2 per cent. If this percentage seems large, let us present the fact that most jobs that we have worked with showed errors, at the start, of over 35 per cent, between front and rear cylinders.

It is, of course, necessary to have an adjustable carburetor in order to be able to check a range of mixtures from lean to rich or vice versa. This requirement in itself, at one time, represented no mean problem since small variations for increments of fuel are necessary, and it is impossible to do this requirement by mechanical means without affecting the distribution of the mixture out of the carburetor.

Adjustment of the fuel must be made possible by changing the level in the jets which will not affect the qualitative flow, but will change the quantity.

Several years ago the Chevrolet Engineering Laboratories developed a sensitive needle valve with a micrometer adjustment as shown in Fig. 9. This valve was piped by a rubber tubing to the float bowl of the carburetor. The float bowl is sealed properly and a vacuum is induced in the float chamber, either from an outside source or, preferably, from the carburetor itself. The needle valve is used to bleed air into the float bowl, partially relieving the vacuum. This action gives not only an effective adjustment, but one of micrometer order.

A valve of this type is indispensable in an engine laboratory, and is in constant use at Chevrolet and at Vauxhall. Running engines with a fixed carburetor is done only when tests are to be made of an engine as installed in a car. Other than this, the variable-adjustment carburetor hook-up is used so that every test represents the engine potential less carburetor.

With the variable adjustment for the carburetor and the R.K. system for measuring distribution, we can proceed to dig out a great deal about distribution.

Organizing the data is important because, the simpler they are, the easier it is to dig through a great deal of work and arrive at causes and cures. Fig. 10 shows a work sheet of a distribution check. We ignore the actual temperature read-

1000 R.P.M.- FULL THROTTLE-CARBURETOR INVESTIGATION SERIES X

	Cylinder No.						Beam Load	Fuel Time	% Fuel
	1	2	3	4	5	6			
Miss Point	462	446	522	330	300	328	51.0	0.200	80.0
	469	441	512	331	300	336	51.5	0.195	82.0
	477	430	480	326	340	350	53.0	0.190	84.0
	478	424	466	321	366	359	53.5	0.180	88.7
	479			320	498	372	57.0	0.175	91.4
	477				507	382	57.9	0.170	94.8
	465				512	391	58.0	0.165	96.8
	466				532	396	58.5	0.160	100.0
					533	410	58.4	0.154	103.0
					522	412	58.1	0.150	106.5
					510	410	58.0	0.150	106.5

Potentiometer Readings

Fig. 10 – Sample Log Sheet for Mixture Distribution Based on the R. K. System

ings and pay attention only to the relative temperature peaks, and the fuel flow at each peak. The difference in fuel flow for the relative peaks indicates the difference in mixture ratio at which each cylinder arrives at its maximum economy. Each peak can be assumed to be the maximum economy point for each cylinder, and the percentage in fuel-flow difference is the percentage difference in the fuel flowing to each cylinder.

A mark at each peak under each cylinder quickly points out the extent and the location of the variation. Later these simple curves can be used for comparative purposes.

Our recent investigation covering the Vauxhall 14-hp. engine brought us face to face with the fact that no serious investigations could have been carried on to determine the quality of distribution at the carburetor proper. We found that most carburetors that we checked were in a rather bad way for distribution, the front cylinders showing rich, the best being 30 per cent richer in the front than in the rear. There was no question of whether the carburetor was at fault because turning the carburetor 180 deg. reversed the rich and lean cylinders. The error was fairly consistent throughout the range, but applied mostly to full-throttle operation. This type of error is due to wet fuel, and the dryer mixture or part throttle was thus not as much as affected. Discovery of this condition here was most discouraging because, in the United States between 1934 and 1935, we had cooperated with E. H. Shepard of Chevrolet in correction of this fault. We had learned that major variations would come as a result of impossibly small differences in the "innards" of the carburetor.

English carburetors ran true to form where the carburetor had not been given consideration for distribution. To illustrate the nature of these variations we submit the following:

The position of the fuel nozzle in the carburetor was found to be extremely sensitive in its relative position to the center of the carburetor. Each size of venturi or choke required a different position of the nozzle end. This carburetor giblet must be held to position to $\pm 1/64$ in. Bars and special points near the nozzle, originally installed according to the carburetor engineers to provide good distribution, were found to be responsible for bad distribution. This condition indicates that very little fundamental work had been carried on heretofore. However, this arrangement did permit us to make improvements by eliminating parts, which is always a happy event. We had considerable difficulty in duplicating our good results with various carburetors of the same model. Investigation showed that the venturi in the original carburetor was 0.008 in. eccentric, whereas others were concentric and still others eccentric 0.008 in. in the opposite direction. The difference between these venturi in their effect on distribution is shown in Fig. 11.

This figure gives some idea of the control necessary in loose-fitting parts in the carburetor. When carburetors are built up in sections with stand pipes and other gadgets innocently slipped into position, you may expect trouble. We checked one carburetor that was so bad that the same cylinder would have two maximum economy peaks. As the amount of fuel flowing from jets increased or decreased, the wet fuel changed its direction and went to other cylinders.

How many such carburetors are in use we do not know, but we do know that similar constructions are in use.

The progress that can be made by such studies can be indicated best by studying different carburetors calibrated to meter the leanest mixture under the varying conditions.

In the flow box we have a steady flow which is the direct reflection of the ability of the carburetor to pass so much air and fuel and the slightest interference with this flow will be indicated.

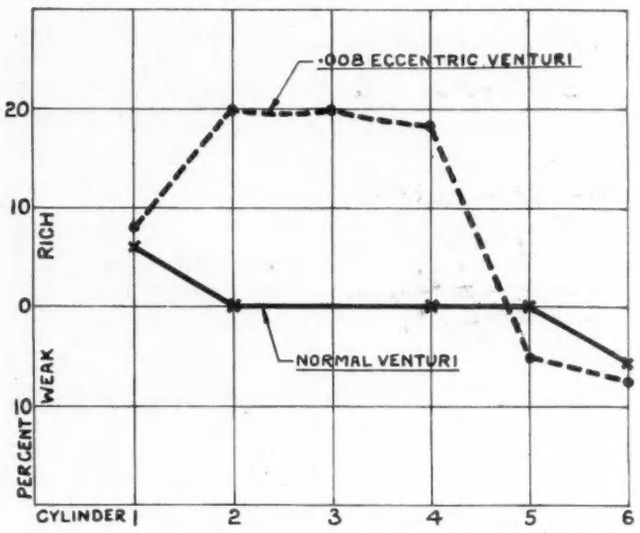


Fig. 11 – Mixture-Distribution Variation due to Venturi Eccentricity Using R. K. System

Fuel flow and air flow through the carburetor on the engine are influenced by pulsations which, when present, cause the fuel particularly to surge out of the jets. This condition varies with speed and the number of cylinders operating from one jet or nozzle.

In order to do a complete job of calibrating a carburetor to an engine, assuming that we have the equipment at hand, we should be certain that the necessary development work to obtain the maximum ability to burn the mixture and the easiest mixture to burn has been completed. We should know by this work the spark advance, the exact voltage, spark-plug gap, spark-plug position, and spark-plug grade. We should settle the valve timing, manifold design, mixture temperature, size of carburetor components with exception of jets and, last but not least, the exhaust and intake silencer, at least to approximate what will be used on the car. One cannot go on leaving this work unfinished and expect the carburetor engineer to fill in the gaps.

Remember that we have seen as much as 4 miles per gal. difference in fuel consumption through the part-throttle range, even at such low speeds as 20 m.p.h., between two exhaust mufflers of different design. Even mufflers of the same design will give variations in fuel consumption.

Full-throttle calibrations have their own peculiar difficulties due to the limitations of the present-day carburetor. Data must be obtained on the dynamometer for the full-throttle range and consist of the following: all speeds from 400 to 4000 r.p.m. in increments of at least 400 r.p.m.; ft-lb. torque or b.m.e.p.; lb. of fuel per hr.; lb. of fuel per b.hp-hr. and manifold depressions. These data must be obtained with a variable-mixture ratio carburetor in order that the results may represent the exact engine demand.

We must now obtain a fixed-jet carburetor to do as well at each speed. Here is where the carburetor man can use his sleight of hand, drills, and toothpicks. The important thing is to get one carburetor, no matter how freakish the drilling, but the "innards" of the carburetor must remain unchanged.

These carburetor gentry must be watched keenly because, just as soon as they change venturi or fuel nozzles, or do anything "unimportant" that affects the contour of the inner flow lines of the carburetor, they will have changed the distribution characteristic, and this work will have to be done over again. It is perhaps well since measurement of distribution can be so simple that, whenever a carburetor has arrived at a given state of development, it should be checked for this

quality. Even production carburetor samples should be checked. The engineer will then get some idea how much revision and development work lie ahead in carburetor design to arrive at respectable uniformity. This factor has been one of the most discouraging conditions of English carburetion. When we find 30 per cent variation in distribution with pieces made from the same dies and requiring microscopic investigation to locate, then you may understand what happens in carburetors with moving weights and venturi – carburetors that never have the same shape or flow lines for two instants. Carburetors made with fuel nozzles that theoretically stand straight are supposed to rain fuel concentrically. What discouraging messes they turn out to be when they are investigated thoroughly! I am sure that, if equipment is made available by the engine designer, carburetor engineers would modify their constructions.

Getting back to carburetor calibration, as stated previously, a fixed-jet carburetor is required that meets the engine requirements. When completed, this carburetor is put in the flow box, and records are made allowing a reasonable amount for production variation; thus we establish high and low

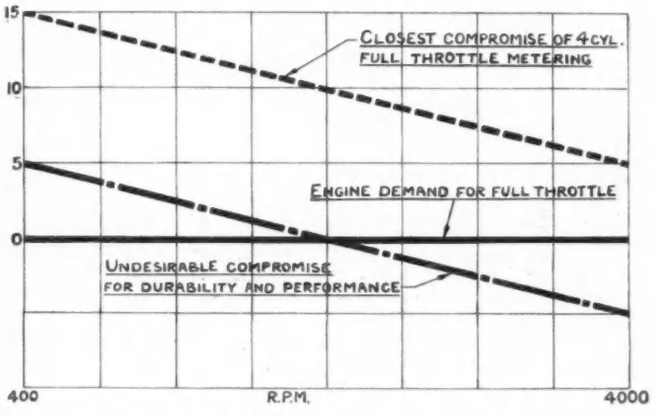


Fig. 12 – Fuel Requirements for Engine Demand

limits for the carburetor. From this point we can carry on in the flow box where jet changing is very easy and a practical combination can be worked out. If all conditions are straightforward, you have a carburetor near completion in a very short time. However, the work may not be straightforward, for instance, assuming that the fuel flow for the wide-open range engine demand is completed, we represent this flow with a horizontal straight line in Fig. 12.

We plot relative to this line the fuel flow of the fixed-jet carburetor, and then the trouble begins. The slow-speed range is too rich. It is desirable to arrange for at least 5 per cent rich for the high-speed full-throttle conditions. However, due to normal hydrostatics of the carburetor, we must take with it a far-too-rich low end, particularly for four-cylinder engines.

The rich high speed is necessary for engine durability and cooling. The rich low end, however, is responsible for the manifold loading under pulling conditions. Eliminate this condition, and it is surprising how stable a very ordinary manifold will become.

This condition is not bad in six-cylinder engines because the average carburetor can be calibrated to give fairly close metering at the low end within 10 per cent, and this percentage, rich for the low end, does not give too much trouble if the manifold is not altogether bad.

In the four-cylinder engine, however, the conditions are different. Here, due to pulsation, the low-speed full-throttle is normally very rich. This characteristic applies to almost any of the carburetors used in production. There may be some specials somewhere but, by and large, all carburetors used on four-cylinder engines in Europe are, to date, incapable of metering the required fuel. Let there be no misunderstanding about this point. We are not criticizing English carburetors, but pointing out the weaknesses of the present practice with a definite objective. We refuse to take a passive attitude toward error unless there is no way out of it. We have objected to this condition and have pointed out to our carburetor supplier that there is available to the carburetor engineer two factors that can be utilized to overcome this difficulty. One is the fact that there is a depression that varies with speed that can be put to work and there is a velocity variation that may be utilized. Work is in hand, and we know that this fundamental error can be overcome.

Today, with the knowledge that the error exists and that it is of a large order – the mixture being 30 per cent richer at low-speed full throttle than the engine demand – by use of the flow box we can make some compensation. To illustrate this condition we show Fig. 13, a flow-box chart of mixture ratios for full throttle of some typical English carburetors. Curve *A* shows a modified 10-hp. calibration, wherein the low-end mixture ratio is 15 per cent richer than needed. These curves being taken in the flow box are minus pulsation, therefore although Curve *A* shows leaner at the low end, it is actually 10 per cent richer than the high-speed end when on the engine. Curves *B* and *C* are taken from representative production 10-hp. cars. You note that Curve *A* slopes in the right direction slightly and Curve *B*, which is parallel, must be very rich on the low end. Studying Curve *C* we are at a loss for words. Here is an almost unbelievable calibration, in fact, it is not a calibration. I doubt very much if a real carburetor man ever had anything to do with it.

Getting back to Curve *A* we do get some idea from this curve, coupled with our knowledge of how far it is off the line, as to the effect of pulsation on the carburetor calibration. In the United States, where the four-cylinder engine is almost extinct, this problem is not so troublesome, at least carburetor men do not worry about it much. I am not certain that they are as pure as they think they are. There are many eight-cylinder engines in use in the States using dual carburetion which, like the four-cylinder engine, resolves itself into four-cylinder carburetion in duplicate. I firmly believe that a lot of the carburetor difficulties associated with "eights" will be found chargeable to this inability to meter to engine demand due to pulsation.

Reverting to the study of overall calibration, we now find

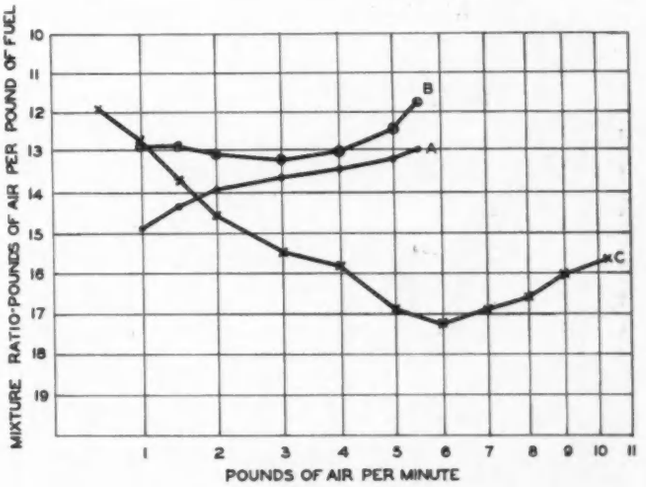


Fig. 13 – Comparison of Three Full-Throttle Flow-Box Curves

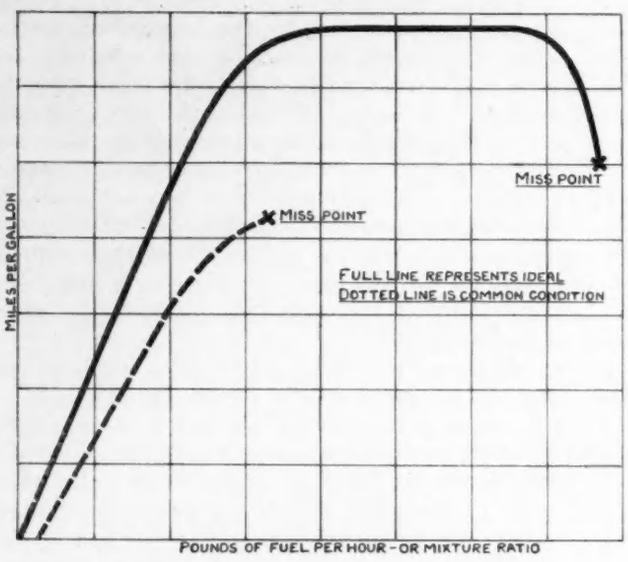


Fig. 14 – Constant-Speed and Constant-Load Part-Throttle Mixture Curve

ourselves at the part-throttle consideration. Here, if at all, our plans for miles per gallon must be laid.

We will have before us, if we have completed our work, a potential curve representing the miles per gallon through the part-throttle range. This curve is made up from the peaks of smaller part-throttle loops. The business of making part-throttle loops is necessary in the study of the effect of the various factors on economy. Part-throttle loops can be made plotting miles per gallon against pounds of fuel per hour or miles per gallon against manifold depression. The beam load is kept constant, and the throttle is opened or closed to maintain speed and load. This procedure permits determination of the maximum miles per gallon, without missing for each speed. If our work is carried through properly, our miles per gallon for each speed will level out and decrease before missing occurs.

Fig. 14 gives an illustration of two types of part-throttle curves. Unless a specific study for direct comparison or specific development is under consideration, the actual charting of loops is unnecessary. Economizers – devices for leaning out part throttle – are very old, and we have been puzzled by their absence in England. We mentioned this fact to a leading manufacturer in England two years ago, asking why vital parts were left out of English carburetors. He blamed the carburetor companies for not bringing these improvements to their notice. The carburetor companies answered the question by saying that the motor companies would not pay for the pieces.

The strange truth of the matter is that there are more than enough useless pieces in the carburetor to pay for economizers and other necessary gadgets and leave a margin of cost to spare. Fig. 15 shows the effect of economizers applied properly. Curve *A* shows the calibration that we started with, which is typical of 90 per cent of English cars up to the middle of 1937. Here we find, for full throttle and part throttle, approximately the same mixture ratio. The full throttle was very lean, and the job was obviously not a flexible one and very slow in the warm up.

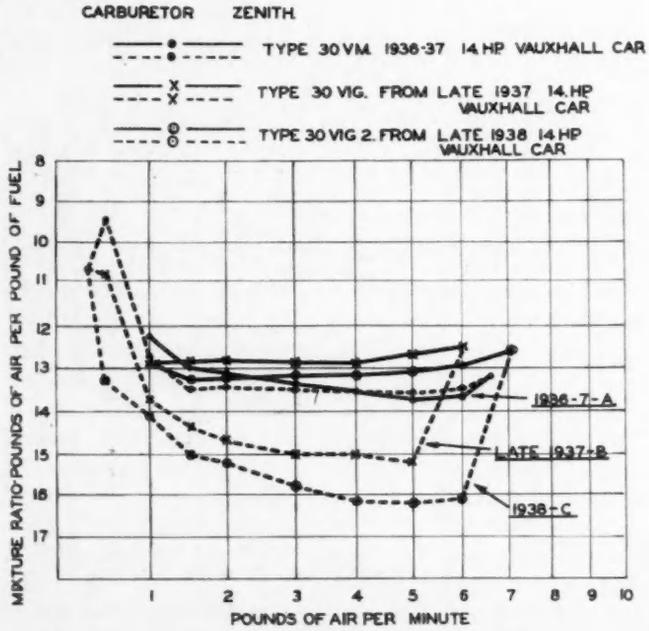


Fig. 15 – Effect of Economizers Applied Properly

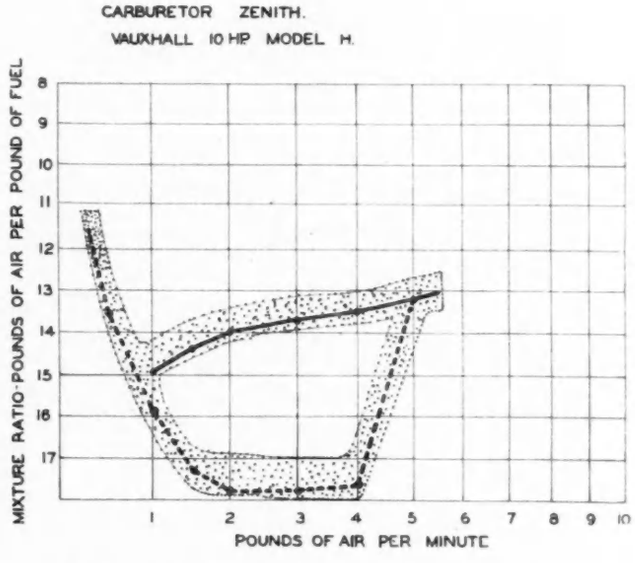


Fig. 16 – Flow-Box Chart for Small "Four"

In the new calibration, Curve *B*, because the economy range was separated from the performance range, it was possible to enrich the full throttle from one half to one full ratio. The part throttle was leaned out from one to one and a half ratios. The net gain was gratifying because the job became very flexible and a pleasure to operate, and the tank mileage went up 4 miles per gal. at the same time. This improvement was accomplished by improving the ignition and calibrating a carburetor with independent part-throttle and full-throttle ranges. Because much that we worked with could not be changed, it was necessary to take a second bite at this cherry. With corrected manifolds and corrected carburetor distribution, we picked up 3 additional miles per gal. and cut the warm-up time down to a fraction of what it was with the old manifold.

Fig. 15 shows the flow-box chart of the second revision. With this calibration we have increased flexibility and have it quickly as compared with the original. We also have gained approximately 7 miles per gal. The full throttle is leaner than the first correction because full-throttle distribution is better, due to manifolding and carburetor correction. Part throttle is leaner, due to manifolding and correct application of heat.

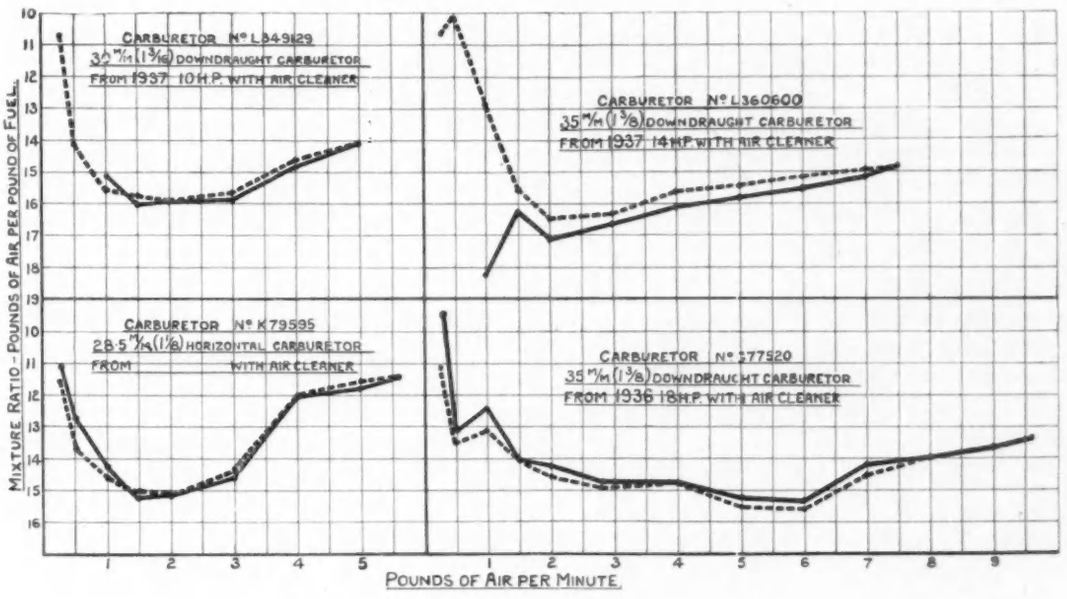


Fig. 17 – Flow-Box Charts of Four English Carburetors

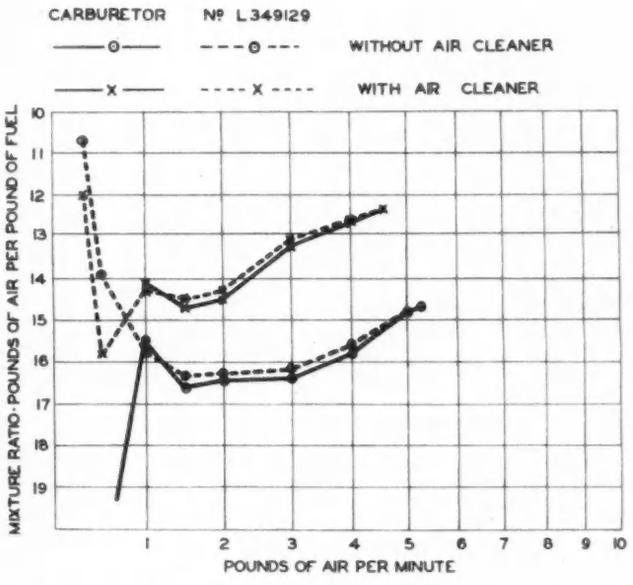


Fig. 18 – Comparison of Flow Curves with and without Air Cleaner and Silencer

Fig. 16 shows perhaps the leanest mixture ratio in use anywhere with good flexibility and quick warm up. Here is the result of doing everything for a small "four" that has been done in the past for large sixes. When this job was started, we were told that it was useless to do anything for small "fours" because they would not respond to treatment. There were two items in particular that we were warned against as being useless – one was the part-throttle economizer, and the other, vacuum over spark advance for part throttle. We were given flow curves for English carburetors to prove the point – we shall see these later. This flow chart of the 10-hp. Vauxhall proves that any engine, including a small "four," responds to sound treatment. We find 3½ to 4½ ratios between full and part throttle. Incidentally, the over spark advance for part throttle is 20 deg. more than for full throttle.

Fig. 17 shows flow-box charts of four English carburetors. Note that no consideration has been given to road economy. We must again warn against the thought that we are being super critical. We are still driving forward in the quest of better general carburetion in England – it can be better than it is, and it will be, provided sufficient cooperative attention is focused on the problem.

The chart of an English carburetor produced in large quantities is shown in the upper left-hand corner of Fig. 17. Not only are part throttle and full throttle alike, but the flow is of such a nature that we refused to accept it because it seemed to be off the map. However, repeat checks verified it much to our bewilderment.

Fig. 18 is an illustration of another type of error that must be avoided. This figure shows flow curves of a typical result with and without air cleaner and silencer. Here we find the influence of the silencer entirely out of bounds, as much as three ratios, in some places. With such a condition the air silencer becomes too important a part of calibration. Each silencer would, of necessity, have to be checked for resistance. If for any reason the owner had to operate without the silencer, we would be in trouble. Surely it is obvious that a silencer of the type used in motor cars is not fit to be a serious factor in fuel metering. Fig. 19 shows a reasonable combination of less than half a ratio with and without

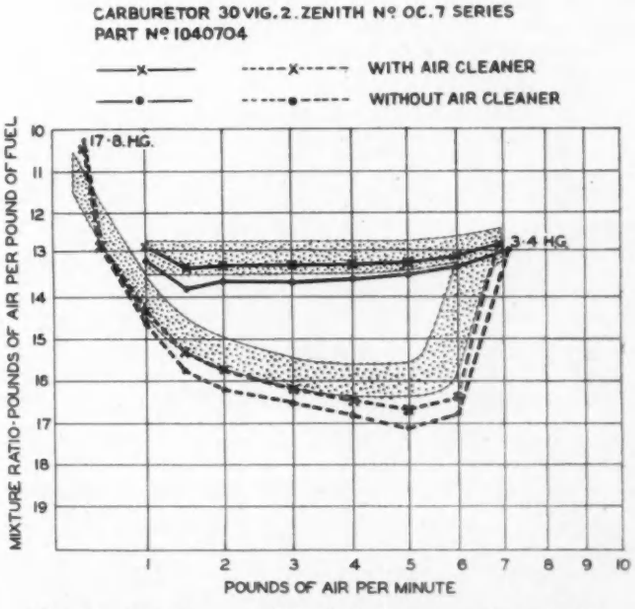


Fig. 19 – Flow Curves of Stock Carburetor with and without Air Cleaner and Silencer

silencer. In the United States this difference has been reduced to zero by balancing the pressure in the air horn and float chamber.

In order to obtain good flexibility with lean part throttle and to make possible the use of wide-gap spark-plugs for part-throttle economy, an accelerator pump is incorporated in all Vauxhall models. This pump gives the necessary correction to the mixture ratio to compensate for the fuel that lags behind. Thus, every time the velocity of the mixture is increased, there is available a small extra charge of fuel that is forced into the stream.

The accelerator pump has its friends and others. These others maintain that the English driver "jiggles" his accelerator a great deal and thus would promote trouble by dumping fuel into the manifold. The English driver jiggles his accelerator to keep his engine from dying. In other words, a bad carburetor has led to bad habits. Should this condition mean that we should not correct this evil? If you tell your English driver – who, incidentally, is the smartest average driver in the world – that jiggling his accelerator is no longer necessary, he will respond – in fact we are glad to add that he has responded.

If I have appeared to take advantage of the carburetor men with whom I have worked these past twenty years, it is only because among them are many understanding friends. If I have appeared cold and critical toward British carburetion, it is only because the heat of battle still flares. Once satisfactory carburetion becomes established in England, we will take you boys on for a scrap for miles per gallon, and give you a run for your money unless you do better than you are now doing.

Discussion

Warns Against Excessive Cylinder Temperatures

– Max M. Roensch
Chrysler Corp.

WE were somewhat puzzled two years ago when two prominent engineers from the Lucas Co. called upon us to discuss their cold-starting problems. When I asked them what the temperatures were that they encountered, they replied: "It sometimes got down to freezing." Little did I realize that they were trying to start and run engines with the equivalent of an S.A.E. 60 oil.

The use of these heavy oils, coupled with the limited supply, also explains our failure to confirm the excessive cylinder wear during cold-starting operation reported by English engineers.

I would be more inclined to believe that the difference in wear between Cylinders 1 and 6 shown in Fig. 2 of Mr. Taub's paper, is due to overcooling of the front cylinders. This belief could be checked by placing thermocouples in the cylinder walls so as to take the temperature at the operating surface. Although I heartily agree with the writer that the cylinder-wall temperatures must be kept up for light-load operation, it must not be allowed to run too high for full-load operation or the oil film will break down and abrasion or scuffing of the rings will result, followed by rapid cylinder and ring wear until the ring reseats itself.

A lot of damage to rings and bores occurs during the breaking-in period and also during periods of high speeds and loads. This damage can be reduced greatly by the use of coated rings. There are several kinds on the market, all of which seem to go a long way toward reducing these difficulties. The best known coatings are Ferrox, Granoseal and Altinizing.

The carburetor development in general in current practice in this country: controlled mixture ratios on part throttle, wide open – starting and warm up, are established fairly well. There is considerable room for improvement in this country on distribution as influenced by the carburetor. It still remains as the greatest single disturbing factor.

I am at a loss to know whether or not to praise Mr. Taub for his wonderful increase in economy obtained on the Vauxhall for, either he has done a marvelous job to gain 7 miles per gal., or the car was very poor to begin with. Unfortunately we are given only relative figures and not actual miles per gallon at a given operating condition.

Sees Further Efficiency Gains with Leaner Mixtures

– R. N. Janeway
Chrysler Corp.

IN my opinion, Mr. Taub's insistence on the possibilities of burning leaner mixtures by improved ignition is most significant in pointing the way to unexplored heights of part-load efficiency.

If we can achieve complete combustion of mixtures leaner than commonly accepted ratios, there are several factors which will produce cumulative gains in efficiency. Considering the thermodynamics of the actual cycle and avoiding the hypothetical and misleading theoretical air cycle, we find, first, that the limiting efficiency increases as the temperature range of the cycle is reduced as a result of the lower mean specific heat of the gases. This factor alone can readily yield a 10 per cent gain. Second, the heat loss to the jackets will be reduced by the lower gas temperatures, provided combustion is completed in the normal time period. Lastly, the leaner mixture will be accompanied by reduced manifold vacuum for a given part-throttle power output, which reduction will tend to reduce pumping losses. This important factor is illustrated clearly by considering that output can be governed by fuel quantity alone, without air throttling, down to the leanest mixture that can be burned.

The operation of these several factors, each contributing an independent gain, can conceivably combine to produce an economy never heretofore visualized as within the capabilities of the gasoline engine.

Overlap in Valve Timing Defended

– Stanwood W. Sparrow
The Studebaker Corp.

FOR the most part, Mr. Taub's conclusions are in accord with the results of our own experiments. Hence, these comments may sound more like the chant of a "yes man" than like a forthright discussion. In the first place we are in complete agreement with him as to the desirability of an ample supply of lubricant on the cylinder walls with suitable oil mileage secured by the use of effective oil control rings. "Throw it on and scrape it off" is a slogan that gives good results in cylinder-bore lubrication.

The influence of mixture ratio on wear has not been encountered – or perhaps not recognized – in our tests. Perhaps this condition is due to the fact that we have dealt with bores that were fairly well lubricated. We have, however, seen ample evidence of excessive wear as the result of employing a lubricant of insufficient viscosity and it is not difficult to understand why the thinning of the oil film by a rich mixture may be detrimental.

When we have found differences in the wear of individual cylinders of the magnitude shown in Fig. 2, the cause has been hot-spots rather than low temperatures or rich mixtures. In the absence of evidence to the contrary, we would be inclined to suspect hot-spots in the front cylinders of this engine despite the fact that the water pump is located at the front of the cylinder-block. For this reason, namely, because of the difficulty of getting uniformity in cylinder-wall temperatures, we do not feel it is wise to strive for extremely high jacket water temperatures.

As regards blowby, we believe that ring flutter is a forced vibration and we agree with Mr. Taub that the natural period of vibration of the ring is not of particular importance. Frequently the engine speed at which the break in the blowby curve appears increases with a reduction in the inertia of the piston. This condition would hardly be true if the natural frequency of the rings were an important factor in blowby.

Occasionally we have tested rings which improved in the control of blowby in much the same fashion as is shown in Fig. 7. It is our guess that this improvement is contingent upon better seating of the rings on the cylinder walls which reduces the opportunity for gas or oil to exert pressure on the face of the ring and help to collapse it.

As for carburetion, there is no question as to the desirability of holding to a minimum the dilution of the fresh charge by exhaust gases. Nevertheless, we feel moved to say a word in defense of the overlap in valve timing as more than once we have had such overlap to thank for providing much needed power. To be sure we can remember one engine which gave a lot of trouble in idling, and this engine did have a camshaft with a generous overlap. The carburetor men never permitted us

to forget this overlap until, in desperation, we gave them a camshaft with the valve timing their hearts desired. With this camshaft the idle was worse than before, and ere long a few magic changes were made in the carburetor whereupon the engine idled smoothly with the original camshaft, overlap and all. This example does not mean that overlap is unimportant, but it does mean that the factor of prime importance is the amount of exhaust gas in the combustion-chamber and not the appearance of the valve-timing diagram. The intake valve is the door that controls the opening through which fresh charge enters the cylinder; the exhaust valve is the door that controls the opening through which the burned gases leave the cylinder. The fact that both doors happen to be open a crack at the same time may be much less harmful than to close the exhaust valve too soon and thus fail to provide an ample opening through which the exhaust gases can escape as the piston nears top-center.

The emphasis which Mr. Taub places on carburetor details, such as the eccentricity of the venturi, is justified. Many of us once believed that, if the fuel left the jet in a fine fog, all would be well. We now know that, when the gases move through the manifold at low velocity, the fine fuel spray tends to cling together in much the same way that a wisp of cigarette smoke will float across a room. If satisfactory results are to be attained, the spray must be aimed, and the aim must be good.

Reviews Requirements of Piston-Rings

– **Macy O. Teetor**
The Perfect Circle Co.

A PISTON-RING'S importance seems to increase constantly. Piston-rings really affect almost every phase of engine operation. The difference of opinion between English and American piston-ring manufacturers is mostly due to the difference in concept of the problems involved. Unquestionably there is a difference in the importance of the problems involved, but this cannot be the only reason for the difference in opinion because American piston-ring manufacturers differ considerably among themselves.

A piston-ring has only a few properties but some or all of them are difficult to attain. In addition to having suitable design and dimensional accuracy, a piston-ring must have correct shape and tension. It must be made of a material that will retain these qualities for a satisfactory operating period without wearing the cylinder excessively. Suitable, correct, satisfactory, and excessively, are the words that cause the arguments.

The process of manufacture is only a means to an end. A manufacturing process remains satisfactory as long as it does not offer obstacles that may prevent the attaining of the desired standard of perfection.

High tension is desirable but not at the expense of correct shape and satisfactory wear characteristics. Correct shape is extremely important. Of course, what is known about shape and its relation to performance has been acquired by experience. One experience dictates one shape and another experience, another shape. A meeting of minds on a universal shape is, therefore, very difficult to achieve.

The selection of a suitable material for piston-rings has received considerable attention. Specifications for an ideal material are easy to write but producing a ring from such a material is a different story. The job is a little like working a cross-word puzzle. One necessary property is always interfering with the securing of a desired characteristic and, if an ideal material is secured, the rings must still be produced by some process of manufacture which presents a new set of problems.

Our thoughts on piston ring and cylinder wear were given in our paper, "The Reduction of Piston Ring and Cylinder Wear,"[a] at the Annual Meeting of the Society in Detroit, January, 1938. We are glad to see more attention given to cylinder temperature. We would change the statement "high wall temperature, uniform throughout" to uniform temperature, not too high or too low. Our thoughts on this subject were given in a paper, "Cylinder Temperature,"[b] presented at the 1936 Semi-Annual Meeting of the Society.

We do agree that a poorly shaped piston ring will work better pinned than free, but one problem is evident. The lower face of a pinned ring and the lower face of the ring groove are inclined to conform to their irregularities produced by the movement of the ring in the groove. The resulting ring-groove surface makes a poor seat for a replacement ring.

Mr. Taub has certainly made an important point in stating that he is not desirous of writing for posterity. If more of us felt that way, more interesting papers could be presented for thought and consideration. If we all write things that we know to be facts and wait until we are sure they are facts, very little will be written and much information that could be available will not be utilized.

[a] See S.A.E. TRANSACTIONS, April, 1938, pp. 137-140: "The Reduction of Piston-Ring and Cylinder Wear," by Macy O. Teetor.

[b] See S.A.E. TRANSACTIONS, Vol. 31, August, 1936, pp. 328-332: "Cylinder Temperature," by Macy O. Teetor.

Contributes Pertinent Experiences

– **F. F. Kishline**
Graham-Paige Motors, Inc.

I AM sure that most of us will agree with Mr. Taub's statements that much credit is due to the American piston-ring makers who have never been satisfied and are carrying on their development continuously, in many cases urged by one or more of the engineering fraternity who, at the moment, is in a jam.

One of these jams occurred to me about two years ago, wherein ring-scuffing and poor oil control broke out in an engine which had theretofore been reasonably satisfactory. After quite a period of experimenting, fussing, and grasping, a few of the contributing causes were learned. One of the most important was scuffing of piston-ring faces in new engines, particularly when started after becoming cold. In aggravated cases, this scuffing appeared on both pistons and rings, and we wasted much time trying to determine at which point it started. Eventually we learned that it could start either on the piston or on the rings and, in these aggravated cases, at both places simultaneously.

First, the matter of lubrication was gone into by the simple method of cutting open the end of an engine and looking at it through a 'scope first while cranking and then while running in the cold room. Much to our surprise, it was learned that oil, which according to the pictures in the catalog was supposed to squirt up on the cylinder wall, was doing no such thing for quite a period of time. It merely oozed out of the end of the hole. In the meantime, gasoline was being supplied in liquid form to the top of the piston by choking and, where more than 10 sec. was consumed in cranking, the rings were washed clean of oil so that, when the engine started and speeded, scuffing should have been expected. A lubrication change was devised which carried oil directly to the piston through the cylinder walls under pressure from the gallery line, so that, immediately when pressure was present in this line, oil started to flow to the piston surfaces. This change helped tremendously but, under the rather strenuous conditions of testing used, it did not cure. Next, the pistons were tinned on the surface which, in combination with the oiling, stopped scuffing on them.

Then it was found that the compression rings were still scuffing independently and, after trying nearly everything as a coating from cadmium to paint, we arrived at nickel plating as being the best material to prevent this condition on the ring surface. In the meantime, of course, some other deficiencies including bore distortion, improper clearances, and so on, were improved with a net result that proved to be very satisfactory in production. Thus, I wish to point out that, although we may assume that we have a copious oil supply, it may be similar to copious rains in the desert and not be present at all when really needed the most. I might add that the use of the foregoing improvements in production for nearly two years has almost entirely eliminated complaints from the field of high oil consumption and piston slapping formerly experienced.

Recently the nickel plating has been replaced by Ferrox developed by the ring manufacturers and is found to be just as effective and a great deal more lasting than was the nickel.

The other difficulty experienced with rings was that of scuffing at high speeds and seems to be caused by conditions entirely unrelated to those described heretofore. In engines running well over 100 lb. per sq. in. mean effective pressure at speeds up to 4750 r.p.m., it was, and to some extent still is, difficult to prevent compression-ring damage. This latter condition has been improved by a selection of ring type, location with reference to the top of the piston, and by better cooling of the hot area of the top end of the cylinder bore. I shall not go into detail, but return to the point of discussion, which is this:

I believe that it is commonly assumed that ring wear occurs at a somewhat constant rate, whereas these experiences have convinced me that most of what we call ring wear occurs in extremely short periods of time under conditions too little recognized and understood and which, in my opinion, have received very scanty attention. For instance, not more than two years ago, I have seen a new engine completely worn out in 10 or 12 hr. by simply letting it idle at a low speed during that time; yet this engine certainly had a copious oil supply at any higher operating speeds.

I certainly wish to go along with Mr. Taub in his treatment of carburetion and economy, particularly where he points out that lean mixtures do not necessarily cause heating as commonly supposed. There have been some very interesting experimental results obtained recently by using extremely lean mixtures in combination with compression ratios as high as 10:1 or 12:1, which have shown specific fuel consumption as low as 0.32 lb. per hp-hr. Does that not indicate the possibility of maintaining a desirable fuel density in the compressed charge which will be ignitible readily and become inflamed quickly throughout before detonation occurs? This is merely a new thought and may be limited entirely to combustion members of very small dimensions.

1937 Road Knock Tests

Report from Cooperative Fuel Research Committee

Presented by T. A. Boyd

THIS paper deals with the road-test portion of the extensive efforts made during 1937 by the Cooperative Fuel Research Committee to get as precise a correlation as possible between the laboratory knock ratings of automobile fuels and their corresponding ratings in cars on the road. It is anticipated that the comprehensive results of car tests reported here, taken together with the results of the laboratory rating program reported in the companion paper, will serve as the basis of the continuing studies aimed at developing the best possible correlation between road and laboratory knock ratings.

Work similar to that reported here has been conducted concurrently in England by the Institution of Petroleum Technologists, using British cars and fuels. An exchange of information between the British and American groups working on this problem is being made. This commendable cooperation is indicative of the wide scope of these studies, and it is to be hoped that, as an outgrowth of this extensive cooperation, a solution will be found for the problem of making the laboratory ratings of motor fuels agree with ratings in cars which will be so satisfactory as to be universally acceptable.

EVER since the first cooperative road knock tests were made at Uniontown, Pa., in 1932,[1] it has been the policy of the Cooperative Fuel Research Committee to sponsor such additional road tests as have appeared to be necessary in order to keep the correlation between laboratory knock ratings and road knock ratings up to date. In accordance with this policy, an extensive series of road tests was again made at Uniontown in 1934, the results of which were reported to this Society in January, 1935.[2] In both cases it was concluded that, on the whole, a satisfactory degree of correlation existed between average road knock ratings and laboratory knock ratings as determined by the C.F.R. motor method currently designated as A.S.T.M. Method D-357-36T.

Continuing this policy of keeping the correlation up to date, about a year ago the Cooperative Fuel Research Committee again authorized an immediate and active study of the correlation between laboratory knock ratings and road knock ratings in current automobiles. As a preliminary step in this study, a survey was made of road-test data that were made available to the committee by individual companies on 1934, 1935, and 1936 cars. This survey, which was completed in December, 1936, indicated certain departures from correlation between laboratory knock ratings and road knock ratings. Accordingly, a program of coordinated but decentralized road tests on 1937 cars to be carried out by individual cooperating companies was authorized on Jan. 9, 1937, for the purpose of obtaining more extensive data on this subject.

A Road-Test Planning Group was appointed to cooperate with the Automotive Survey Committee of the American Petroleum Institute in making arrangements for the tests.[3]

Participation

An invitation to participate in these tests was extended to all organizations which might be in a position to take an active part in them. Among the contributors to this work were a number of companies who had not taken part previously in programs of this kind and whose participation was welcomed both because of the added strength given to the data thereby and because of the wider appreciation of the meaning of these tests which naturally would result from the widest possible representation.

In all, 24 companies volunteered to take part in this program and they are listed below:

The Atlantic Refining Co.
Chrysler Corp.
Continental Oil Co.
Deep Rock Oil Corp.
Ethyl Gasoline Corp.
General Motors Corp.
Gulf Research and Development Co.
Mid Continent Petroleum Co.
National Bureau of Standards.
Pacific Coast Technical Group:
- General Petroleum Corp. of California.
- Shell Oil Co.
- Standard Oil Co. of Calif.
- Tide Water Associated Oil Co., Associated Division.
- Union Oil Co. of Calif.

Panhandle Refining Co.
Phillips Petroleum Corp.
Shell Petroleum Corp.
Sinclair Refining Co.
Skelly Oil Co.
Socony-Vacuum Oil Co., Inc.
Standard Oil Development Co.
Standard Oil Co. (Ind.)
The Texas Co.
Tide Water Associated Oil Co.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 14, 1938.]

[1] See S.A.E. TRANSACTIONS, Vol. 28, March, 1933, pp. 105-120: "Anti-knock Research Coordinates Laboratory and Road Results," by C. B. Veal, H. W. Best, J. M. Campbell, and W. M. Holaday.

[2] See S.A.E. TRANSACTIONS, Vol. 30, May, 1935, pp. 165-179: "C.F.R. Committee Report on 1934 Detonation Road Tests," by C. B. Veal.

[3] Personnel of the C.F.R. Road-Test Planning Group: T. A. Boyd (chairman), A. L. Clayden, L. L. Davis, Graham Edgar, J. B. Macauley, J. R. MacGregor, J. B. Rather, and J. R. Sabina.

The A.P.I. representatives of the Automotive Survey Committee were: A. L. Clayden, B. R. Carney, and Neil MacCoull.

Table 1 – Description of Summer and Winter Fuels

Two series of test gasolines were employed: *a*, a series having winter volatility characteristics and designated by the prefix letter *W*; *b*, a corresponding series of fuels of summer volatility and designated by the prefix letter *S*.

Fuel Designation			Added Material	
Winter	Summer	Type	Winter	Summer
WA-1*	SA-1*	Pennsylvania straight-run	$PbEt_4$ (2 cc.)	$PbEt_4$ (2 cc.)
WB-1	SB-1	Mid Continent (50 per cent straight-run, 50 per cent cracked)	None	None
WB-2*	SB-2*	Mid Continent (30 per cent straight-run, 70 per cent cracked)	$PbEt_4$ (0.2 cc.)	$PbEt_4$ (0.6 cc.)
WB-3*	SB-3*	Mid Continent (70 per cent straight-run, 30 per cent cracked)	$PbEt_4$ (2.2 cc.)	$PbEt_4$ (2.8 cc.)
WB-4*	SB-4*	Mid Continent (50 per cent straight-run, 50 per cent cracked)	Alcohol (10 per cent)	Alcohol (10 per cent)
WB-5*	SB-5*	Mid Continent (50 per cent straight-run, 50 per cent cracked)	Benzol (18 per cent)	Benzol (18 per cent)
WC-1	SC-1	California straight-run (100 per cent)	None	None
WC-2	SC-2	California straight-run (100 per cent – not same as WC-1)	$PbEt_4$ (0.5 cc.)	$PbEt_4$ (0.62 cc.)
WD-1		Pennsylvania cracked (100 per cent)	None	
WD-2*		Pennsylvania cracked (100 per cent)	$PbEt_4$ (0.4 cc.)	
	SD-3	Pennsylvania cracked (100 per cent)		$PbEt_4$ (0.92 cc.)
WE-1	SE-1	Mid Continent reformed (100 per cent)	None	None
WE-2*	SE-2*	Mid Continent reformed (100 per cent)	$PbEt_4$ (0.3 cc.)	$PbEt_4$ (0.3 cc.)
WF-1		California (40 per cent straight-run, 60 per cent cracked)	None	
	SF-1*	California (34 per cent straight-run, 66 per cent cracked)		None
WF-2*	SF-2*	Blend of California cracked and straight-run	$PbEt_4$ (0.2 cc.)	$PbEt_4$ (0.5 cc.)
WG-1*		Mid Continent cracked (100 per cent)	None	
WH-1*	SH-1*	Coastal cracked (100 per cent)	None	None
WH-2	SH-2	Coastal cracked (100 per cent)	$PbEt_4$ (0.48 cc.)	$PbEt_4$ (1.6 cc.)
WI-1	SI-1	East Texas cracked (100 per cent)	$PbEt_4$ (0.65 cc.)	$PbEt_4$ (0.74 cc.)
WI-2	SI-2	East Texas straight-run (100 per cent)	$PbEt_4$ (0.73 cc.)	$PbEt_4$ (0.83 cc.)

* Preferred.

Fuels Used in 1937 Road Tests

Two series of fuels were designated for this work. One of these was a series of 19 different fuels of conventional winter volatility and representing a wide variety of gasolines; the other was a corresponding series representing the same types of gasoline but having summer volatility characteristics. A tabulation of these fuels is given in Table 1. Since one of the purposes of the tests was to investigate the effects of atmospheric temperatures on road knock ratings, the fuels in the series having winter volatility, and designated as the *W* series, were tested at comparatively low atmospheric temperatures. The fuels having summer volatility characteristics, designated as the *S* series, were tested at summer atmospheric temperatures.

Preference was given to 10 of the *W* series and 9 of the *S* series fuels when, for some reason, any one organization was unable to use all the fuels listed. As a result, the data on the preferred fuels are more complete than those pertaining to the remaining fuels. The preferred fuels are so designated (by an asterisk) in Table 1.

Tables 2 and 3 give the laboratory ratings of the winter and summer fuels by the Motor (A.S.T.M.), Research, and L-3[4] Methods as reported by the various cooperating companies at the same time that the road ratings were reported. The averages of the July and August "exchange-group"[5] ratings on the fuels are included for comparison with the average of the ratings submitted by the "road group."

It was believed that, since the road group made both road and laboratory ratings on samples from the same drum of fuel, their average laboratory ratings were more representative for comparison with the road ratings. Comparison of the road-group ratings by the Motor Method with those made some months later by the July exchange group indicated that there had been no significant change in the fuels during storage, however. Because only a limited number of ratings using the L-3 method were submitted by the road group, it

[4] The L-3 Method involves the use of the same procedure as the Motor (A.S.T.M.) Method except for the following conditions: mixture, 260 deg. fahr.; spark, 16 deg. at a compression ratio of 5:1; and a knock intensity equivalent to that produced by a 65-octane fuel at a compression ratio of 5.5:1.

[5] A group of 20 laboratories which cooperate in making a monthly check on knock ratings. Usually three different samples are distributed each month.

Table 2 – Laboratory Ratings of Winter Fuels

	Motor (ASTM)		Research		L-3	
Fuel	Road Group	Exchange Group	Road Group	Exchange Group	Road Group	Exchange Group
WA-1	69.2	69.8	68.9	69.1	69.8	70.5
WB-1	66.6	66.6	68.6	67.8	67.0	67.3
WB-2	69.1	69.2	70.9	71.0	69.9	70.2
WB-3	77.8	78.2	80.1	79.8	78.9	79.4
WB-4	72.6	72.2	76.6	75.7	73.7	73.7
WB-5	71.3	70.7	75.0	74.4	71.8	71.9
WC-1	71.1	70.9	71.5	72.1	71.6	71.8
WC-2	71.3	72.2	69.9	70.6	72.0	72.7
WD-1	66.6	66.9	68.8	68.9	67.8	69.2
WD-2	69.7	69.5	71.6	71.4	71.0	70.8
WE-1	68.2	68.2	70.5	70.5	69.3	69.4
WE-2	70.8	70.8	73.1	73.1	72.0	72.4
WF-1	70.7	71.4	77.6	77.1	72.6	73.9
WF-2	69.5	69.2	71.4	71.0	70.6	70.7
WG-1	69.6	69.7	78.6	75.6	71.9	72.2
WH-1	75.6	75.7	82.0	81.5	77.6	77.6
WH-2	79.8	80.2	86.0	85.7	80.4	82.0
WI-1	70.4	71.3	72.3	73.4	70.7	73.2
WI-2	70.5	70.9	69.8	69.2	70.6	71.4

Table 3 – Laboratory Ratings of Summer Fuels

	Motor (ASTM)		Research		L-3	
Fuel	Road Group	Exchange Group	Road Group	Exchange Group	Road Group	Exchange Group
SA-1	68.9	68.8	68.7	68.3	69.8	70.0
SB-1	63.2	63.6	66.1	65.8	65.2	64.7
SB-2	70.1	70.5	72.6	72.8	71.3	72.3
SB-3	78.1	78.3	80.7	80.9	79.1	80.1
SB-4	69.7	70.0	74.2	74.2	71.4	71.8
SB-5	67.8	68.0	73.0	72.7	70.0	69.8
SC-1	72.6	72.0	73.2	72.8	...	73.2
SC-2	73.0	73.2	72.2	71.7	...	73.7
SD-3	71.4	71.0	72.4	72.1	72.3	72.1
SE-1	68.4	68.1	70.9	70.5	...	69.5
SE-2	70.7	70.7	73.3	73.3	72.0	72.5
SF-1	70.4	70.8	76.1	74.9	73.5	72.3
SF-2	69.8	69.8	72.4	72.8	71.7	71.5
SH-1	71.3	71.6	77.9	76.6	74.4	73.5
SH-2	79.6	79.5	84.3	84.3	...	81.9
SI-1	68.4	68.6	69.8	70.2	69.4	70.3
SI-2	70.6	69.9	69.4	69.2	71.3	70.3

was felt desirable to use the exchange-group L-3 ratings for comparison with the road ratings for both summer and winter fuels.

Distillation characteristics, A.P.I. gravities, vapor pressures, and lead analyses for the test fuels are given in Tables 4 and 5.

1937 Cars

The selection of cars for these tests was based on approximate relative sales volume. Originally there were at least two cars of each make chosen for the tests and, for the more popular cars, more than this number were included. Although in the data as reported there have been some deviations from the original choice of cars so that, in some cases, only one car of certain makes was tested, the number of individual cars of each make has been roughly proportional to relative sales volume in both the winter and summer tests. A list of the cars included in these tests and the number of each make is given in Table 6.

Due to the wide participation in these tests, it has been possible to include in these about three times as many different cars as were used in the 1934 tests at Uniontown.

The actual road tests were made at various points throughout the country, and the results submitted to the secretary of the C.F.R. Motor Fuels Committee. These results have been compiled and reported to the Motor Fuels Section by a special "analyzing group" composed of men familiar with the details of the road-test procedure.[6] The data as presented herein represent a condensation of the report made by that group.

Obtaining the Average Road Rating

There are several alternative methods by means of which an average road rating for a given fuel may be computed from the individual road ratings submitted. The three methods considered are discussed below:

(a) Straight arithmetical average of the average ratings in each make of car.

(b) Average weighted strictly according to relative sales volume.

(c) Average weighted according to number of individual car ratings.

In Method (*a*) – "straight arithmetical average of the average car ratings for each make of car" – equal weight is given

[6] The personnel of the analyzing group was as follows: J. M. Campbell, General Motors Corp.; H. J. Gibson, Ethyl Gasoline Corp.; L. E. Hebl, Shell Petroleum Corp.; W. M. Holaday, Socony-Vacuum Oil Co., Inc.; O. D. Horne, Continental Oil Co.; B. A. Kulason, The Texas Co.; H. A. Mason, General Petroleum Corp.; and J. R. Sabina, The Atlantic Refining Co.

Table 4 – Summary of Winter-Fuel Inspection Data

	Gravity (Deg. API)			10 Per Cent Evaporated			50 Per Cent Evaporated			90 Per Cent Evaporated			Reid Vapor Pressure (Lb.)			Average cc. of Lead Per Gallon
Fuel	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	
WA-1	66.5	66.9	65.9	119	131	109	234	245	225	354	372	346	12.2	13.1	10.5	1.9
WB-1	67.5	67.6	67.3	111	117	106	200	207	195	316	322	310	12.1	14.2	11.0	0.0
WB-2	66.2	68.1	65.7	110	115	106	209	216	197	338	354	328	12.2	13.4	11.5	0.3
WB-3	68.1	68.4	67.8	108	115	103	201	205	196	320	347	307	12.8	13.4	12.1	2.0
WB-4	65.2	65.6	65.0	114	117	109	158	162	157	314	320	311	11.7	13.3	11.0	0.0
WB-5	59.1	59.5	58.7	125	140	119	187	202	183	308	315	304	10.4	11.1	8.8	0.0
WC-1	62.6	62.8	62.4	128	131	126	218	221	213	307	309	303	8.8	10.1	7.9	0.0
WC-2	61.6	61.8	61.4	137	140	132	230	232	228	322	325	323	8.4	8.9	8.0	0.6
WD-1	62.2	62.5	61.9	118	129	110	242	250	238	353	361	348	12.3	13.0	11.2	0.0
WD-2	61.5	61.7	61.2	124	131	113	246	252	239	355	361	351	11.3	11.6	10.6	0.3
WE-1	63.1	63.5	62.9	133	142	126	207	209	204	321	327	319	9.7	10.1	9.4	0.0
WE-2	62.8	63.0	62.5	134	140	122	208	215	203	323	334	315	9.9	10.3	9.4	0.3
WF-1	59.8	60.0	59.4	117	124	111	239	243	236	360	363	358	10.9	11.9	9.9	0.0
WF-2	59.6	60.0	59.1	121	131	99	254	257	249	351	360	347	9.2	10.9	9.8	0.2
WG-1	60.6	60.9	60.2	115	125	108	243	258	232	351	365	346	11.6	12.4	10.7	0.0
WH-1	65.4	65.7	65.1	128	133	125	192	196	188	271	277	269	9.6	11.4	9.0	0.0
WH-2	65.5	65.7	65.2	126	131	124	191	194	189	273	277	270	9.9	13.1	8.4	0.4
WI-1	61.9	61.9	61.9	122	124	120	243	245	241	346	349	345	11.4	11.5	11.2	0.6
WI-2	64.3	64.6	64.1	122	127	118	229	233	227	341	346	338	10.7	11.1	10.1	0.8
A-5	61.5			191			262			341			3.7			
C-10	67.5			165			193			240			6.2			

Table 5 – Summary of Summer-Fuel Inspection Data

	Gravity (Deg. API)			10 Per Cent Evaporated			50 Per Cent Evaporated			90 Per Cent Evaporated			Reid Vapor Pressure (Lb.)			Average cc. of Lead Per Gallon
Fuel	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maxi mum	Minimum	Average	Maximum	Minimum	
SA-1	64.6	65.3	64.1	133	142	125	241	245	229	354	358	347	8.8	10.0	5.5	1.9
SB-1	61.7	61.9	61.4	143	149	138	242	244	239	340	344	336	9.4	9.8	8.2	0.0
SB-2	59.8	60.3	59.5	153	157	133	253	258	244	351	360	340	8.8	9.5	7.0	0.6
SB-3	61.6	62.0	61.4	140	152	130	236	242	224	335	342	326	8.2	9.0	7.7	2.8
SB-4	60.2	60.5	59.2	132	137	129	228	232	221	333	340	327	8.6	10.6	7.4	0.0
SB-5	54.4	54.6	54.2	147	153	141	206	211	201	328	336	324	8.1	8.7	7.3	0.0
SC-1	61.2	61.3	61.1	146	150	142	214	216	212	310	312	306	6.6	7.6	5.7	0.0
SC-2	59.5	59.7	59.4	140	145	137	247	248	246	344	350	341	6.7	7.2	6.5	0.6
SD-3	63.8	64.6	63.4	140	145	133	231	234	226	321	324	314	7.5	8.5	6.5	0.9
SE-1	62.8	63.2	62.6	138	140	132	209	210	207	325	330	319	9.0	9.6	8.6	0.0
SE-2	62.9	63.0	62.5	136	143	124	208	215	202	328	335	320	9.3	10.3	8.4	0.3
SF-1	58.2	58.9	57.6	150	157	144	239	243	232	340	347	335	6.6	7.0	5.9	0.0
SF-2	57.8	58.3	57.2	149	157	141	259	262	255	354	362	349	7.3	7.9	6.3	0.5
SH-1	59.2	59.5	59.1	153	157	147	230	232	225	316	320	304	6.0	6.7	5.4	0.0
SH-2	58.8	59.2	58.1	155	159	151	231	234	228	318	320	317	5.5	6.3	4.6	1.5
SI-1	59.6	59.9	59.2	150	156	146	257	265	253	352	354	350	6.8	7.4	6.0	0.7
SI-2	62.4	62.8	62.2	140	145	136	238	241	236	341	345	339	7.9	8.6	6.6	0.8
A-5	61.5			191			262			341			3.7			
C-10	67.5			165			193			240			6.2			

Table 6 – 1937 Cars Used in Road Tests

Name of Car	Number of Cars Fuels W	S
Ford 60	4	4
Ford 85	10	7
Chevrolet	12	12
Plymouth	6	5
Dodge	4	3
Pontiac 6	2	2
Pontiac 8	3	3
Oldsmobile 6	3	2
Oldsmobile 8	1	1
Buick 40	1	1
Buick 60	1	1
Buick 80	0	1
Packard 6	2	1
Packard 120	2	1
Terraplane	1	1
Hudson	1	1
Chrysler 6	2	2
Chrysler 8	2	1
Lafayette	1	1
Studebaker 6	2	2
Studebaker 8	1	1
De Soto	2	2
Cadillac	1	1
Total	64	56

to each make of car. In following this procedure, a single best rating of a given fuel is first determined for each make of car. This rating may be an average of a number of independent ratings made in different cars of the same make, or it may be only a single rating where only one car of that make was included in the data. The weakness of this method is due: first, to the fact that results from cars of limited sales volume are given equal weight with cars of large sales volume; and, second, to the fact that a single car rating in one make of car is given the same weight as an average car rating based on several ratings in another make of car.

Method (*b*) – "weighting strictly according to sales volume" – begins with the same average ratings for individual makes of car that are used in method (*a*). These averages for each make of car are multiplied by the fractional proportion of total sales for each car make, and the products are added to give the weighted average. This method has disadvantages where a certain fuel has been rated in only a limited number of cars.

Method (*c*) – "weighting according to the number of ratings submitted" – gives equal weight to each individual car rating. Since the number of cars of each make selected for these tests correspond, within reasonable limits, to the respective sales volume for each make of car, this method of averaging weights the data in approximate accord with relative sales volume.

A comparison of the results obtained by each of these three methods of computation using a sensitive type of fuel (WG-1) is shown in Table 7. In this example the average weighted strictly according to sales volume was equal to that based on weighting according to number of ratings reported. This agreement is the result of the similarity between the relative sales volume and the distribution of cars of each make used in these tests. The arithmetical average obtained by Method (*a*) differed from the other two averages by 0.5 of an octane number. In this case the straight arithmetical average was influenced appreciably upward by a comparatively few high ratings in cars representing a low percentage of the total sales volume.

Inasmuch as some measure of weighting according to car sales volume was desired, Method (*c*) was chosen as the most desirable method for use, since it provided the most direct method of using the data submitted without materially changing the average that would have been obtained by using Method (*b*) even if that method could have been applied conveniently.

Tabulation of Ratings

The average car and laboratory ratings for each fuel are tabulated in Tables 8 to 11. In these tables all ratings, both laboratory and road, are expressed in terms of the octane number of equivalent reference-fuel blends using the accepted Motor (A.S.T.M.) Method calibration curve for reference fuels C-10 and A-5. The cars have been coded according to make, and the ratings entered for each make of car represent the average rating on each fuel, irrespective of whether the fuel was rated in one car or in 10 cars of that make. The average road ratings tabulated at the foot of each column in these tables are weighted in proportion to the number of ratings reported for each make of car in accordance with the procedure outlined under Method (*c*) in the previous section. It should be recognized that these average road ratings which are recommended for correlation purposes are averages and, as such, do not apply specifically to any one make of car. The tabulation of the ratings for each make of car clearly shows how wide the spread actually was among different makes of cars. Furthermore, the original data showed nearly the same spread in ratings among different cars of the same make. Even in one car there was a considerable spread in ratings when ratings were repeated from day to day. For these reasons, there is a considerable degree of uncertainty about the individual car ratings on all but the three low-priced cars of which there were from four to twelve cars of each make.

Effect of Atmospheric Temperature

To determine the effect of atmospheric temperature on road ratings, similar fuels were rated in the winter and sum-

Table 7 – Road Ratings Obtained by Various Weighting Methods

Fuel WG-1

1 Car	2 No. of Cars	3 Average Car Octane	4 Sum of Individual Ratings	5 Per Cent Sales Volume	6 Product Columns 3, 5
A	12	72.1	865.9	21.2	1,528.5
B	6	76.5	459.1	13.6	1,040.4
C,D*	13	73.1	943.2	25.5	1,864.1
E,F*	4	73.2	293.0	2.4	175.7
G	3	74.3	222.8	7.4	549.8
I,J*	2	72.4	144.8	5.5	398.2
K,L*	5	75.2	380.2	6.0	451.2
M,N*	2	79.1	158.2	5.4	427.1
O	1	67.2	67.2	1.1	73.9
P,Q*	4	72.9	295.2	2.9	211.4
V,R*	1	69.7	69.7	2.7	188.2
S	1	83.9	83.9	2.1	176.2
T,U*	3	76.2	227.3	2.1	160.0
Totals 13	57	965.8	4,210.5	97.9 †	7,244.7

Weighting according to car sales volume $\left\{\frac{\text{column 6}}{\text{column 5}}\right\}$ Mean octane 73.9

Weighting according to car ratings reported $\left\{\frac{\text{column 4}}{\text{column 2}}\right\}$ Mean octane 73.9

Equal weighting given to each make (arithmetic average) $\left\{\frac{\text{column 3}}{\text{column 1}}\right\}$ Mean octane 74.4

* Different models of the same make, for example, a 6 and an 8 listed together in sales figures.

† Car "H," representing 2 per cent of car sales, not included because no ratings were reported on this fuel in this car.

Table 8 – Average Car and Laboratory Ratings – 10 Preferred Winter Fuels

(Octane Number)

Car	WA-1	WB-2	WB-3	WB-4	WB-5	WD-2	WE-2	WF-2	WG-1	WH-1
A	67.9	68.1	77.2	74.4	74.5	69.0	70.8	69.8	72.1	78.6
B	67.6	70.2	77.5	75.4	74.6	68.2	69.2	70.5	76.5	78.9
C	67.0	68.2	75.8	72.6	73.7	69.2	70.5	69.2	71.7	80.4
D	68.7	68.6	79.1	72.3	73.7	68.8	70.8	68.8	74.4	80.3
E	69.0	70.8	77.0	73.7	73.5	69.8	74.0	71.5	73.7	78.8
F	66.0	69.0	74.0	...	75.0	69.0	71.0	72.0	72.7	79.0
G	68.5	69.4	79.2	77.0	75.7	70.8	72.5	71.7	75.4	78.7
H	67.8	67.2	...	...	75.0	68.0	69.1	69.1	...	...
I	65.8	65.8	79.1	73.3	71.0	65.8	69.8	...	68.3	78.8
J	68.3	69.0	77.5	74.5	73.0	71.0	71.9	71.7	74.2	77.9
K	69.3	68.8	78.6	76.7	73.0	71.0	71.7	72.5	77.0	78.0
L	66.0	66.0	77.7	75.2	74.1	64.5	72.0	71.7	74.6	78.7
M	67.6	72.3	78.8	...	75.1	70.8	71.3	71.7	80.6	80.6
N	68.2	68.6	78.4	75.4	74.9	71.3	72.0	72.3	77.6	79.5
O	66.8	66.7	75.0	69.7	71.0	67.8	73.0	70.4	67.2	75.0
P	65.4	66.8	79.2	75.7	72.6	68.5	72.3	70.5	71.1	78.9
Q	68.8	69.2	77.1	74.8	75.3	...	72.4	69.2	75.3	80.2
R	66.3	...	...	...	77.0	...	...	...	...	81.8
S	67.6	78.8	80.8	78.8	78.8	78.8	77.0	78.8	83.9	81.7
T	67.9	68.8	78.6	75.5	77.2	68.3	72.2	70.8	74.8	79.2
U	70.2	70.6	77.9	75.7	76.2	70.7	73.5	68.2	77.6	79.9
V	69.3	67.5	75.3	...	...	69.1	69.9	67.2	69.7	...
Average road rating *	67.8	68.8	77.6	74.4	74.2	69.1	71.2	70.4	73.9	79.0
Average motor	69.2	69.1	77.8	72.6	71.3	69.7	70.8	69.5	69.6	75.6
Average research	68.9	70.9	80.1	76.6	75.0	71.6	73.1	71.4	78.6	82.0
Average L-3	70.5	70.2	79.4	73.7	71.9	70.8	72.4	70.7	72.2	77.6

* Average road rating weighted in proportion to number of ratings in each make of car. Expressed in octane number.

Table 9 – Average Car and Laboratory Ratings – Winter Fuels

(Octane Number)

Car	WB-1	WC-1	WC-2	WD-1	WE-1	WF-1	WH-2	WI-1	WI-2
A	64.9	70.3	68.8	66.8	70.6	73.2	83.1	70.0	67.0
B	...	70.6	...	...	71.0	...	...	...	...
C	64.7	71.3	68.4	64.4	68.2	72.2	84.5	69.0	68.1
D	66.5	71.0	70.0	66.8	69.0	74.9	84.0	71.0	66.0
E	67.0	71.3	69.5	68.5	70.0	73.0	80.5	70.0	68.8
F	...	...	...	...	...	...	...	...	...
G	66.3	73.4	70.0	69.7	71.2	71.7	82.5	...	68.0
H	...	...	...	...	...	...	...	...	...
I	...	...	...	...	...	69.8	...	...	...
J	71.0	71.8	72.0	69.5	68.0	73.5	74.5	72.0	71.0
K	...	...	...	...	...	...	...	...	...
L	...	...	...	...	...	...	...	...	...
M	...	71.3	...	...	72.1	...	...	...	...
N	...	...	...	...	...	...	...	...	...
O	65.3	67.9	67.8	66.5	67.8	73.0	75.0	69.1	69.0
P	65.0	72.0	68.0	65.3	68.8	73.9	80.5	68.5	68.7
Q	65.0	...	...	...	69.0	...	...	...	...
R	...	69.1	...	...	67.6	...	...	...	...
S	...	73.0	...	...	74.3	...	...	...	...
T	...	71.3	...	...	68.5	...	...	...	...
U	...	...	...	...	...	...	...	...	...
V	...	...	...	...	...	...	...	...	...
Average road rating *	65.5	71.3	69.1	66.7	69.6	72.9	81.1	69.7	68.2
Average motor	66.6	71.1	71.3	66.6	68.2	70.7	69.8	70.4	70.5
Average research	68.6	71.5	69.9	68.8	70.5	77.6	86.0	72.3	69.8
Average L-3	67.3	71.8	72.7	69.2	69.4	73.9	82.0	73.2	71.4

* Average road rating weighted in proportion to number of ratings in each make of car. Expressed in octane number.

Table 10 – Average Car and Laboratory Ratings – 9 Preferred Summer Fuels

(Octane Number)

Car	SA-1	SB-2	SB-3	SB-4	SB-5	SE-2	SF-1	SF-2	SH-1
A	66.9	68.3	77.4	69.7	69.5	70.5	72.4	69.7	74.3
B	67.0	70.2	79.0	68.6	70.2	71.2	73.8	70.2	75.2
C	66.1	68.1	74.8	68.8	68.7	68.1	69.3	68.9	71.1
D	67.4	68.1	77.0	70.8	71.9	70.9	73.9	69.8	75.3
E	68.0	69.8	80.0	72.0	71.0	72.2	74.0	70.5	74.8
F	66.0	...	...	...	70.5	72.0	70.5	72.0	71.5
G	68.7	69.5	74.6	70.0	69.0	70.6	71.2	70.6	72.2
H	68.4	71.8	...	70.2	70.0	71.0	70.0	71.5	73.0
I	67.2	66.4	78.8	66.4	67.2	69.9	69.9	68.4	72.2
J	68.5	70.0	78.0	72.0	72.0	71.2	70.8	70.2	73.5
K	67.7	69.5	76.2	70.0	71.5	71.0	74.0	69.2	73.0
L	67.8	68.2	78.2	68.7	68.6	69.5	72.6	69.7	74.0
M	69.5	73.2	78.8	73.6	75.5	71.4	74.5	72.2	74.5
N	67.4	71.0	78.3	72.5	75.7	70.6	72.5	72.5	77.7
O	67.0	71.4	...	66.3	66.6	71.0	...	71.0	71.0
P	66.3	68.6	78.2	74.4	71.6	71.8	76.0	67.9	76.0
Q	68.7	70.9	79.0	68.8	70.2	...	69.0	71.5	73.5
R	67.6	67.6	76.9	72.3	71.3	68.6	75.1	71.3	75.2
S	72.3	72.3	79.8	75.1	78.2	75.1	79.2	77.9	76.6
T	69.6	71.9	80.1	72.4	73.2	73.1	74.3	72.8	74.2
U	70.6	69.9	...	...	77.4	74.7	77.1	74.9	78.0
V	65.2	...	...	...	...	...	...	...	...
W	68.0	70.0	76.0	...	...	70.0	70.0	70.0	...
Average road rating *	67.5	69.2	77.8	70.2	70.6	70.7	72.6	70.3	73.9
Average motor octane	68.9	70.1	78.1	69.7	67.8	70.7	70.4	69.8	71.3
Average research octane	68.7	72.6	80.7	74.2	73.0	73.3	76.1	72.4	77.9
Average L-3 octane	70.0	72.3	80.1	71.8	69.8	72.5	72.3	71.5	73.5

* Average road rating weighted in proportion to number of ratings in each make of car. Expressed in octane number.

Table 11 – Average Car and Laboratory Ratings – Summer Fuels

(Octane Number)

Car	SB-1	SC-1	SC-2	SD-3	SE-1	SH-2	SI-1	SI-2
A	63.8	73.2	69.8	70.6	68.6	84.0	68.3	68.2
B	...	...	...	68.7	...	...	68.8	...
C	63.5	72.0	70.0	68.5	66.6	77.0	66.4	67.5
D	64.7	71.2	70.5	69.8	68.5	79.8	66.6	66.8
E	63.5	74.0	72.5	70.5	70.0	84.5	...	66.5
F	...	...	...	...	...	...	...	...
G	64.5	71.4	72.2	71.7	68.0	76.3	...	69.6
H	...	...	...	...	...	...	...	...
I	...	...	...	67.2	...	...	...	...
J	...	...	...	69.8	...	...	...	...
K	...	...	...	70.6	...	...	...	...
L	...	...	...	67.8	...	...	...	...
M	...	74.6	...	...	67.6	...	...	...
N	66.0	...	...	68.2	...	...	...	66.4
O	63.8	74.9	72.4	69.8	68.5	78.0	67.6	70.0
P	59.6	75.4	68.6	...	69.9	79.0	66.6	69.1
Q	64.5	73.0	...	...	...	...	...	...
R	...	73.1	...	...	68.6	...	...	...
S	...	74.5	...	...	74.5	...	...	...
T	...	73.8	...	70.0	72.0	...	...	...
U	66.5	...	...	69.8	...	...	...	68.5
V	...	...	...	...	...	...	...	...
W	...	...	...	69.0	...	...	...	...
Average road rating *	64.0	73.3	70.6	69.6	69.0	79.9	67.4	68.0
Average motor octane	63.2	72.6	73.0	71.4	68.4	79.6	68.4	70.6
Average research octane	66.1	73.2	72.2	72.4	70.9	84.3	69.8	69.4
Average L-3 octane	64.7	73.2	73.7	72.1	69.5	81.9	70.3	70.3

* Average road rating weighted in proportion to number of ratings in each make of car. Expressed in octane number.

Table 12 – Analysis of Road Ratings Made at Different Atmospheric Temperatures

	1	2	3	4	5	6	7	8	9	10	11	12	13
	Winter						Summer						
Fuel	Deg. F. Average	Road	Motor	Research	Research-Motor	Road-Motor	Deg. F. Average	Road	Motor	Research	Research-Motor	Road-Motor	Column 6-Column 12
A-1	42	67.8	69.2	68.9	−0.3	−1.4	86	67.5	68.9	68.7	−0.2	−1.4	0.0
B-1	44	65.5	66.6	68.6	2.0	−1.1	84	64.0	63.2	66.1	2.9	0.8	−1.9
B-2	41	68.8	69.1	70.9	1.8	−0.3	87	69.2	70.1	72.6	2.5	−0.9	0.6
B-3	41	77.6	77.8	80.1	2.3	−0.2	87	77.8	78.1	80.7	2.6	−0.3	0.1
B-4	41	74.4	72.6	76.6	4.0	1.8	85	70.2	69.7	74.2	4.5	0.5	1.3
B-5	42	74.2	71.3	75.0	3.7	2.9	85	70.6	67.8	73.0	5.2	2.8	0.1
C-1	40	71.3	71.1	71.5	0.4	0.2	81	73.3	72.6	73.2	0.6	0.7	−0.5
C-2	47	69.1	71.3	69.9	−1.4	−2.2	82	70.6	73.0	72.2	−0.8	−2.4	0.2
D-1	46	66.7	66.6	68.8	2.2	0.1	..	...	...	...	..	..	..
D-2	44	69.1	69.7	71.6	1.9	−0.6	..	...	...	...	..	..	..
D-3	..	...	...	...	..	..	89	69.6	71.4	72.4	1.0	−1.8	..
E-1	39	69.6	68.2	70.5	2.3	1.4	80	69.0	68.4	70.9	2.5	0.6	0.8
E-2	44	71.2	70.8	73.0	2.2	0.4	86	70.7	70.7	73.3	2.6	0.0	0.4
F-1	45	72.9	70.7	77.6	6.9	2.2	86	72.6	70.4	76.1	5.7	2.2	0.0
F-2	43	70.4	69.5	71.4	1.9	0.9	86	70.3	69.8	72.4	2.6	0.5	0.4
G-1	42	73.9	69.6	78.6	9.0	4.3	..	...	...	...	..	..	..
H-1	41	79.0	75.6	82.0	6.4	3.4	85	73.9	71.3	77.9	6.6	2.6	0.8
H-2	43	81.1	79.8	86.0	6.2	1.3	83	79.9	79.6	84.3	4.7	0.3	1.0
I-1	40	69.7	70.4	72.3	1.9	−0.7	84	67.4	68.4	69.8	1.4	−1.0	0.3
I-2	40	68.2	70.5	69.8	−0.7	−2.3	86	68.0	70.6	69.4	−1.2	−2.6	0.3
Average	42	71.7	71.1	73.8	2.78	+0.6	85	70.9	70.8	73.4	2.5	+0.1	+0.4

mer in the same cars under approximately the same conditions. Laboratory ratings were obtained at the same time. The average atmospheric temperatures for both groups of fuels are shown in Table 12.

To obtain a direct measure of the effect of temperature, ratings were compared on the basis of the difference between road and motor values. This comparison was necessary to eliminate changes in the octane level of the winter and summer fuels. The values are given in column 13 of Table 12, showing an average difference of 0.4 of an octane number between the winter and summer ratings. The discrepancy between +0.5 of an octane number obtained by subtracting

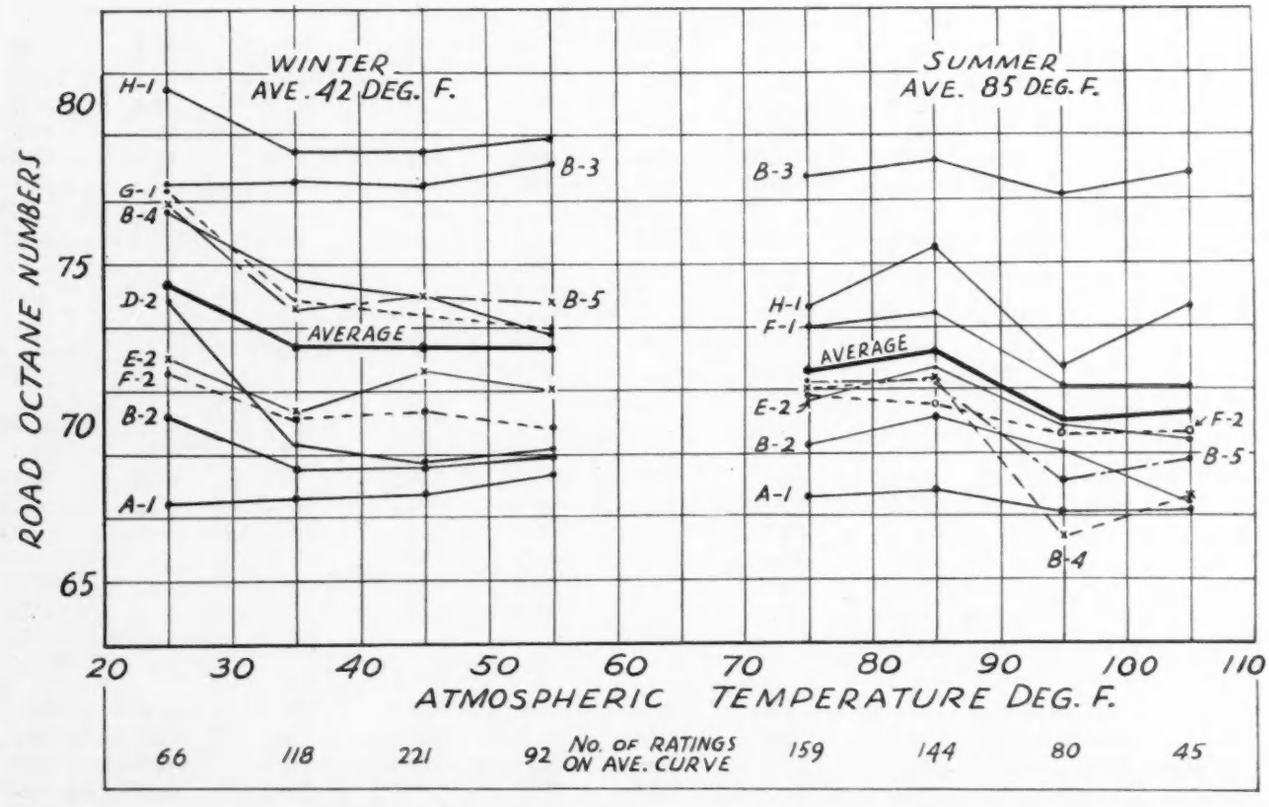


Fig. 1 – Effect of Atmospheric Temperature on Road Ratings – Preferred Fuels

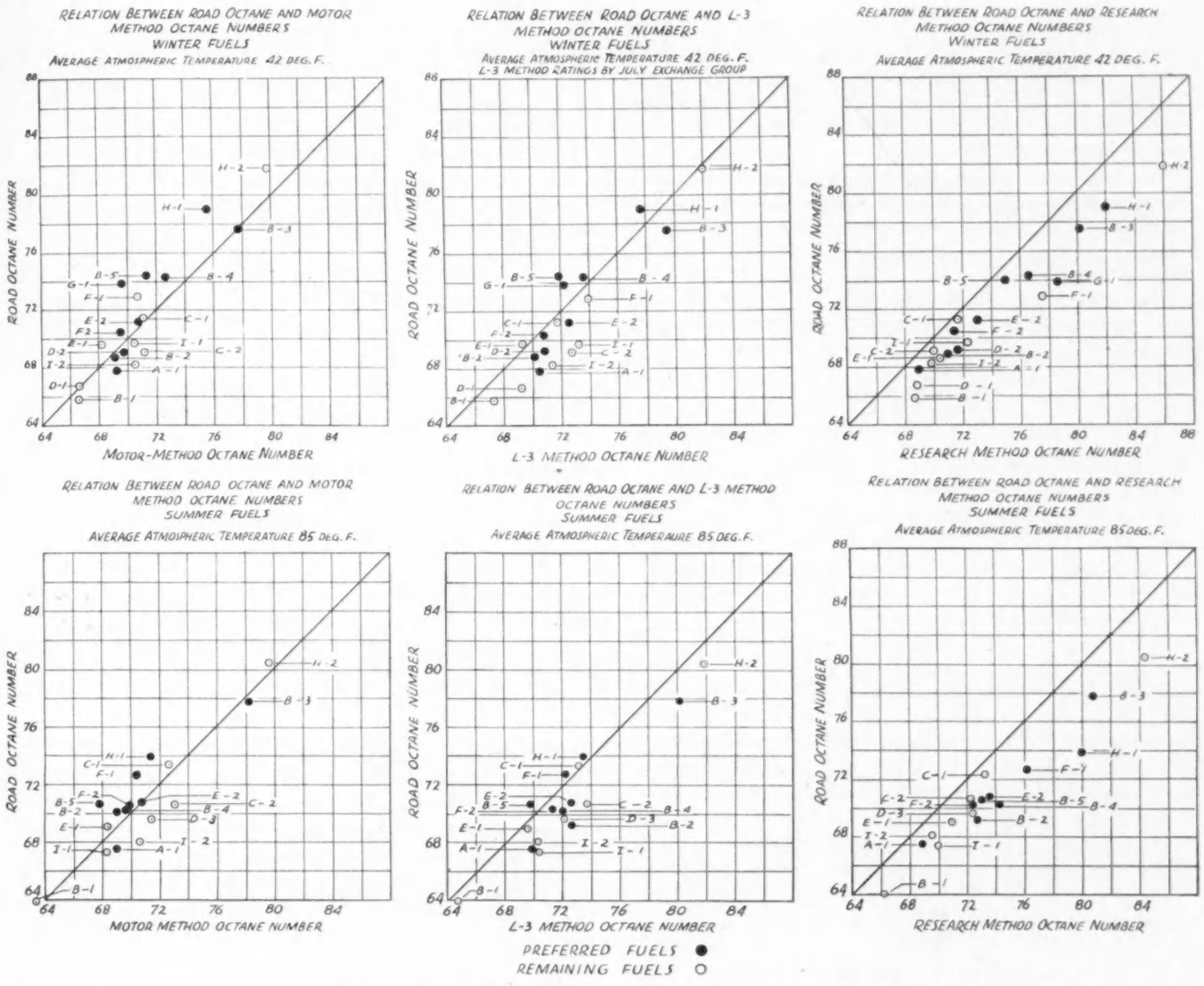


Fig. 2 – Correlation Between Laboratory Methods and Road Ratings for Summer and Winter Fuels

the averages in columns 6 and 12, respectively, in Table 12 and +0.4 of an octane number is presumably because there were no summer fuels corresponding to WD-1, WD-2, and WG-1.

A subdivision of the data on the preferred fuels according to temperature at which they were obtained within each of the two major temperature groups is shown in Fig. 1.

In the winter subdivision, the average curve shows that there was a trend upward in the low-temperature bracket. This trend is believed to be largely due to accidental grouping

Table 13 – Laboratory and Road – Grand Averages

(Octane Number)

	Winter Fuels			Summer Fuels		
	10. Preferred	9. Remaining	All	9. Preferred	8. Remaining	All
Road rating	72.6	70.6	71.7	71.4	70.3	70.9
Motor rating	71.5	70.6	71.1	70.8	70.9	70.8
Research rating	74.8	72.8	73.8	74.3	72.3	73.4
L-3 rating	72.9	72.3	72.6	72.6	72.0	72.3
Road minus motor	1.1	0	0.6	0.6	−0.6	0.1
Road minus research	−2.2	−2.2	−2.1	−2.9	−2.0	−2.5
Road minus L-3	−0.3	−1.7	−0.9	−1.2	−1.7	−1.4
Research minus motor	3.3	2.2	2.7	3.5	1.4	2.6

of the higher-rating cars in this bracket, which unduly influenced the average obtained on relatively few cars. Where sufficient information was available to obtain representative averages, no trend is apparent. Little significance, therefore, can be attached to the increase in octane number at the lowest temperatures.

In the summer subdivision, car distribution again played a preponderant part in the average ratings. As with the winter fuels, the high-rating cars were grouped in the low end of the temperature range, leaving the lower-rating cars at the high end, producing an apparent downward trend with an increase in temperature. Where a sufficient number of ratings were obtained on the same make of car in the various temperature brackets, no definite trend was discernible.

Correlation of 1937 Cars

The correlation between the laboratory methods and road ratings for the summer and winter fuels is shown in Fig. 2.

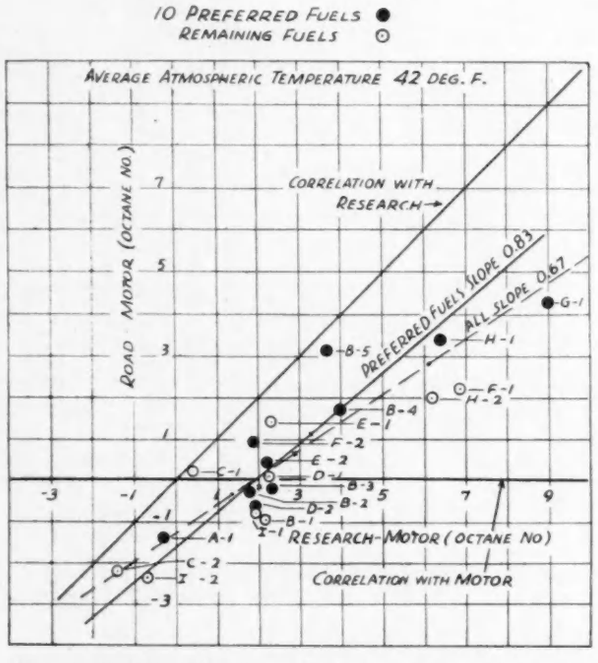


Fig. 3 – Relation Between Road Appreciation from the Motor Method and Fuel Sensitivity – Winter Fuels

The correlation when the fuels were classified according to fuel sensitivity, expressed as the difference between the Research and Motor (A.S.T.M.) Methods, and road appreciation from the Motor (A.S.T.M.) Method is shown in Figs. 3 and 4. Table 13 shows the overall fuel averages by the road and laboratory methods.

It will be noted in Table 13 that, when the ratings of all fuels are included, the Motor (A.S.T.M.) Method appears to give the best average correlation both in winter and summer. An examination of the correlation charts, particularly Figs. 3 and 4, shows, however, that although the correlation is reasonably good for fuels of moderate sensitivities, individual fuels which were either highly sensitive, or which ranged from low to negative sensitivities, deviated appreciably and in a consistent pattern from the average correlation line. The more sensitive fuels were rated above the Motor Rating on the road, and the less sensitive fuels were rated below the Motor Rating on the road; so that, with a fairly even distribution of fuels, the plus and minus deviations offset each other in such a way that the averages in Table 13 are in best agreement with the Motor (A.S.T.M.) ratings. This statement does not mean, however, that the motor method gave satisfactory correlation between road and laboratory ratings. On the contrary, the data show beyond any reasonable doubt that certain types of fuels were rated too low by the Motor Method, and others were rated too high by this method. It can be concluded, therefore, that there is need for a laboratory test method which will give a more satisfactory evaluation of the relative knocking characteristics of all types of fuels than the present Motor (A.S.T.M.) Method.

Significance of These Tests

In addition to the direct application of these data to the important problem of testing and improving the reliability of present knock-testing methods, these tests are of particular significance to the automobile engineer in some respects. In the first place, the wide participation among the oil companies in obtaining these data is indicative of the growing tendency on the part of the oil-producing companies to avail themselves of personnel and equipment for the direct testing of petroleum products, in this case gasoline, in automobiles. But, at the same time that a gasoline is being tested, the car is being tested also. As a result of this work and of the cooperation between the two industries which is cleared through the Cooperative Fuel Research Committee, there are being developed improved methods of testing both fuels and cars in so far as certain aspects of the utilization of gasoline is concerned.

In the particular case under consideration here, knocking characteristics are the main consideration, of course. The data presented in this report illustrate how the existing technical information relative to the behavior of different gasolines in various automobiles can be extended through cooperative programs of this kind. Present indications are that some of the most important questions with respect to this problem can be approached only from a statistical standpoint. Where such is the case, the cooperative method which has been employed in this work has proved highly advantageous, and the automotive and petroleum industries as a whole are accordingly indebted to those companies which have been and are contributing to this program.

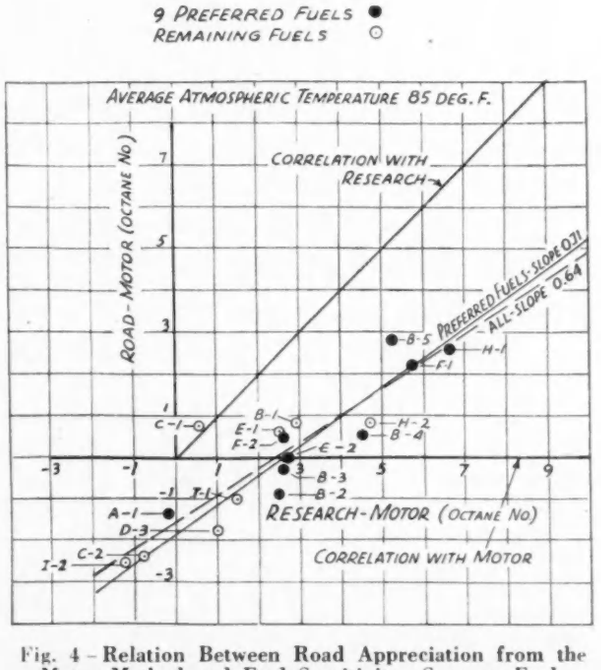


Fig. 4 – Relation Between Road Appreciation from the Motor Method and Fuel Sensitivity – Summer Fuels

Airplane Performance Calculations by Means of Logarithmic Graphs

By Ivan H. Driggs
The Glenn L. Martin Co.

THE ideas presented in this paper are designed to eliminate all the faults of previous formulas or charts for finding airplane performance, and to provide a method which will be satisfactory for all types of propellers, engines, and airplanes. The same equation for the power required is retained as was used previously, but corrections are applied to it to take care of the increase in power as the burble points are reached.

By a simple device employing two parameters determined by the airplane dimensions, the power-required curve is generalized and then plotted logarithmically. The power-available curve is generalized in a similar manner and likewise plotted logarithmically. Then, by a simple calculation, the method of placing properly these two generalized curves one over the other is found. This procedure results in being able to read directly with no plotting the values from which the performance can be obtained.

FROM time to time various short methods of finding the performance of an airplane by means of simple formulas or charts have been developed and have been under wide use among aeronautical engineers. The most notable published examples of such work are the Air Service Information Circular No. 553[1] and N.A.C.A. Technical Report No. 408[2] (listed in chronological order of their appearance). The present writer also developed a chart method similar to N.A.C.A. Technical Report No. 408[2] which never has been published.

All of these methods replace the power-required curve of an airplane by an approximate equation of excellent accuracy throughout the greater portion of the flying range of the airplane. The power-available curve also was treated in a similar manner for all the various schemes, in that the power was assumed to vary as some simple exponential function of the ratio of the flight speed to maximum speed.

Since not all airplanes were equipped with propellers that gave their maximum efficiency at maximum speed, very appreciable errors were introduced when estimating rate of climb with a so-called "climbing propeller."

All of this work also was written before the advent of the constant-speed propeller and the supercharged engine. Although the supercharging calculation could be carried out readily with but little added complication, the constant-speed propeller introduced an obstacle that could not be surmounted without more complexity than the use of a simplified method warranted. This complexity was due to the fact that the power-available curve could not be replaced by a mathematical expression that could be differentiated in a simple manner.

This difficulty suggested the idea that, even if a chart were worked out for the constant-speed propeller, some future change again might destroy its usefulness, and it further pointed out the desirability of devising a method that would be independent of propeller characteristics.

Another difficulty experienced in using the older methods is that the actual power-required curve did not follow the assumed algebraic expression for all airplanes at both ends of the speed range due to the low and high angle of attack burbles. This difficulty resulted in the overestimation of the maximum velocity in some cases, and in the overestimation of ceiling in others. These difficulties cannot be eliminated easily from any chart method.

The ideas presented in this report are designed to eliminate all the previous faults and to provide a method which will be satisfactory for all types of propellers, engines, and airplanes. The same equation for the power required is retained as was used previously, but corrections are applied to it to take care of the increase in power as the burble points are reached.

By a simple device employing two parameters determined by the airplane dimensions, the power-required curve is generalized and then plotted logarithmically. The power-available curve is generalized in a similar manner and likewise plotted logarithmically. Then, by a simple calculation, the method of properly placing these two generalized curves one over the other is found. This operation results in being able to read directly with no plotting the values from which the performance can be obtained.

For different altitudes the generalized curve of power available is shifted to a new position on the curve of power required, and new values are read. The data required for plotting a complete performance graph are thus obtained very rapidly. These charts also are extended to the computation of range.

Although the generalized curve of power available with

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1938.]

[1] See Air Service Information Circular No. 553, February, 1926: "A Simple Theoretical Method of Analyzing and Predicting Airplane Performance," by Ivan H. Driggs.

[2] See N.A.C.A. Technical Report No. 408, 1932: "General Formulas and Charts for the Calculation of Airplane Performance," by W. Bailey Oswald.

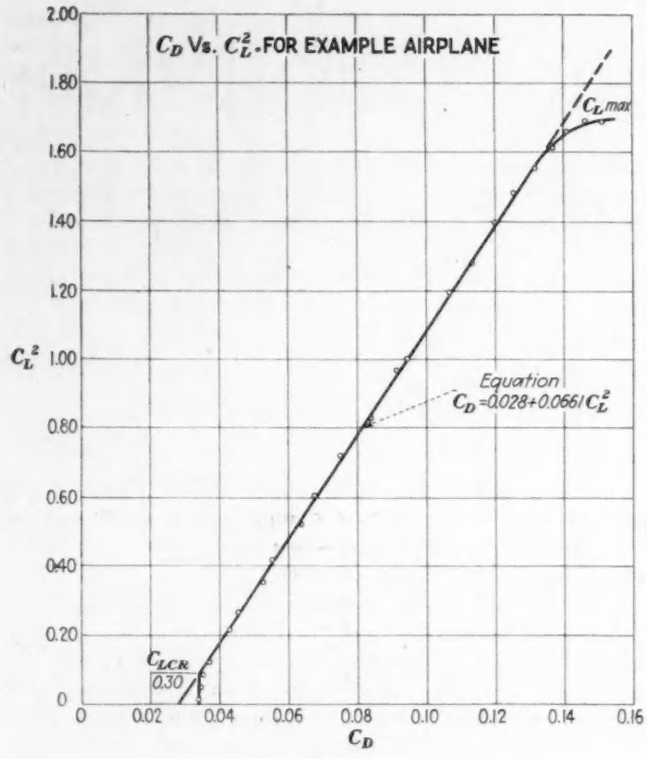


Fig. 1

constant-speed propeller was determined from one particular propeller test, any other propeller may be used and a new curve may be constructed very readily. Similarly tests upon but one propeller are used for the fixed-pitch generalization.

Development of Theory

Power Required

We shall assume without discussion that the polar curve of an airplane may be expressed as:

$$C_D = C_{D_o} + L\,C_L^2 \qquad (1)$$

Fig. 1 shows a wind-tunnel test plotted as C_D vs. C_L^2 to illustrate this equation. Converting to a force equation:

$$\text{Drag} = 0.00256\,f\sigma V^2 + 124.8\left(\frac{W}{b.e}\right)^2 \Big/ \sigma V^2 \qquad (2)$$

Where:

C_D = absolute drag coefficient = Drag/qS.
C_{D_o} = absolute profile drag coefficient.
C_L = lift coefficient = Lift/qS.
L = slope = dC_D/dC_L^2
f = $C_{D_o}S$
S = Area of wing, sq. ft.
V = Velocity, m.p.h.
W = Gross Weight = Lift, lb.
b = Span of wing, ft.
e = airplane *span* efficiency factor.
= $1/\sqrt{\pi AL}$
A = Aspect Ratio.
= b^2/S
σ = Relative density = ρ/ρ_o

Converting Equation (2) to a power equation:

$$\text{T. Hp.}_{req.} = 0.00000683\,f\,\sigma\,V^3 + 0.3329\left(\frac{W}{b.e}\right)^2 \Big/ \sigma V \qquad (3)$$

If Equation (3) be differentiated for the value of V, that will make the power required a minimum, we find:

$$V_{min.\ T.\ Hp.} = 11.28\left(\frac{W}{b.e}\right)^{1/2} \Big/ (f^{1/4}\sigma^{1/2}) \qquad (4)$$

Repeating for Equation (2)

$$V_{min.\ Drag} = 14.85\left(\frac{W}{b.e}\right)^{1/2} \Big/ (f^{1/4}\sigma^{1/2}) \qquad (5)$$

It is evident from the preceding that any speed may be represented by the product of a numerical factor and an algebraic expression depending upon the dimensions of the airplane, that is:

$V = K_v P_v$ when

$P_v = \left(\frac{W}{b.e}\right)^{1/2} \Big/ (f^{1/4}\sigma^{1/2})$ – definition – Speed Parameter.

K_v = a number – Speed Factor.

Substituting the preceding value of V in Equation (3) we have:

$$\text{T. Hp.}_{req.} = \frac{0.00000683\,\sigma f\left(\frac{W}{b.e}\right)^{3/2} K_v^3}{f^{3/4}\,\sigma^{3/2}} + \frac{0.3329\left(\frac{W}{b.e}\right)^2 f^{1/4}}{\sigma\left(\frac{W}{b.e.\sigma}\right)^{1/2} K_v}$$

$$= \frac{0.00000683\ ^{1/4}\,(W/b.e)^{3/2} K_v^3}{\sigma^{1/2}} + \frac{0.3329\,f^{1/4}\,(W/b.e)^{3/2}}{\sigma^{1/2}\,K_v}$$

$$\text{Or} \quad \frac{\text{T. Hp.}\ \sigma^{1/2}}{f^{1/4}\left(\frac{W}{b.e}\right)^{3/2}} = 0.00000683\,K_v^3 + 0.3329/K_v$$

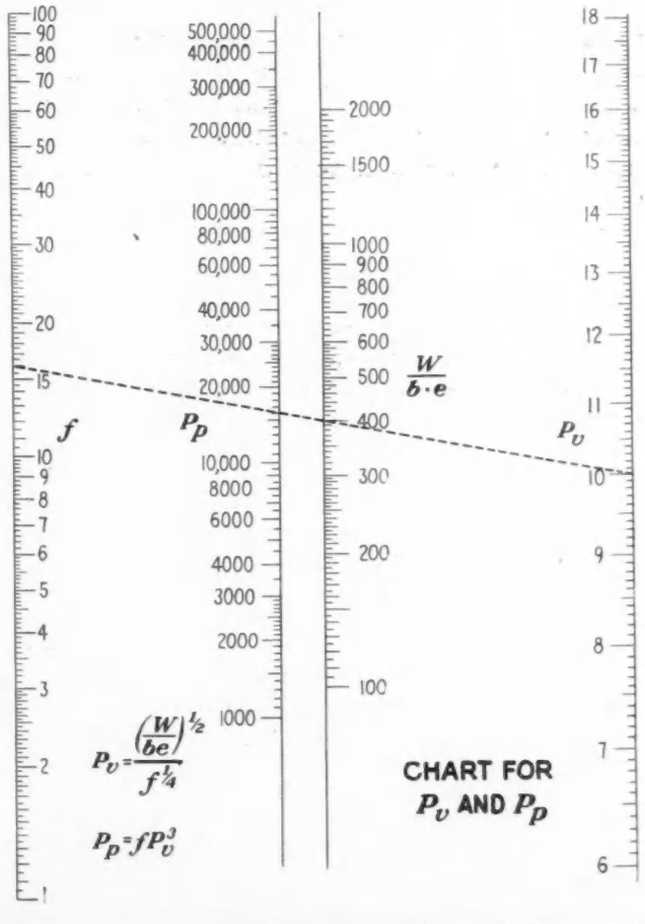


Fig. 2

Let P_p = Power Parameter.

$$P_p = f^{1/4}\left(\frac{W}{b.e}\right)^{3/2} \Big/ \sigma^{1/2}$$

Also $P_p = f\,\sigma\,P_v^3$ by substitution in preceding definitions.

Then T. Hp. $_{req.}/P_p = 0.00000683\,K_v^3 + 0.3329/K_v$

Let T. Hp.$_{req.}/P_p = K_p$ = Power Factor — a number

$$\therefore \quad K_p = 0.00000683\,K_v^3 + 0.3329/K_v \qquad (6)$$

This equation gives the generalized power-required curve for all airplanes, and the parameters P_p and P_v simply determine the scale of the coordinates, vertically and horizontally, respectively.

The Chart, Fig. 2, is provided for the ready solution for P_v, the main performance parameter. Fig. 2 also gives the solution for P_p, and applies to sea-level values only.

We have thus reduced the power-required calculations of an airplane to a very simple basis. The curve of Equation 6

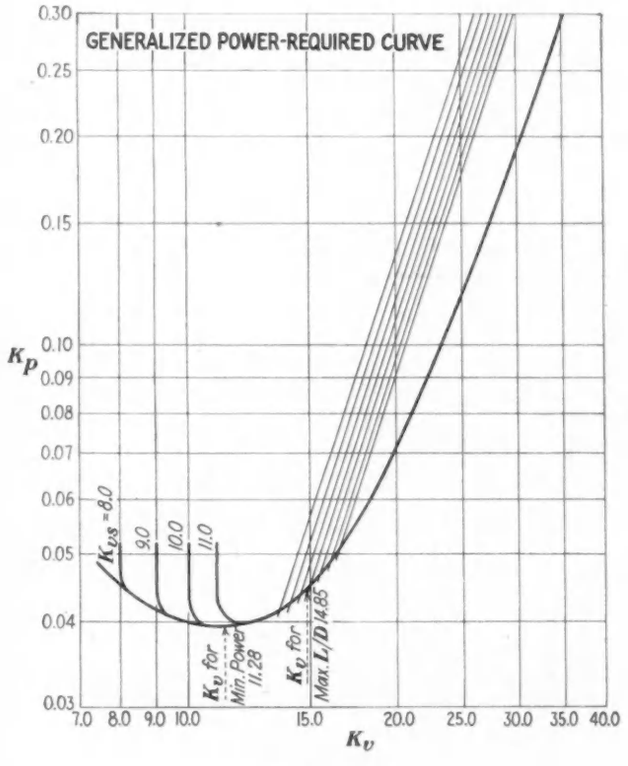


Fig. 3

(plotted logarithmically in Fig. 3) and the two simple parameters completely define the power-required characteristics of any airplane.

No mention has been made previously of the fact that the airplane stalls and that it may not be possible to realize, in some cases, the theoretical minimum power required. Therefore, any performance calculation which fails to take this fact into account may give erroneous results at the very low speeds. If we divide V_{stall} by P_v, we can strike a limit below which our chart does not apply. This calculation has been done for a series of values of K_{v_s} (V_{stall}/P_v), and the limits have been shown in Fig. 3. It will be noted that this limit line has been faired back into the general curve to give the increase in power occurring just prior to stall. Needless to say, the proper

[3] See page 337 "Engineering Aerodynamics," by Walter S. Diehl, The Ronald Press Co., second edition, 1936.

[4] See N.A.C.A. Technical Report No. 350, 1930: "Working Charts for the Selection of Aluminum-Alloy Propellers of a Standard Form to Operate with Various Aircraft Engines and Bodies," by Fred E. Weick.

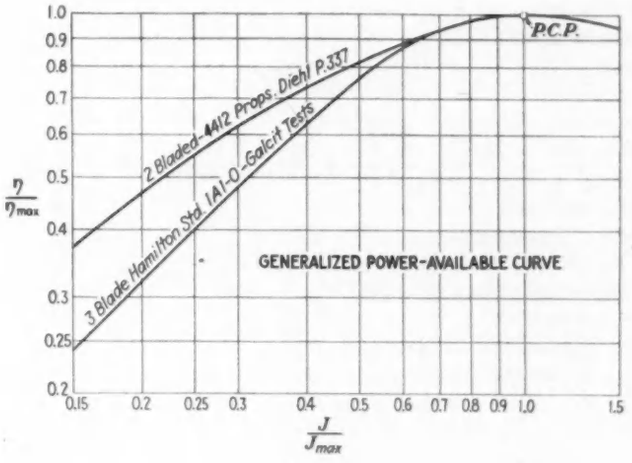


Fig. 4

value of K_{v_s} should be used when computing climb at very low values of K_v. Since P_v varies as $1/\sqrt{\sigma}$ and V_{stall} also varies in the same manner with altitude, the value K_{v_s} is a constant and need not be recalculated for each altitude to be investigated.

Similar to the high angle of attack stall, there is a marked deviation from the theoretical curve of Equation 1 at low values of the lift coefficient. This low-angle burble may cause an appreciable error in high-speed calculations, particularly so if the wing has been set on the body to give least drag at, say, cruising speed. Fig. 1 shows an airplane polar curve plotted C_D vs. C_L^2 which illustrates this phenomenon very well. It is a relatively simple matter to take care of this deviation if we assume that, below a certain definite value of C_L, the drag coefficient becomes constant as shown on Fig. 1.

Let V_{cr} be the velocity at which the deviation from Equation 1 starts as shown on Fig. 1. Then $K_{v_{cr}} = V_{cr}/P_v$.

If a straight line is drawn on Fig. 3, with a slope of 3:1 intersecting the generalized power curve at the value of $K_{v_{cr}}$, we will have a new power curve for low angles of attack. For the determination of high speed, this correction to the power-required curve always should be made. When specific data are lacking as to the value of $C_{L_{cr}}$, it is suggested that:

$$C_{L_{cr}} = 0.3$$

Power Available

Commander Diehl gives a general efficiency curve for constant-speed propellers on page 337 of "Engineering Aerodynamics"[3] revised edition. This figure plots $\eta/\eta_{\max.}$ vs. $V/V_{\max.}$ and may be used directly as one generalized curve of power available if it be plotted logarithmically. Fig. 4 repeats this curve which applies to two-blade propellers, and also gives a plot of such a curve taken from Galcit tests, on the Hamilton Standard three-blade 1A1-0 propeller, which has given excellent results.

For the fixed-pitch propeller no such simple curve can be developed since, as the speed changes, the r.p.m. also changes. In the two references quoted, namely: A.S.I.C. No. 553[1] and N.A.C.A. Technical Report No. 408,[2] the curve of overall propeller efficiency is replaced by an exponential function in the form of $\frac{\eta}{\eta_{max.}} = \left(\frac{V}{V_{max.}}\right)^m$, where the exponent "$m$" a value of approximately 0.5. This same law is assumed to hold here, except that, by a series of calculations based upon N.A.C.A. Technical Report No. 350,[4] the value of "m" has

been determined as a function of the power coefficient, C_p, at maximum efficiency, and also as a function of the manner in which the engine power varies with r.p.m.

It is a well-known fact that b.hp. does not vary, as N in a linear manner for all engines, but rather as N^p, where p is some exponent, easily found by plotting the curve of b.hp. versus r.p.m. logarithmically and determining the slope of the resulting line.

Figs. 5 and 6 give the results of this calculation for the exponent "m," plotted against the power coefficient, C_p, at maximum efficiency; the parameter, p, relates to the exponent of the rate of change of engine power with r.p.m. as just described.

It is to be noted that the curve of overall efficiency (this term takes care of both drop in r.p.m. and propeller efficiency) is a straight line when plotted logarithmically with a slope corresponding to the value of "m" from Figs. 5 and 6.

We still must find a way to place Fig. 4 over Fig. 3 in such manner that they have the proper relative positions for any problem that is to be solved, if we are to use a constant-speed propeller, or to locate the straight line representing the fixed-pitch propeller curve if using that type.

The two fundamental propeller parameters are taken as the power coefficient, C_p, and the advance ratio J $(= V/nD)$. In Fig. 5, we have plotted $\eta_{max.}$ and J for $\eta_{max.}$, both against C_p. These curves apply to a three-blade Hamilton-Standard 1A1-0 Propeller taken from Galcit results. Fig. 6 shows corresponding values for a two-bladed 4412 propeller taken from N.A.C.A. Technical Report 564,[5] and Technical Report 350.[4]

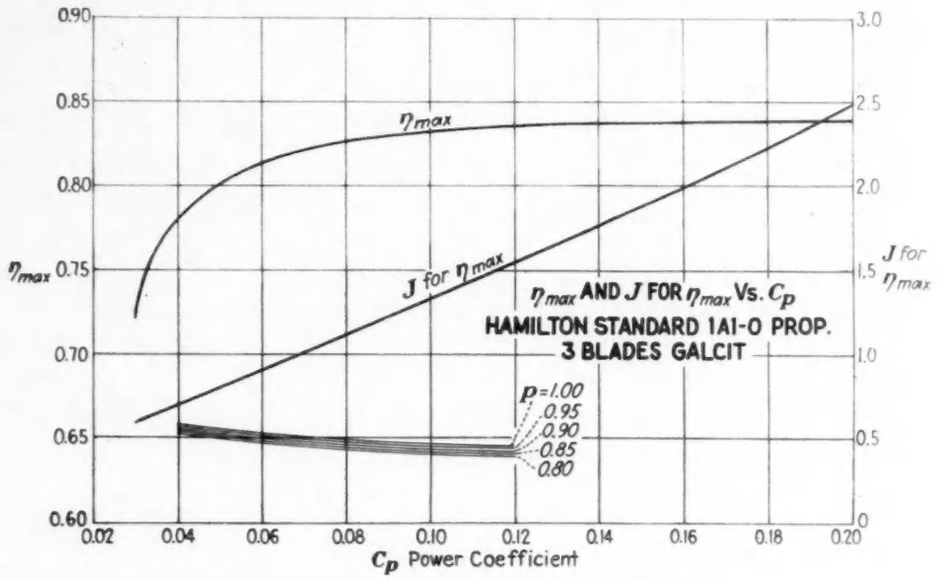


Fig. 5

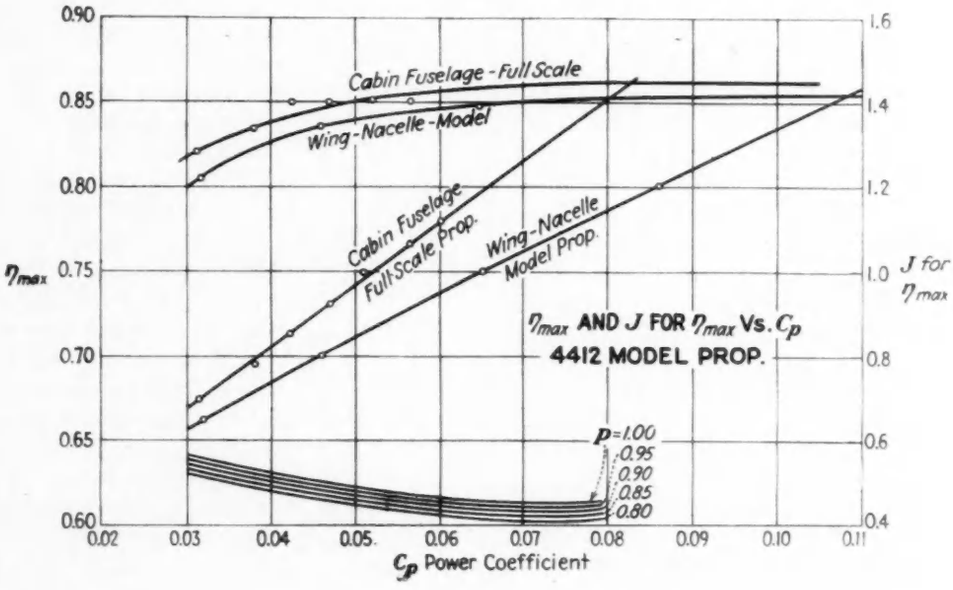


Fig. 6

$$C_p = \frac{\text{B. Hp. x 550}}{\rho\, n^3\, D^5}$$

and

$$C_p = \frac{50.15\ \text{B. Hp.}}{\sigma \left(\frac{N}{1000}\right)^3 D^5} \qquad (8)$$

n = r.p.s.
D = Diameter, ft.
J = v/nD, v in. ft. per sec.
N = r.p.m.
$\therefore$ J = 88 V/ND, V in m.p.h.

$$V = NDJ/88 \qquad (7)$$

After calculation of the power coefficient from Equation (8), we can determine $\eta_{max.}$ and J for $\eta_{max.}$ from Fig. 5 or 6.

Then

$$\text{T. Hp.}_{a_{max.}} = \text{B. Hp.}_{\eta max.}$$

$$V_{T.\,Hp.\,max.} = \frac{ND\,J \text{ at } \eta_{max.}}{88}$$

The small circle on Fig. 4 denoting the position of $\eta_{max.}$ will be called the "Propeller Control Point" and is designated "P.C.P." as shown on this chart.

If we convert T. Hp.$_{a_{max.}}$ and $V_{T.Hp._{a}max.}$ to K_{P_a} and K_v terms by dividing by P_p and P_v, respectively, we can locate the point of $\eta_{max.}$ on the power-required chart of Fig. 3. This procedure gives us the coordinates of the "P.C.P.," as follows:

[5] See N.A.C.A. Technical Report No. 564, 1936: "Tests of a Wing-Nacelle-Propeller Combination at Several Pitch Settings up to 42 Deg.," by Ray Windler.

$$K_{P_{P.C.P.}} = \text{B. Hp.}\eta_{max.} \Big/ P_p \qquad (9)$$

$$K_{V_{P.C.P.}} = \frac{NDJ\eta_{max.}}{P_v} \Big/ \qquad (10)$$

Since Fig. 3 has been plotted logarithmically, Fig. 4 should be also so that multiplication factors due to changes in scale of either of the charts become a matter of addition of a constant representing the log of the multiplying scale factor. Since such scale factors would be constants for each curve in both horizontal and vertical directions, the correct location of *one point* on the K_{P_a} curve with reference to the K_{P_R} curve will locate all points on this curve correctly. This single point we have called the "P.C.P." whose ordinates were just calculated.

A celluloid curve should be made up to fit either one or both of the curves of Fig. 4 plotted logarithmically as shown on Fig. 4. This curve may be placed upon Fig. 3 and the power-available line drawn at once, with the "P.C.P." on Fig. 4 coinciding with that just determined and located upon Fig. 3.

The following outline will give the steps to follow in using this method for constant-speed propellers:

(1) Calculate P_v (Fig. 1, or definition), use $\sigma = 1$.
(2) Calculate P_p (Fig. 1, or definition), use $\sigma = 1$.
(3) Calculate C_p.
(4) Find $\eta_{max.}$ *and* $J\eta_{max.}$ from Fig. 5 or 6.
(5) Calculate P.C.P. ordinates Equations 9 and 10.

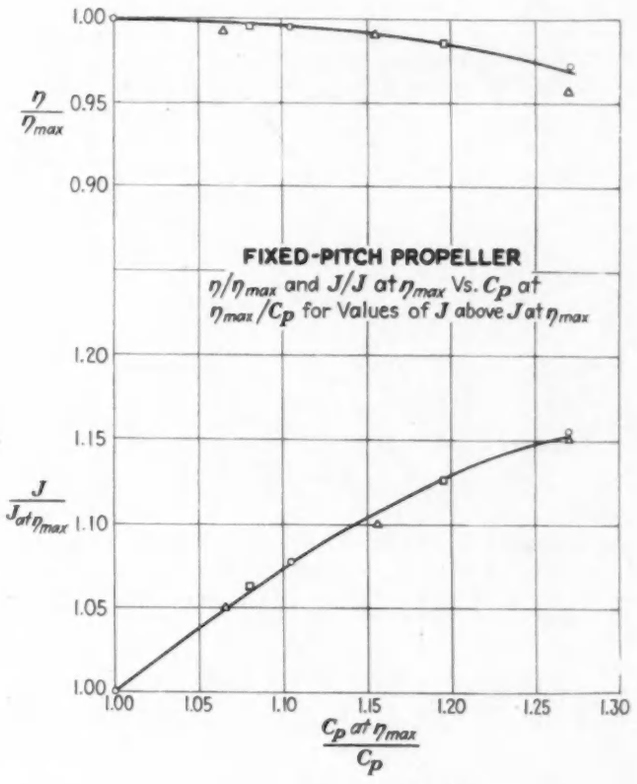


Fig. 7

Altitude	σ	$1/\sqrt{\sigma}$	σ_m	P_v	P_p	C_p	$\eta_{max.}$	$J\eta_{max.}$	P. C. P. K_{P_a}	P. C. P. K_v

Note:

(a) P_v at altitude $= \sqrt{1/\sigma}\, P_v$ (at ground).
(b) P_p at altitude $= \sqrt{1/\sigma}\, P_p$ (at ground).
(c) C_p at altitude $= 1/\sigma\, C_p$ (at ground) up to critical altitude.
(d) C_p at altitude $= \sigma_m/\sigma C_p$ (at ground) above critical altitude.

σ_m = relative B. Hp. at any altitude above critical altitude.

(*Continue on opposite column*)

$$\sigma_m = \frac{\frac{\sigma}{\sigma_{cr}} - 0.117}{0.883} \qquad (11)$$

σ_{cr} = relative density at critical altitude.

(6) Locate the required series of "P.C.P."-s on Fig. 3, noting thereon the altitudes at which they apply.

(7) Locate Fig. 4 (made up in celluloid) over Fig. 3 with the circle of Fig. 4 located over the various "P.C.P."-s in turn.

(8) Fill out the following table: .

Altitude	P_v	P_p	$K_{v\,max.}$	$V_{max.}$	$(K_{pa}-K_{pr})_{max.}$	Rate of Climb	Climb K_v	Climb V

Note:

(a) $V_{max.} = K_{v\,max.} P_v$

(b) Rate of climb $= (33000/W)P_p\,(K_{pa} - K_{PR})_{max.}$

(Due to logarithmic plotting the actual values of K_{PR} and K_{pa} must be used and subtracted rather than transferring the difference measured by dividers to left hand origin.)

(c) K_v (climb) occurs where $K_{pa} - K_{PR}$ is a maximum.

The fixed-pitch propeller introduces more complication than does the constant-speed type, but it is still susceptible to relatively simple solutions. Some blade angle must be chosen that will correspond to some fixed condition and give a maximum value for η at some value of J. This setting corresponds to one *particular* condition of the constant-speed propeller, which may be at any altitude, but most likely will be at the critical if the engine is supercharged.

Fig. 7 is a plot of $(C_p$ at $\eta_{max.})/C_p$ vs. $J/J\eta_{max.}$ for

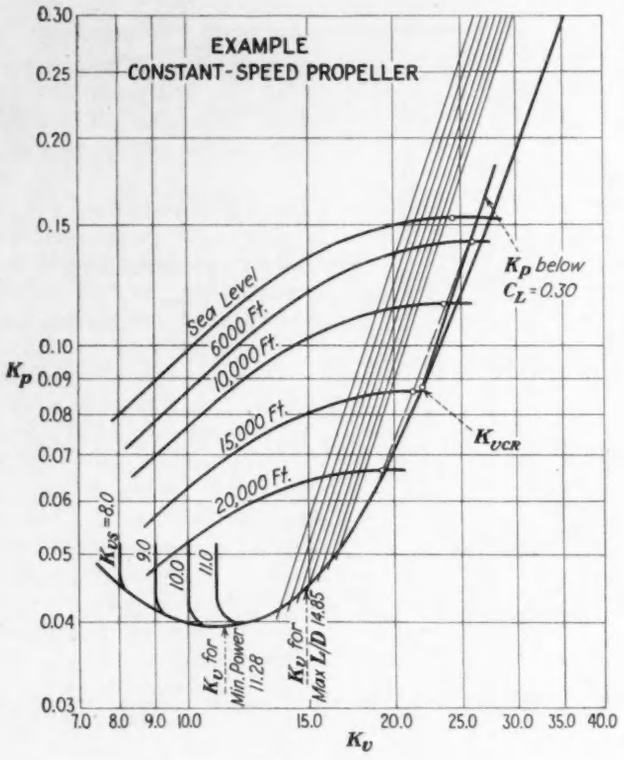


Fig. 8

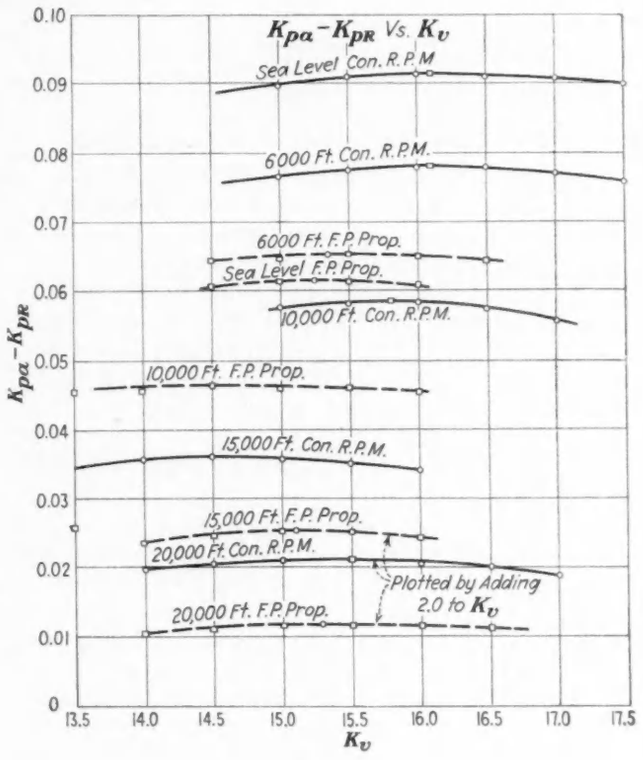


Fig. 9

both two- and three-blade propellers at constant blade angle. A curve of $\eta/\eta_{max.}$ also is supplied.

Let $C_{p_1} = C_{p_{\eta max.}}$ = Power coefficient at chosen condition and blade angle.

C_{p_2} = Power coefficient at any other condition at *full design r.p.m.* but at chosen blade angle.

From Fig. 7 we may then find $J_2/J_{\eta max.}$ and $\eta_2/\eta_{max.}$ since they are both plotted versus C_{p_1}/C_{p_2}.

Since the r.p.m. taken for C_{p_2} is the maximum value permissible, and is the same as at C_{p_1}, the value of V corresponding to C_{p_2} follows immediately from J_2. Likewise η_2 is found quickly.

The table following illustrates this calculation:

Illustrative Examples

Constant-Speed Propellers – An example will serve to clarify the whole process:

Let:

W = 14,800 lb.
D = 11.5 ft., three blade
B. Hp. = 820 (each of 2) below critical altitude.
N = 1444 r.p.m.
C_{D_o} = 0.028 (Fig. 1)
L = 0.0661 (Fig. 1)
Critical Altitude = 6000 ft.
b = 70.5 ft.
$C_{L_{cr.}}$ = 0.30 (Fig. 1)
S_w = 682 sq. ft.

C_{p_1} = . $J_{\eta max.}$ = $\eta_{max.}$ = m =

Altitude	σ	$1/\sqrt{\sigma}$	P_v	P_p	σ_m	C_{p_2}	$\frac{C_{p_1}}{C_{p_2}}$	$\frac{J_2}{J_{\eta max.}}$	$\eta/\eta_{max.}$	J_2	η	P. C. P. K_{pa}	P. C. P. K_v

When the coordinates of the control points have been determined and located on Fig. 3, straight lines are drawn through them with the slope "m," and a table similar to that used for the constant-speed propeller is filled out.

If the P.C.P. falls to the left of the curve, Fig. 3, the propeller will turn more than rated r.p.m. at maximum velocity. If this speed is not permissible, a line should be drawn *downward* and to the right with a slope of ½ to intersect the power-required curve to take care of the throttling of the engine at speeds above the design r.p.m.

$$e = \frac{1}{\sqrt{\pi \times 7.29 \times 0.0661}}$$
$$= 0.813$$
$$= 0.028 \times 682$$
$$= 19.1$$

Then $P_v = \left(\frac{W}{b.e}\right)^{1/2} \Big/ f^{1/4}$ at ground

$$= (14{,}800/0.813 \times 70.5)^{1/2} \Big/ 19.1^{1/4} = 16.07/2.09 = 7.70$$

$P_p = 7.70^3 \times 19.1 = 8730$ at ground.

$C_p = \dfrac{50.15 \times 820}{1.444^3 \times 11.5^5} = 0.0679$ at ground

$\eta_{max.} = 0.819$ (Fig. 5)

$J\eta_{max.} = 0.99$ (Fig. 5)

Ground $\begin{cases} K_{p_a}\text{ (P.C.P.)} = 2\times820\times0.819/8730 = 0.154 \\ K_v\text{ (P.C.P.)} = (1444\times11.5\times0.99)/(88\times7.70) = 24.24 \end{cases}$

$K_{v_{cr.}} = \dfrac{168.4}{7.70} = 21.9$ at $C_{L_{cr.}} = 0.30$

Altitude	σ	$1/\sqrt{\sigma}$	P_v	P_p	C_p	σ_m	$\eta_{max.}$	$J\eta_{max.}$	P. C. P. K_{pa}	P. C. P. K_v
0	1.00	1.00	7.70	8,730	.0679	1.00	.819	0.99	0.154	24.24
6,000	0.8358	1.0938	8.41	9,550	.0812	1.00	.827	1.135	0.142	25.43
10,000	0.7384	1.1637	8.95	10,155	.0798	0.868	.826	1.115	0.1155	23.51
15,000	0.6291	1.2608	9.71	11,010	.0777	0.720	.825	1.09	0.0862	21.2
20,000	0.5327	1.3701	10.55	11,960	.0756	0.590	.824	1.07	0.0666	19.19

In order to illustrate this problem, Fig. 8 has been plotted, which shows the generalized power-required and power-available curves plotted one over the other. Each power-available curve for the preceding altitudes has been drawn in completely, and labeled properly.

It is to be noted that the "P.C.P." travels in straight lines; one line sloping downward and to the right up to critical altitude, but above critical the trace of the point moves downward and to the left.

Filling out the following table from Fig. 8, we have:

In order to find the maximum value of $K_{pa}-K_{pR}$ the various values read from Fig. 8 are plotted on Fig. 9 from which the above table is filled out.

$$\text{Rate of Climb} = 2.23\ P_p\ (K_{pa}-K_{pR})$$

These results are plotted on Fig. 10.

Fixed-Pitch Propeller – In order to illustrate the solution for a fixed-pitch propeller, the foregoing problem is repeated.

The propeller is assumed to be designed for the critical altitude and the proper blade angle chosen to give the follow-

Altitude	P_v	P_p	$K_{v\,max.}$	$V_{max.}$	$K_{v_{CL.}}$	V_{CL}	$K_{pa} - K_{pR}$	Rate of Climb
0	7.70	8,730	27.0	208.0	16.1	124.0	0.0913	1947
6,000	8.41	9,950	26.50	223.0	16.1	135.5	0.078	1730
10,000	8.95	10,155	24.50	219.0	15.8	141.5	0.0585	1325
15,000	9.71	11,010	21.90	213.0	14.70	143	0.0366	898
20,000	10.55	11,960	19.40	205.0	13.5	142.3	0.0210	558

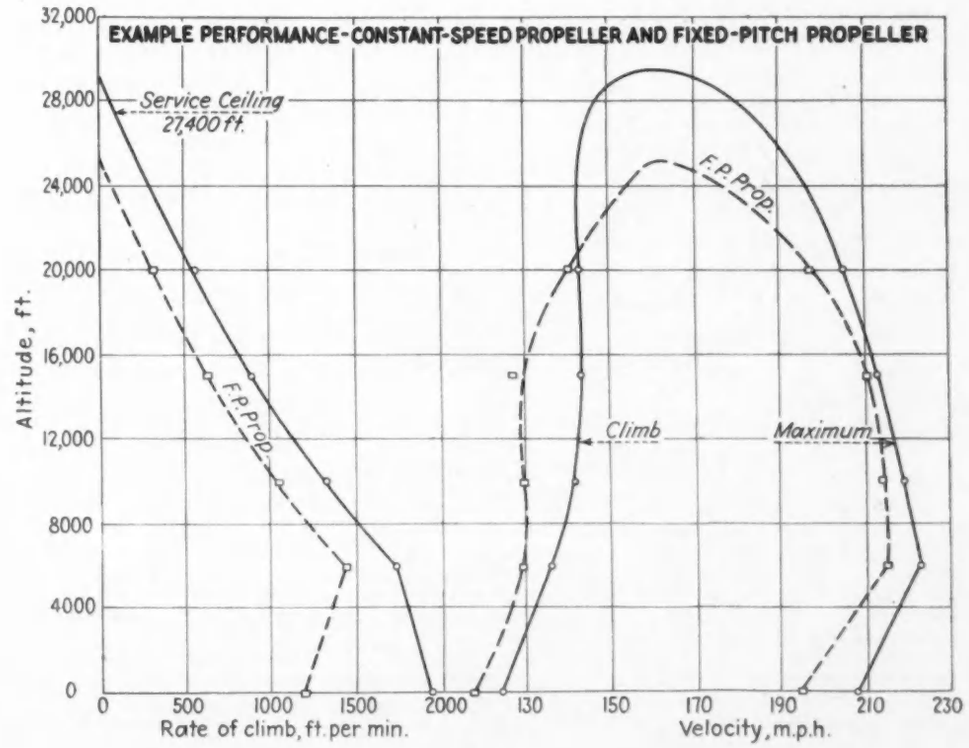


Fig. 10

ing values:

$C_{p_1} = 0.0812$ η max. $= 0.827$ $m = 0.48$ $J_{\eta\,max.} = 1.135$

When using the throttling curves this table will be found more convenient than trying to read Fig. 3.

Altitude	P_v	P_p	C_{p_2}	$\frac{C_{p_1}}{C_{p_2}}$	$\frac{J_2}{J_{\eta\,max.}}$	J_2	$\eta/\eta_{max.}$	η	P. C. P. K_{pa}	P. C. P. K_v
0	7.70	8,730	0.0679	1.196	1.127	1.28	0.985	0.815	0.154	31.35
6,000	8.41	9,550	0.0812	1.00	1.00	1.135	1.00	0.827	0.142	25.43
10,000	8.95	10,155	0.0798	1.018	1.014	1.151	0.999	0.826	0.1155	24.52
15,000	9.71	11,010	0.0777	1.045	1.034	1.162	0.998	0.825	0.0862	22.60
20,000	10.55	11,960	0.0756	1.075	1.055	1.197	0.997	0.824	0.0666	21.48

Through the coordinates of the control points straight lines with slope of 0.48 are drawn. The following table is filled out from Fig. 11.

We are interested in finding the greatest possible number of miles per pound of combustibles at any given weight which may be integrated for maximum range.

Altitude	P_v	P_p	$K_{v\,max.}$	$V_{max.}$	$K_{v_{CL}}$	V_{CL}	$K_{pa} - K_{pR}$	Rate of Climb
0	7.70	8,730	25.3	194.7	15.25	117.5	0.0614	1195
6,000	8.41	9,950	25.6	215.2	15.35	129.1	0.0652	1445
10,000	8.95	10,155	23.9	214	14.50	129.8	0.0463	1050
15,000	9.71	11,010	21.7	211	13.10	127.2	0.0253	621
20,000	10.55	11,960	18.7	197.3	13.28	140.1	0.0118	315

These results are also plotted on Fig. 10 for comparison.

Range Solution

Problems covering range also can be solved by use of this logarithmic chart. It will be noted that a series of straight lines have been drawn on Fig. 3 sloping upward and to the right through a series of definite values of K_v. These lines represent curves at constant C_p, J, η and blade angle θ.

Since for constant $J\,(=V/nD)$ the velocity must vary linearly with the r.p.m., and since for constant-power coefficient, C_p, the B.Hp. must vary as N^3, then the B.Hp. must vary as V^3 as the engine is throttled. Since for a constant blade angle, constant J and constant C_p the propeller efficiency is also constant, the thrust power available will also vary as V^3.

The straight lines just mentioned have a slope of 3:1 on the logarithmic plot of Fig. 3; that is, each line may become the trace of any point on a power-available curve when the engine is throttled in cruising.

The values of K_v through which the throttling curves are drawn are given in the following table, with the corresponding value of K_p and the ratio of K_v/K_p:

K_v	K_p	K_v/K_p
13	.0406	320
13.5	.0414	326.0
14	.0425	329.6
14.5	.0438	331.0
15	.0453	331.0
15.5	.0469	330.5
16	.0488	328
16.5	.0509	324
17	.05306	320.4
17.5	.0556	314.9
18	.0583	309.0
18.5	.0611	302.9
19.0	.0643	295.7
19.5	.0677	288.0
20.0	.0713	280.6

It may be proved that:

$$\text{miles per lb.} = K_v/K_p \times \eta/C \times P_v/P_p \qquad (12)$$

K_v/K_p is given in this table for each cubic.

P_v/P_p can be found from the airplane parameter and is constant for any gross weight.

η_{ic} is found for each cubic by the intersection of that cubic with the curve of Fig. 4 placed over Fig. 3 with P.C.P. in proper place for the weight to be investigated. A value of K_{pc} is read at this intersection.

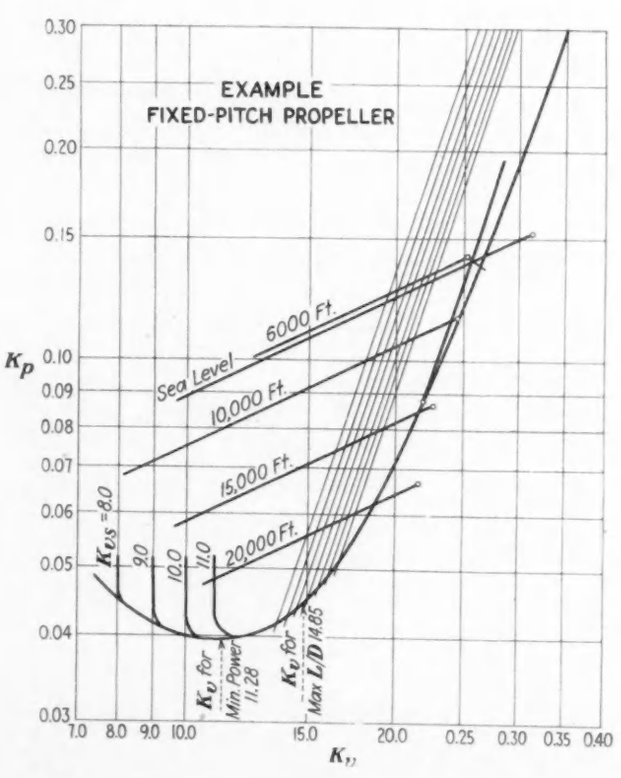


Fig. 11

Then:

$$\eta_c = \eta\, max. \frac{K_{pc}}{K_p\ (\text{P.C.P.})} \qquad (13)$$

The specific fuel and oil consumption is given by the engine manufacturer for the engine used. The following table, made out for each weight, varying by even increments from initial load to final weight, will serve to clarify the procedure:

K_v	K_v/K_p	K_{pc}	η	B. Hp.	N (R.P.M.)	C (Lb. per B.hp.hr.)	$\frac{P_v \times K_v \times \eta}{P_p K_p c}$	$V_{cru.}$
13.5	326 0							
14	329.6							
14.5	331							
15	331							
15.5	330.5							
16	328							
16.5	324							

Plot the results in the tables as miles per lb. vs. $V_{cru.}$ and determine a maximum, and the cruising speed at which it occurs. Then repeat for a series of gross weights and integrate for maximum range.

Note:

P_v varies as $W^{1/2}$ B. Hp. $= \dfrac{P_p K_p}{\eta}$

P_p varies as $W^{3/2}$ $N = N_{max.} \left(\dfrac{\text{B. Hp.}}{\text{B. Hp. } max.}\right)^{1/3}$

$V_{CR} = K_v P_v$

The coordinates of the P.C.P. vary as follows:

K_v varies as $\dfrac{1}{W^{1/2}}$ since C_p, $J_{\eta\, max.}$ and $\eta\, max.$ are all constant with weight.

K_p varies as $\dfrac{1}{W^{3/2}}$

With B.Hp. and N determined, C is found from the engine manufacturer's data.

The trace of the P.C.P. will be a straight line on logarithmic paper with the same slope as the throttling cubics. This line will slope upward and to the right. Since the trace of the P.C.P. as the weight decreases is parallel to the throttling cubics, the first four columns in the preceding table will be identical for each weight.

No example shall be worked out for the range solution, since it is believed the method can be followed without it.

Discussion

Considers Methods only as First Approximations

– David Biermann

Assistant Aeronautical Engineer, National Advisory Committee for Aeronautics

COMMENTS on Mr. Driggs' paper follow:

(1) This particular method is basically similar to several other methods such as developed by Oswald, Rockefeller, and so on; but the form that it has taken is such that it is simple and compact. In addition, Driggs seems to have improved the accuracy at both ends of the speed range (marginal performance computations).

(2) Simplifying and generalizing almost inevitably reduce the accuracy. In this method, generalized power-required and power-available curves are derived, both of which are probably good approximations, but they cannot be considered highly accurate. We know from experience that there are a great number of variables in the design of a propeller, for example, which affect its performance. A generalized curve necessarily must be based on one propeller design and one body style, which is justifiable for certain types of computations such as preliminary performance. The generalized power-required curves neglect Reynolds Number, which must be taken into account separately. Such items as change of rivet drag with speed and compressibility effects do not follow the square law, so separate corrections must be made if they are judged to be of sizeable proportions.

(3) It is not the intention to show that such methods are worthless by pointing out some of the apparent limitations and sources of error, but merely to show that the methods can be considered only as first approximations. A neglect of secondary effects is justifiable if, by so doing, the approximate answer is arrived at quickly; or, if the effect of changing one variable at a time is desired. Simplified methods are powerful tools in the hands of designers who must find the right combination of variables in a reasonable time.

(4) Mr. Driggs is to be congratulated on the clever method that he has developed. Airplane performance computations have almost been reduced to slide-rule simplicity. After all, a slide rule gives only the approximate answer, but the advantage of speed more than offsets any inaccuracy.

Amplification of Paper Desired

– Elliott G. Reid

Stanford University

ALTHOUGH it would be premature to comment upon the accuracy and convenience of the new method before gaining considerable experience in its use, Mr. Driggs' accomplishment in transforming the cumbersome equation for power requirements into a conveniently generalized one which involves only non-dimensional coefficients, sheds a new and welcome light upon a problem which, of late, has become tremendously complicated. It is anticipated that this development ultimately will be recognized as having an importance in its field fully comparable with that of the non-dimensional analysis of dynamic stability developed some years ago by Glauert.

If any criticism of Mr. Driggs' work is justifiable, it is that his paper has been condensed to the point of almost obscuring the significance of some of its essential details. For example, the constancy at all altitudes of $\sigma^{3/2}$ T. Hp. (the "indicated" power required) is not even mentioned and the meaning of the minimum value of K_p (which may not be evident to those not familiar with Schrenk's analysis) is not stated. It is hoped, therefore, that the original text may be amplified before publication or that the author will contribute somewhat extensively to the discussion.

Author Answers Discussers

– Ivan H. Driggs

The Glenn L. Martin Co.

IN answer to Mr. Biermann's comments, I present the following:

(1) It is realized that this scheme is basically similar to that previously developed by other writers, but it is believed that some simplification and clarification results from this method of treatment.

(2) Mr. Biermann's comments regarding the effect of Reynolds Number and roughness were not considered in the development of this paper, since it was not desired to enter into a discussion of the calculation of the two airplane factors "*f*" and "*e*." Both of these quantities naturally will be affected by both Reynolds Number and roughness, and there is no reason whatever why these corrections should not be considered when originally determining the airplane polar. If wind-tunnel results obtained at low Reynolds Number are used with the calculation performance by this method, the same errors will result as for any other procedure. It is rather difficult to see why the process of generalization of the power-

required curve can be considered only a rough approximation. If the airplane performance is to be calculated, *some* power-required curve must be obtained, which presumably will take into account the effects of Reynolds Number, roughness, protuberances, cooling drag, and so on. This power-required curve must be based upon *some* polar from which coefficients "*f*" and "*e*" may be determined by the methods outlined. An approximation exists only in the degree of accuracy with which the mathematical expression of Equation 3 with its corrections at both ends of the speed range can be made to fit the correct curve. It will be found that the two curves will fit with a precision greater than the accuracy for flight-test results.

The generalized propeller curves were included in this paper for illustrative purposes only. There is no reason why the designer, wishing to take into account the effects of tip speed, blade width, blade twist and blade section, may not use any particular curve that he desires. The generality of the whole scheme is in no way impaired by so doing.

(3) It was not the intention of this paper to introduce so much simplification that accuracy would be lost, but rather to arrange the method so that accuracy could be gained by making it possible to account for all the factors affecting airplane performance in a relatively simple manner. There is basically no difference between the generalized logarithmic plotting developed in this paper and the normal calculation in plotting of power-required and available curves for a series of weights and altitudes.

(4) Mr. Biermann's remarks are much appreciated, and it is sincerely hoped that he, as well as other designers, will find the paper very useful.

Replying to Mr. Reid I would like to say that it is experienced at The Glenn L. Martin Company, where this method of calculating airplane performance has now replaced all other schemes for performance calculation, that a great deal of saving in time has resulted. Probably a far greater advantage of this generalized method lies in the fact that all the factors affecting airplane performance are given their true relative values in the two performance parameters which promotes a more fundamental and a better understanding of the whole subject of airplane performance.

The writer is sorry indeed that the press of work in a commercial organization did not permit a greater explanation and discussion regarding the meaning of some of the various terms. It is believed, however, that the references given in the paper, plus the work by "Schrenk," supply a discussion upon the points mentioned. There are, of course, many applications of this generalized formula which the writer has worked out, but which were not included, for the previous reason.

It is hoped, at a later date, that an opportunity will present itself for further amplification of this whole method.

Operating Costs of Planes, Cars, and Boats

IN order to obtain some comparison of the operating costs of airplanes now available to the private pilot with those of other forms of private vehicles, the table below has been made showing the relative costs of operation of the airplane, the automobile, and the motor boat, giving as much detail as seemed desirable for such a comparison. It will be evident that, to compile such a table, it is necessary to choose certain factors, such as hours of use, average speed, reasonable costs for insurance, depreciation, and other such factors which are involved in an estimate of operating figures. It also will be evident that some relationship must be established concerning the character of service to which these conveyances are to be put. The airplane, the automobile, and the motor boat are used normally for different purposes and are, therefore, in a sense, not directly comparable. Bearing this point in mind, the use of the airplane has been considered as a means of transportation from point to point on trips of 50 miles and more. The use of the automobile has been considered to be its operation in suburban districts for shopping and local transportation including trips of 40 to 50 miles. The motor boat has been considered as operating in harbors and on short cruises along the coast. The types of equipment chosen for the comparison include a typical small cabin plane (Monocoupe 90 hp.), a larger cabin plane (Stinson 245 hp.), a relatively small automobile (Ford 85-hp. Coupe), a larger automobile (Cadillac 140 hp.). The motor boat is a typical small cabin cruiser (Elco 35 ft.).

Such a table undoubtedly invites argument. It is significant, however, that, even though the figures of individual items may be changed, the effect on unit mileage cost will be relatively small.

Excerpts from the paper: "Private Flying," by Luis de Florez, presented at the National Aeronautic Meeting of the Society, Washington, D. C., Mar. 10, 1938.

COMPARATIVE OPERATING COST OF 2 AIRPLANES: 2 AUTOMOBILES and 1 MOTOR BOAT

	AIRPLANES		AUTOMOBILES		MOTOR BOAT
	450 hrs. flying per year		450 hrs. driving per year at average 25 m.p.h.		150 hrs. per year at average 12 m.p.h.
	112 m.p.h. Monocoupe	125 m.p.h. Stinson	Ford Coupe	Cadillac #75	Elco 35 ft.
Passenger Capacity	2	4 (5)	2 (4)	5	10
Price (delivered)	$3,884.00	$9,000.00	$748.00	$3,290.00	$6,000.00
Miles per year	50,400	56,250	11,250	11,250	1800
COST OF OWNERSHIP & OPERATION					
Depreciation (25%)	$971.00	$1,800.00	$187.00	$822.00	(15%) $900.00
All Insurance	325.00	548.00	86.00	145.00	160.00
Total Fixed Charges	1,296.00	2,348.00	273.00	967.00	1,060.00
Gasoline	470.00	1,485.00	127.00	225.00	144.00
Oil	113.00	158.00	15.00	22.00	30.00
Storage and Cleaning	365.00	550.00	180.00	360.00	500.00
Repairs and Misc.	290.00	650.00	110:00	220.00	250.00
Total Cost of Operation	1,238.00	2,843.00	432.00	827.00	924.00
GRAND TOTAL	2,534.00	5,191.00	705.00	1,794.00	1,984.00
Cost in $ per hour	5.63	11.53	1.57	3.98	13.23
Cost in $ per mile	050	092	.063	.159	1.102
Cost per passenger mile (based on capacity)	.025	.023 (.018)	.031(.015)	.032	.110

Air Conditioning of Automobiles and Buses

By L. W. Child
Evans Products Co.

ALL factors necessary for year-around air conditioning of cars and buses are covered generally in this paper. How the desired results were obtained in both winter and summer air conditioning is explained with the aid of a chart of air requirements.

Types of equipment are discussed, especially the refrigerating system, giving powers, capacities, and safety factors.

THE term "air conditioning" probably had its first official use in Willis Carrier's original paper entitled "Rational Psychrometric Formulae" presented at the annual meeting of the A.S.M.E. in 1911.[1] The first sentence of this paper is as follows: "A specialized engineering field has developed recently, technically known as air conditioning, or the artificial regulation of atmospheric moisture." This sentence is quoted to give an idea of just what was originally meant by "air conditioning," and what constituted its primary meaning up to a few years ago. Even today, the regulation of the moisture content of the air is the first prerequisite for a genuine air-conditioning job. It matters not how this regulation of moisture content is accomplished, whether by heating, cooling or humidifying, it must be controlled to classify as air conditioning.

Unfortunately, the term air conditioning must have appealed to some advertising or publicity man, with the net result that, for several years, air conditioning was used to cover anything remotely connected with air. This state still appears to exist in the minds of some advertising men. The general public, however, has come to regard air conditioning as synonymous with cooling, and to classify any system without refrigeration as heating, ventilation, or humidification. The Heating & Ventilating Engineers, in an attempt to arrive at a safe definition, say: "Air conditioning is the science of controlling the temperature, humidity, motion, and cleanliness of the air within an enclosure." This last definition, although it does not state to what limits these factors must be controlled, appears to be the most satisfactory definition of air conditioning.

However, irrespective of terminology, we are interested in producing conditions inside a vehicle that will give satisfaction to the majority of the customers. If this purpose can be accomplished at a price within the range of the majority for eight out of the twelve months of the year, it is certainly justified, no matter what it is called. That a market for a complete year-around automotive conditioner is available at present cannot be doubted but, when price is considered, it is obvious that a far greater market exists for winter conditioning systems. As far as the results in the car are concerned, both systems are figured by identically the same method – both control humidity, temperature, and cleanliness of the air.

The primary problem in any conditioning system is the maintaining of the desired amount of water vapor in the air. The use of a psychrometric chart greatly simplifies the calculations in such a problem, and it is recommended that anyone faced with this type of work familiarize himself with one form of this chart. However, as the intent of this paper is to give an idea of what problems confront the engineer in applying air conditioning to a vehicle, rather than to serve as an advanced course in psychrometry, the discussion of phenomena met in this branch of engineering will be limited to a few necessary definitions.

In all problems of air conditioning the engineer is dealing with a mixture of a vapor and a gas. As a simple definition, we can consider a vapor to be a substance in gaseous form that normally exists as a liquid. As an example, gasoline is normally a liquid, but gasoline vapor is, in reality, a gas. Likewise, water is normally considered a liquid, whereas water vapor is a gas, commonly called steam. The laws governing the action of a mixture of air and water vapor are identical with those governing a mixture of air and gasoline vapor. The problems encountered with water vapor, however, are greatly simplified because we are working with a single, simple vapor whose properties are known better than those of any other vapor.

Dry Air Defined

Air, in the conditioning field, commonly denotes atmospheric air with its accompanying water vapor. When air is spoken of as a gas, without the presence of vapor, it is termed dry air. In all cases the adjective, dry, denotes the absence of water in any form, and is not to be confused with its common meaning, the absence of liquid water.

Since dry air and water vapor form a true mixture, each component acts as if the other were not present. The water vapor exerts its own pressure and, like any other steam, it will condense if cooled below its saturation temperature. Also, the more water vapor present, the more pressure it will exert and, since the saturation temperature of steam increases with the pressure, the temperature at which the water starts to condense out of atmospheric air must be a measure of the actual amount of vapor present. Since water condensed from air is commonly called dew, the temperature at which it condenses is called the dew point of the air. Also, since air cooled to its dew point contains the greatest amount of water vapor possible for that temperature, it is said to be saturated. From this reasoning it is evident that air at a temperature higher than its dew point does not contain as much water vapor as possible. If, for example, it contains half as much as it could at this higher temperature, it is said to have a relative humidity of 50 per cent.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1938.]

[1] See A.S.M.E. Transactions, Vol. 33, 1911, pp. 1005-1039: "Rational Psychrometric Formulae," by Willis H. Carrier.

This term is not to be confused with absolute humidity, which is the actual weight of water vapor per pound of air. As this net weight is extremely small, absolute humidity is generally expressed in grains of water vapor per pound of air rather than in pounds per pound. Since there are 7000 grains in a pound, this method enables us to express absolute humidity in whole numbers.

The conditions of existing air are determined generally by the use of a wet-bulb thermometer in conjunction with a common, or dry-bulb, thermometer. Because the wet-bulb temperature is a measure of the heat content of the air, it will be found most useful in calculations.

The application of air-conditioning principles to an automobile or bus calls for somewhat different considerations than the problem of conditioning a home or public building. The space per person, especially in the bus, is considerably smaller than that found in the most crowded of buildings. The air for conditioning must be delivered in close proximity to the passengers, giving little chance for mixing it with the surrounding air before coming in contact with the occupants. Unlike the building, the car is moving at widely varying speeds causing correspondingly widely varying amounts of fresh air to leak into the car and so upset any carefully figured conditions. There is also the serious problem of available space for equipment.

Consider the relatively crowded condition in a five-passenger car as compared to a crowded restaurant. A restaurant 60 ft. x 40 ft. x 10 ft. high might, by dint of extreme squeezing, seat 200 persons. Each individual would then have 120 cu. ft. of space all to himself. Taking the passenger space in a five-passenger car as 150 cu. ft., the space per passenger is only 30 cu. ft. To equal this condition, the restaurant would have to pack in 800 patrons which, under their original table plan, would call for stacking them into four layers one on top of the other. Now imagine these 800 patrons all smoking black cigars, and you will have some idea of the ventilating problem presented by an everyday automobile. Obviously such a condition calls for a first-class deodorizing plant, or plenty of fresh air; and the fresh air will be found to give the most economical solution.

Cold Drafts Avoided

A cold draft will do more to ruin a conditioning job than almost any other factor. In a building, the supply outlets are located carefully to obviate this trouble, whether the system is used for heating or for cooling, by placing them far enough away to allow the air to slow down before coming in contact with the occupants. As this arrangement is impossible in a car, it is necessary to direct the air carefully away from the occupants, and supply it at the highest possible temperature and the lowest possible velocity, whether the system be for heating or cooling. In a bus this purpose is accomplished best by distributing the air through duct work, or by using a series of individual supply units.

The leakage of outside air into a car or bus would seem to be an uncontrollable factor. It varies with the speed, the wind velocity and direction, the type of construction, and the age of the vehicle. The most logical way to attempt to control leakage is to supply enough outside air, properly conditioned, to cause the air in the body to leak out rather than in. Although this method appears to be impractical on very leaky cars, it is satisfactory on those produced within the last few years, and is a distinct improvement in any vehicle, even though insufficient air is delivered to stop all inward leaks. Rough tests have shown that 60 to 80 cu. ft. of air per min. delivered from outside into the body will stop leaks on most of today's four-door cars at speeds up to 60 m.p.h.

Considering the atmosphere inside the car, the most annoying phenomenon of winter driving is the formation of fog on the windows and the odor of wet woolen clothes and stale tobacco smoke. If we supply sufficient fresh air to eliminate the possibility of fog, experience has shown that we also will eliminate the odor and smoke, stale or otherwise. Fogging of windows is caused by increased amounts of water vapor in the air due to that given off by the passengers. This fogging can be eliminated only by keeping the quantity of water vapor in the air low enough to prevent condensation at the temperature of the glass. Technically, this condition means that the dew point of the air must be held lower than the temperature of the inside surface of the glass. If the quantity of water vapor given off by each passenger and the temperature of the glass for a given outside condition are known, the problem of determining the minimum volume of fresh air required per passenger is relatively simple.

A normal adult can be expected to give off 0.075 lb. of water vapor per hr., or 8.75 grains per min., at temperatures below 70 deg. fahr. Although this figure is bound to vary, it has served satisfactorily in practice.

At speeds up to 60 m.p.h. rough tests show that the minimum temperature of the inside surface of the laminated glass in a car to be two-tenths of the way between the outside and inside temperatures. For example, with an inside temperature of 70 deg. fahr. and 0 deg. fahr. outside, the glass will be at 14 deg. fahr. This value will vary with wind direction, speed, car construction, and type of glass but, to date, it appears to be a safe figure.

Assume an outside condition of 0 deg. fahr. and 100 per cent relative humidity with the car at 70 deg. fahr. and windows at 14 deg. fahr., inside. As the passengers are the only source of water vapor inside the car, the minimum volume of air required to eliminate fogging can be figured per passenger, the result being multiplied by the number of passengers. The problem, therefore, involves determining how much air can be raised from a 0 deg. fahr. dew point to a 14 deg. fahr. dew point with the addition of 8.75 grains of water vapor. From a psychrometric chart or table, it will be found that a pound of air at 14 deg. fahr. dew point contains 11.2 grains of vapor, whereas 0 deg. fahr. dew point air contains 5.5 grains. Therefore, if we add 11.2 minus 5.5, or 5.7 grains of vapor to one pound of 0 deg. fahr. dew point air, the result will be, to all intents and purposes, 14 deg. fahr. dew point air. Having the quantity of vapor one pound of air will absorb, the number of pounds required to absorb 8.75 grains will be:

$$\frac{8.75}{5.7} = 1.53 \text{ lb. of 0 deg. fahr air}$$

Therefore, under the conditions specified, the minimum quantity of outside air to introduce per passenger is 1.53 lb. per min. To reduce this to cu. ft. per minute we again refer to the psychrometric chart and find one pound of 0 deg. fahr. air contains 11.6 cu. ft., making the volume required 11.6 times 1.53, or 17.8 cu. ft. per min. If the car contains five passengers, approximately 90 cu. ft. per min. will be required.

Fig. 1 shows the minimum volume of fresh air required per passenger to eliminate fogging for different window temperatures with various outside conditions. It will be noted that the curves are for outside dew points, not dry-bulb temperatures. However, as the worst conditions prevail when the outside air is saturated and the dry-bulb temperature approaches the dew point, it is safe to assume that these curves are for actual outside temperatures. If window and outside temperatures are known, the introduction of the volume of air given by these curves will preclude any chance of fog formation on the glass. However, it does not follow that smaller volumes of fresh air will cause serious fogging since the condensation of water vapor from the air is obviously reducing the amount in the air and, thus, reducing the dew point. Thus, with insufficient air delivery, fog

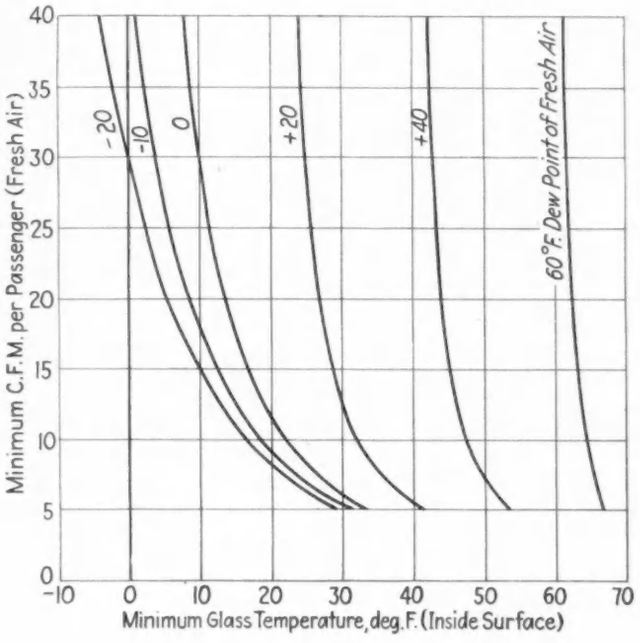


Fig. 1 – Air Required to Eliminate Fogging of Glass (Based on 8.75 Grains per Min. of Water Added per Passenger)

may occur on only the coldest parts of the glass and, in so doing, it will prevent formation on the windshield, which generally runs 1 to 2 deg. fahr. warmer than the side windows.

Since the car must be heated and since a blast of zero air is not exactly comfortable, the logical method to use is to add enough heat to the incoming air to heat the car. This method means that sufficient heat must be added to the air to raise its temperature from zero to, say, 140 deg. fahr. before introducing it into the car. If the car is to be maintained at 70 deg. fahr., the heat in the air, available to heat the car, will only be that given up by the air in cooling from 140 deg. fahr. to 70 deg. fahr., or half the heat originally put into the air in this case. Although this condition would seem to indicate a heater twice as large as is normally used, a little consideration will show that this is not the case. The quantity of heat liberated from a given size heating surface is roughly proportional to the difference in temperature existing between the surface and the air. This difference is far greater with 0 deg. fahr. air entering than with 70 deg. fahr. air and so the same size surface liberates proportionally more heat. Again, with the conventional recirculation-type heater, considerable air leaks into the car, which first must be heated to 70 deg. fahr. by the air in the car before it enters the heater. When sufficient fresh air is introduced to cause the air to leak out rather than in, this part of the heating load is transferred from the air in the car directly to the heating surface, and so must be subtracted from the heating load as figured for a conventional heater. These two factors influence results to such an extent that actual practice shows the heating surface required for a fresh-air heater is not greatly in excess of a recirculating-type heater, when the same inside temperature is maintained.

When hot water is used as a source of heat, the final air temperature is limited by economical design of heating surface to about 140 deg. fahr. If a definite temperature is to be maintained in the car at a given outside temperature, the amount of heat required to maintain this temperature is fixed. Therefore, we must supply enough air at 140 deg. fahr. to take care of this load, irrespective of the volume of air required to take care of glass-fogging.

The actual heating load in passenger cars naturally varies with the individual design of car, as well as with speed. Experience shows a fairly safe figure for present-day five-passenger enclosed cars to be 160 B.t.u. per hr. per deg. difference of inside and outside temperature at 50 m.p.h. with driver only, and excluding outside air leakage. Bus design varies to such an extent that it is impossible to give even a rough figure covering this field. On many bus designs it is not economical to supply sufficient fresh air to exclude leaks, thus adding another variable. In other words, a bus must be figured carefully and should be actually tested for heat loss before specifying equipment.

In testing any vehicle for heat loss or gain, a road test should be used, and losses for conditions other than those of the test should be figured from test results. To date we have no cold or hot rooms that actually will simulate outside conditions on the road at speed, as far as car heat loads are concerned. A car traveling at 60 m.p.h. with a 20 m.p.h. cross wind at zero, has little in common with a car in a cold room at the same temperature. On the road we can get almost any condition except temperature whereas, in the cold room, we can get only temperature. Since heat loads and surface temperatures vary in a complex manner with car speeds and wind direction but in a simple manner with outside temperatures, the fallacy of cold-room tests is obvious. In road-testing a car or bus for winter equipment, the vehicle should be tested at night to eliminate the reduction of load due to the sun in daytime. If more than one person is present on the test, corrections must be made for the amount of heat given off by the extra passengers. Conversely, for summer equipment, tests must be made in bright sunshine as well as at night. In this manner, after making corrections for temperature, the load due to the sun can be calculated and then corrected for different latitudes and times. In all cases, heating or cooling loads due to temperature differences inside and outside the vehicle will be directly proportional to the difference in temperature of the test, other variables being the same.

Assuming a five-passenger car at 50 m.p.h. in zero weather to be maintained at 70 deg. fahr with air supplied from outside and heated to 140 deg. fahr., the heat load will be $160 \times (70 - 0)$ deg. or 11,200 B.t.u. per hr. To deliver this heat, the air will be cooled from 140 to 70 deg. fahr., giving a 70 deg. fahr. temperature drop. Figuring the volume of air at inlet conditions, namely 0 deg. fahr., with a specific heat of 0.241 B.t.u. per lb., the weight of air to supply will be:

$$\frac{11{,}200}{(140 - 70) \times 0.241 \times 60} = 11 \text{ lb. per min.}$$

As one pound of 0-deg. air contains 11.6 cu. ft., the volume of outside air required will be 11×11.6 or 128 cu. ft. per min. Comparing this value with the 90 cu. ft. per min. required to eliminate fog, it is obvious that, in this case, the system must be designed on the basis of heat requirements.

The design of winter equipment for city buses calls for meeting conditions somewhat different than those desired in the private car. City-bus passengers are liable to be out in the open for long periods, and so they are dressed for the occasion. To maintain such a bus at 70 deg. fahr. in zero weather would be inviting a pneumonia epidemic. Fifty deg. fahr. in such weather appears to be more desirable, although the best temperature for city bus service at various outside conditions still needs considerable investigation. In city service a 40-passenger bus may vary from no passengers to the full complement; it may run with doors opening frequently or infrequently; and it may be in the shelter of buildings or on a wind-swept stretch, all in the period of half an hour. Under these conditions the heating load obviously is varying all over the map, and yet we know, from the standpoint of health, that the bus should be maintained at one optimum temperature. This condition calls for a very flexible system under thermostatic control and, from

the mere standpoint of health alone, calls for plenty of fresh air. We cannot stop bacilli from being exhaled, but we can at least dilute them.

The school bus calls for practically the same consideration as the city bus, but unfortunately it gets even less. The inter-city bus falls in practically the same category as the private car as far as desired conditions are concerned. Here the passengers can remove their wraps, and door openings are less frequent.

The correct volume of fresh air to supply per passenger, for bus ventilation alone, to date is not known, and will probably be the source of considerable argument for some time to come. One transit commission has set an arbitrary figure of 10 cu. ft. per min., which appears to be ample for a city bus if smoking is not allowed. Experience points to a minimum of 15 cu. ft. per min. per passenger in private cars with passengers smoking. It would, therefore, seem safe to take this figure as a maximum required in any bus for ventilation alone. The volume of air required to eliminate fogging of windows is a definite quantity for a given set of conditions, and is figured in exactly the same manner as for the passenger car. As in the passenger car it will be found that, if this volume of fresh air is introduced into the bus, ventilation will be ample to satisfy the customers. Considering the number of school-bus accidents attributed to fogged windows, it would appear as though this method of figuring ventilation requirements should be used from a safety standpoint alone.

In general, the amount of air supplied to eliminate fog will be ample to heat the bus. In the case of some inter-city equipment, however, if temperatures of 70 deg. fahr. are to be maintained, this volume of air will be insufficient and, if quantities are increased, the engine cooling system will not be able to supply the heat required when under light load. This condition calls for a system supplying enough fresh air to take care of ventilation, with the remainder supplied from inside the bus as recirculated air. Although this method generally will allow the heating system to get by without overloading the engine cooling system, the day is approaching rapidly when it will be necessary to add exhaust booster heaters to buses. Since transit commissions already are banning exhaust air heaters, boosters will have to be applied to the water systems.

Outlets for Exhaust Air

When fresh air is introduced into an enclosed space, sufficient openings must be provided to allow the displaced air inside to escape, without building up excessive pressures on the supply fan. However, these outlets must not be oversized, as it is necessary to maintain a positive pressure inside the space to eliminate leakage of outside air. Removal of air from private cars at present is dependent upon leakage and upon open windows. This method is uncertain to say the least, and frequently results in insufficient delivery of air to the car when the windows are closed. It is therefore evident that, in future designs, separate provisions must be made to allow displaced air to escape.

The problem of maintaining proper conditions inside a vehicle in summer is approached in the same fundamental manner as the winter problem, except that heat is removed rather than added, and that proper conditions are affected by different factors. In winter, clear vision was felt to be more desirable than maintaining proper humidity for comfort. In summer, clear vision will not be affected by maintaining proper comfort conditions and, therefore, the amount of humidity desired will be based upon comfort alone. Once the desired dew point, heat gains, and temperatures are decided, the problem of how much air to deliver to the passenger space is figured by the same method as was used in the winter problem. Since the dew point of the air entering is under the designer's control in the summer and not fixed as in the winter problem, it is generally necessary to make several trial solutions before arriving at an economical design. Also, since the volume of air required for removing heat is far greater than that required to remove moisture, the problem calls for a system supplying part fresh, and part recirculated air.

The proper conditions of temperature and humidity for maximum comfort depend largely upon outside conditions. The American Society of Heating and Ventilating Engineers has published optimum comfort conditions for different outside temperatures based on prolonged research, and it would appear logical to take its recommendation until experience gives us more information. As an example, for an outside temperature of 95 deg. fahr., they specify an inside temperature of 80 deg. fahr. and a relative humidity of 45 per cent, as the condition satisfying the greatest number of persons. What little information we have tends to show that this inside temperature is somewhat low for comfort where passengers are in the car for short periods, as for example, in taxi service. However, it probably will be demanded for long trips, and so it would seem advisable to design for this condition.

A human is something like an electric percolator in that he gives off heat by convection because he is hotter than his surroundings, and he gives off heat by evaporating water because he evolves steam at a higher pressure than the surrounding water pressure. Likewise he has a definite amount of heat to dissipate. If his surroundings are hotter than he is, he must dissipate all his heat as water vapor and, in so doing, he certainly will not increase the temperature of his surroundings since they are hotter than he. In other words, the quantity of latent heat a human gives up in evaporating water will not increase the sensible heat load on any cooling system. Only that given off by convection as sensible heat will have any effect on this load. The average human will give off 260 B.t.u. per hr. of sensible heat in 80 deg. fahr. and 45 per cent air. He will also give off 17.5 grains of water vapor per min. under these conditions.

Excluding leakage of air from outside into the vehicle, the cooling load inside the body will be due to the difference in inside and outside temperature, the effect of the sun, and the passengers. Each passenger will give up 260 B.t.u. per hr. as previously explained. The load due to conduction through the body in summer cannot be arrived at by a simple constant times the difference between inside and outside air temperature. For winter use, a figure of 160 B.t.u. per hr. per deg. difference was given as a safe figure for most five-passenger cars, including the heat delivered through the dash, plus the effect of the warm air from the engine surrounding part of the body. In summer the heat from the engine is increasing, rather than decreasing this load. Also, since winter fronts are discarded and louvers opened wide, a large volume of hot air now surrounds part of the body, thus increasing the average surrounding temperature. The sun generally heats the roof to a temperature higher than the surrounding air, and so this part of the body load cannot be considered as a direct function of inside and outside temperature. If the roof is hotter than the surrounding air, it must be giving up heat to that air as well as to the inside of the car. Also, as the car speed increases, the amount of heat given up to the outside air by the roof will increase, thus reducing the roof temperature and so reducing the cooling load required for this item. One test showed a drop in roof temperature from 135 deg. fahr. at 20 m.p.h. to 115 deg. fahr. at 60 m.p.h. with a corresponding 35 per cent reduction of roof load. The floor of the car is subjected to radiant heat from the hot road, as well as hot air from the engine. This load will increase or decrease with speed depending upon the individual design of car. The remaining loads are those due to sun entering windows and warming panels,

and conduction due to temperature differences, the latter generally increasing with speed.

Considering these variables, it is obviously impossible to give any safe "rule-of-thumb" figure for summer heat loads due to conduction and sun effect. Therefore, for purposes of illustration, this load will be taken as found experimentally for one medium-priced four-door five-passenger car, namely 6200 B.t.u. per hr. at 95 deg. fahr.

Irrespective of conditions maintained, the proper amount of fresh air for ventilation per passenger must be supplied. The body cannot be bottled up in an attempt to cut down costs, because a properly ventilated space without cooling is far more comfortable than one properly cooled but not ventilated. It must be remembered, at least in a private car, that the customer expects a comfortable condition with all passengers smoking. Also, it is necessary to admit fresh air in order to stop the leakage of outside air into the car.

The following problem is given to illustrate how a cooling system is figured, as well as to give a general idea of what capacities probably will be required. Although a private car is used for illustration, the method applies to any form of vehicle. The car is a five-passenger sedan operating in 95 deg. fahr. and 45 per cent humidity air with 80 deg. fahr. and 45 per cent humidity maintained inside. 100 cu. ft. per min. of fresh air are supplied, which are mixed with the required recirculated air ahead of the cooling coil. This resulting air is then cooled and delivered at 60 deg. fahr.

Heat Load — Total, B.t.u. per hr.

5 passengers at 260 B.t.u. per hr. = 1300

Conduction and solar heat = 6200

B.t.u. per hr. to be removed from body = 7500

Water Vapor Load

5 passengers at 17.5 grains per min. = 87.5 grains per min.

Total lb. of air to deliver per minute

Temperature rise 80 − 60 = 20 deg. fahr.

Specific heat of air in this range = 0.243

$$\text{Lb. per min.} = \frac{7500 \text{ B.t.u. per hr.}}{20 \text{ deg.} \times 0.243 \times 60 \text{ min.}} = 25.7$$

Lb. of fresh air per min.

100 cu. ft. per min. × 0.069 lb. per cu. ft. = 6.9 lb. per min.

Lb. of recirculated air per min.

25.7 − 6.9 = 18.8 lb. per min.

B.t.u. per lb. fresh air entering cooling coil

Conditions, 95 deg. fahr. and 45 per cent relative humidity giving 78 deg. fahr. wet bulb

B.t.u. per lb. of 78 deg. fahr. wet-bulb air = 40.64

B.t.u. per lb. of recirculated air entering cooling coil

Conditions, 80 deg. fahr. and 45 per cent relative humidity giving 65.2 deg. fahr. wet bulb

B.t.u. per lb. of 65.2 deg. fahr. wet-bulb air = 29.80

B.t.u. per lb. of mixed fresh and recirculated air

$$\frac{18.8 \times 29.80 + 6.9 \times 40.64}{18.8 + 6.9} = 32.70$$

Wet bulb temperature of mixture

32.7 B.t.u. per lb. = 69.0 deg. fahr. wet bulb

Dry-bulb temperature of mixture

$$\frac{18.8 \times 80 + 6.9 \times 95}{18.8 + 6.9} = 84 \text{ deg. fahr.}$$

Dew point of mixture entering coil

84 deg. fahr. dry bulb and 69 deg. fahr. wet bulb = 61.5 deg. fahr. dew point

Therefore, the air entering the cooling coil will be at 84 deg. fahr. with a dew point of 61.5 deg. fahr. and have a heat content of 32.70 B.t.u. per lb. This air must be cooled to the selected temperature of 60 deg. fahr., and sufficient water vapor must be removed to take care of that added by the passengers.

Dew point to be maintained in car

80 deg. fahr. and 45 per cent relative humidity air has a dew point of 57 deg. fahr.

Grains of water vapor per lb. of air in car

57 deg. fahr. dew point air contains 69 grains per lb.

Grains added by passengers per lb. of air supplied

$$\frac{87.5 \text{ grains per min.}}{25.7 \text{ lb. per min.}} = 3.4 \text{ grains per lb.}$$

Grains per lb. of air leaving cooling coil

69 − 3.4 = 65.6 grains per lb.

Dew point of air leaving cooling coil

Air with 65.6 grains per lb. has a dew point of 55.5 deg. fahr.

Wet bulb of air leaving cooling coil

Air at 60 deg. fahr. and 55.5 deg. fahr. dew point has a 57.3 deg. fahr. wet bulb

B.t.u. per lb. of air leaving the cooling coil

B.t.u. per lb. of 57.3 deg. fahr. wet-bulb air = 24.44

Therefore, the air leaving the cooling coil will be at 60 deg. fahr., with a dew point of 55.5 deg. fahr., and have a heat content of 24.44 B.t.u. per lb. Since it entered with 32.70 B.t.u. per lb., the coil must have removed 32.70 − 24.44 or 8.26 B.t.u. per pound. The total load will then be

25.7 lb. per min. × 8.26 B.t.u. per lb. = 212 B.t.u. per min.

or approximately one ton of refrigeration. Since one pound of air under discharge conditions contains 13.2 cu. ft., the total volume of air discharged from the cooling coil will be 25.7 lb. per min. × 13.2 = 340 cu. ft. per min. The cooling coil required for this result will have roughly 35 sq. ft. of cooling surface when used with a 30 deg. fahr. refrigerant temperature. Its overall dimensions will be approximately 10 in. x 10 in. x 6 in. deep, not including fan and motor. Positive means of distributing air to the front- and rear-seat compartment will be found necessary if reasonably constant temperatures are to be maintained. It is interesting to note that, although the results shown in this problem are derived from data taken on one car, several installations made by others on similar cars check these figures, both as to refrigerating capacity and required air volume.

The problem of controlling desired conditions in a motor vehicle is far more complex than in a residence or public building. The building is stationary and has comparatively thick walls, plus a large volume of enclosed air at the proper conditions, both of which take care of sudden changes in load. The motor vehicle, on the other hand, is operating at widely varying speeds, with sudden changes in heat load, and practically no stabilizing effect. All the air in the car may be replaced in the space of a minute, an unheard of figure in any building. The walls are of light-weight construction, even if insulated, and carry a large percentage of glass, all of which makes the car particularly responsive to changes in outside conditions, especially to solar radiation. Under these considerations it is evident that any system of control utilized, whether manual or automatic, must be capable of rapid and positive operation.

Under winter conditions the primary prerequisite of comfort is the inside temperature. Since this temperature is to be maintained by introducing heated fresh air, the simplest method of control is to vary the quantity of hot water passing through the heating coil. As previously mentioned, in city buses, it is practically imperative to have this control automatic. In private cars it can be controlled manually but, if continuous comfort is to be sold, it should be controlled automatically. It must be remembered in any manually controlled comfort system that the control is not touched until the occupant becomes uncomfortable.

Comfortable conditions in summer depend upon temperature and humidity. Unfortunately, the average human cannot make up his mind whether it is the heat or the humidity causing his discomfort and, therefore, he would not know what control to operate if given a manually controlled conditioning system. To do a job, the summer system must be controlled automatically. Since a large volume of fresh air is being introduced, it is impractical to use an "on and off" control, as inside conditions would reach those outside in about two minutes, with the refrigerating system shut off. If control is attempted by varying the temperature of the cooling coil, the humidity will vary and cause discomfort. Obviously the control of a summer system is not as simple as might be imagined, and calls for more consideration than one for a winter system.

Considering the domestic market, it is safe to assume that a conditioning system will not be installed for summer conditions alone, but will be designed for year-around operation. In other words, it will have a heating coil operated by hot water and a cooling coil operated by a separate refrigerating system, either of which can be operated independently of the other and both of which are ample to take care of maximum heating or cooling requirements. Such a system is an ideal set-up for what is termed a "dew-point control." This control is invariably used in industrial air conditioning where close control of both humidity and temperature are required. Roughly, under this system, all of the air circulated is first cooled to give the desired dew point, and then sufficient heat is added, before delivery, to maintain a constant temperature in the car. In other words, the system is designed to take care of a maximum load and, whenever the load is not a maximum, the system supplies the additional heat required to keep it a maximum. From a power standpoint this system is too expensive to operate in buildings for comfort conditioning but, in a motor vehicle, the heat is available and the cost of operating a refrigerating system, necessarily designed for maximum conditions, will be substantially the same for full load as for any fraction of full load. Therefore, the operating cost of this system is not a deciding factor in motor vehicles. Since it is one of the easiest systems to control, is one of the fastest in response, and can be used to maintain temperatures in winter as well as conditions in summer, it would appear to be the most logical system to use.

The control mechanism of such a system will consist of a thermostat set to maintain the desired temperature in the car, by controlling the flow of hot water through the heater. A second thermostat, controlling the cooling coil, will maintain the air leaving this coil at a constant temperature. Since the air leaving the cooling coil will not be saturated, and since the required dew point of this air is dependent upon the vapor given off by the passengers, this second thermostat will not control the humidity exactly. However, it will be found to be sufficiently close to maintain comfortable conditions, providing the cooling coil drops the temperature to within 5 or 6 deg. fahr. of saturation. A more accurate control can be maintained by substituting a humidistat for the second thermostat, and so controlling the cooling coil by actual humidity conditions in the car. Although this method is more accurate in this case, it calls for a delicate piece of equipment, plus a more complicated control system, and probably will be found unnecessary.

A refrigerating system is the heart of any summer conditioning equipment, and therefore the engineer must have at least a rudimentary knowledge of this subject before he can approach the problem in a logical manner. However, for the purposes of this paper, it will be sufficient to give only a few definitions and principles.

Since the refrigerating industry was built up around the natural ice industry, the unit of capacity was based on the effect of melting ice, rather than B.t.u.'s, and so we find the "ton" used as a unit, meaning the quantity of heat required to melt one ton of ice in a day. For once, nature gave us a break when she required 144 B.t.u. to melt a pound of ice, since this caused the "ton of refrigeration" to work out to the even figure of 200 B.t.u. per min.

The capacity or "tonnage" of a system must not be confused with the power required to drive the system, as this value will depend upon the temperature difference under which the system is operating. If, for example, the system is removing heat from a cooling coil at 40 deg. fahr. and giving it up to a water-cooled condenser at 100 deg. fahr., it may require 1 hp. per ton. If, on the other hand, the cooling coil is at zero and the condenser is air cooled at 120 deg. fahr., it probably will require twice as much horsepower per ton. Likewise, depending upon the refrigerant used, the compressor may be two to three times as large in the latter case. It is, therefore, desirable to keep the cooling coil at the highest possible temperature and the condenser at the lowest possible temperature to conserve compressor size and power.

In selecting a refrigerating system for a motor vehicle, we again encounter many problems not found in stationary systems. Space is at a premium; vibration may cause breaks in lines; power is available, but at widely varying speeds; occupied space is small, giving rise to serious dangers from leaks; and accidents may entrap passengers in a space open to refrigerant vapors. With these considerations in mind, it is fairly safe to assume the safest system will be one using a low-pressure refrigerant, such as Carrene, Freon 113, Freon 11, or possibly Freon 21. These refrigerants are substantially non-toxic and would exist in a cooling coil at pressures below atmospheric, thus precluding leaks into the car under operating conditions.

A great temptation naturally exists to use an absorption, or heat-operated system of refrigeration in conjunction with the exhaust of an automobile engine. Although such a system is possible, it will be found to be too bulky and too expensive to be practical, to say nothing of the chemical research necessary to utilize refrigerants other than ammonia, or possibly, Freon 21. Insufficient capacity will also be encountered under idling conditions.

When gasoline is vaporized, it absorbs heat in the same manner as any other liquid. It has been proposed, therefore, to utilize this principle for cooling motor vehicles. Assuming a car operating at 30 m.p.h. and running at 10 miles per gal., the best cooling effect possible would be roughly one-third of a ton or about one-third the required rate for comfort. If this system were attempted in conjunction with the standard carburetor by passing the mixture through a cooling coil placed between the carburetor and the engine, the resulting temperature of the mixture would not be low enough to condense moisture out of the air, and would, therefore, be of little benefit. If a fuel such as propane is used instead of gasoline, we will still have roughly one-third the amount of cooling effect required, and it will be found more economical to install a straight refrigerating system.

The most logical system appears to be the use of a compressor operated from the engine through a suitable variable-speed drive. Such a system will require a minimum space and not over 2 hp. – a figure well below the power required to operate the present radiator fan.

Just what form this system will take depends upon future development work. However, it is certain that, in the near future, some form of safe economical automotive refrigerating system will appear and, when it does, we can go merrily on our way, laughing at the weather man and our notoriously variable climate.

Fundamentals of Heat Transmission Applied to Automotive Brakes

By A. S. Van Halteren

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MODERN developments in the automobile industry have created a paradox. On the one hand, increased speeds have placed greater demands on the brakes whereas, on the other hand, the trend toward streamlining has greatly handicapped brake performance. As a result brake drum and wheel diameters have been reduced and the flow of air to the brakes has been restricted by shrouding them with wheels and skirted fenders.

In the solution of brake heat-transmission problems, the subject is considered under the following headings: the amount of heat generated; the manner and rate of heat flow *into* the brake; and the manner and rate of heat flow out of the brake. Heat-transmission calculations of specific examples are made that indicate the amount of heat dissipated by conduction, radiation, and convection.

IN beginning a discussion of the subject of heat transmission in connection with brakes and wheels, I wish to emphasize the growing seriousness of the problems involved. Modern developments in the automobile industry have created a paradox. On the one hand increased speeds have placed greater demands on the brakes whereas, on the other hand, the trend toward car streamlining has greatly handicapped brake performance. As a result, brake drum and wheel diameters have been reduced and the flow of air to the brakes has been restricted by shrouding them with wheels and skirted fenders.

That this subject has received scant attention in the past is shown clearly by the dearth of published matter relating to it. Technical periodicals contain only brief references to the subject of brake heat transmission, the most extensive discussion of the subject discovered being found in the 1917 S.A.E. TRANSACTIONS[1]. It is apparent that such an important subject has not received the recognition that it deserves from automobile engineers in general. Brake engineers, however, have a better appreciation of the problem because they know that heat is responsible either directly or indirectly for 90 per cent or more of their brake difficulties.

In the solution of brake heat-transmission problems, the subject may be considered under the following headings:

1. The amount of heat generated.
2. The manner and rate of heat flow *into* the brake.
3. The manner and rate of heat flow out of the brake.

Basically, a brake is a machine for the conversion of the energy of motion into heat, and the amount of heat generated varies directly as the weight of the car and as the square of the speed. To get an idea of the magnitude of the quantities involved, let us assume a definite car weight and speed.

For example, a car weighing 5500 lb. loaded, stopping with a deceleration of 15 ft. per sec. per sec. from a speed of 80 m.p.h. converts 1,175,762 ft-lb. of energy into heat according to the equation:

$$\text{Kinetic Energy} = 1/2\ M\ V^2.$$

This power is equivalent to 273 hp. or 1511 B.t.u. If we assume that all this heat is held momentarily in the four brake drum rings having, let us say, a combined weight of 45 lb., the average temperature increase per ring would be 305 deg. fahr.[2] These figures clearly illustrate the immense amount of work demanded of the modern car brake.

When brakes are applied, heat flows from the surface of the brake drum into the ring section of the drum and into the brake shoes, the latter absorbing about 5 per cent of the total. At once the temperature of the ring rises above that of its surroundings and heat begins to flow out of the ring by radiation and convection into the air, and by conduction into the drum back. From the back it is conducted to the hub and wheel where it is stored until finally disposed of through radiation and convection. Since most of the heat is absorbed by the brake drum, we have confined our subsequent discussion of heat transmission to that part of the brake.

In taking up the problems connected with heat transmission as applied to brakes, we naturally turned first to the fundamental formulas of heat transmission. We found that these formulas, developed as they are for specific cases, could not be applied directly to brake conditions although they did help us to appreciate the complexity of the problems involved.

For example, the basic formula for the calculation of heat conduction, namely, $Q = K\ A\ Z\ (T_1 - T_2)$ applies only where there is a steady flow of heat. This condition is never obtained in brake work for here the heat flows intermittently and with varying intensity during a stop. Also, because the temperature varies during the stop, it is difficult to choose the proper value for K, the coefficient of thermal conductivity, since this value changes with the temperature. McAdams in his excellent volume, "Heat Transmission," gives the value of K[3]

for cast iron as 29 B.t.u./sq. ft./deg. fahr./ft./hr. at 32 deg. fahr.

for cast iron as 23 B.t.u./sq. ft./deg. fahr./ft./hr. at 572 deg. fahr.

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[1] See S.A.E. TRANSACTIONS, Part I, 1917, pp. 368-376: "Retardation of the Automobile," by John Younger.

[2] For specific heat of iron at various temperatures see "Heat Transmission," by William H. McAdams, p. 330, McGraw-Hill Book Co., 1933.

[3] See "Heat Transmission," by William H. McAdams, p. 308, McGraw-Hill Book Co., 1933.

for mild steel as 36 B.t.u./sq. ft./deg. fahr./ft./hr. at 32 deg. fahr.

for mild steel as 27 B.t.u./sq. ft./deg. fahr./ft./hr. at 572 deg. fahr.

Application of this formula also is made difficult because we have not been able to agree on the temperatures reached at the braking surface.

Because of these complications it was necessary to determine the rate of heat flow through the drum ring by actual experiment. A thermocouple was placed on the outer surface of a brake drum having a ring thickness of ¼ in. It was found that the temperature of the outer surface began to rise within less than 1 sec. after the brake was applied.

Next, in applying the Stefan-Boltzmann law, expressed by the formula:

$$Q = 0.162\ p\ A_1\ (T_1^4 - T_2^4) \times 10^{-8},$$

we are confronted with the same difficulties. This equation gives the quantity of heat radiated but only where a constant temperature difference is maintained between the radiating body and its surroundings. Application of this equation also involves consideration of the effect of the emissivity of the surfaces, their shape, and relative position. The quantity of heat given off by radiation is proportional to the emissivity, p, which varies with temperature and character of the surface. To show how widely this figure varies, we have set down a few of the values for this factor, p, as given by Hottell[4]:

	Deg. Fahr.	p
Cast iron, polished at	392	0.21
Cast iron, newly turned at	72	0.435
Cast iron, rough, strongly oxidized at	100–480	0.95

Since the emissivity of a radiating surface is the ratio of its emissive power to that of a black body, this value can never exceed 1.0. It is, therefore, interesting to note that the rating of 0.95 given rough cast iron shows that a cast-iron drum with a rough oxidized outer surface closely approaches a black body in emissive powers.

The character of the surface is also important in determining the value of the surface conductance factor, k, used in the fundamental formula for calculating the quantity of heat transmitted by convection, namely:

$$Q = kSO_m.$$

The authorities give values for k but only for a few special cases in which the conditions differ materially from those with which we are concerned. The nearest approach to our problem which we have been able to find was made by Hinlein, who determined experimentally values for k as applied to rotating cylinders, such as flywheels and armatures. These values, as set forth in Mark's Handbook under the heading: "Transmission of Heat from the Surface of a Rotating Cylinder to Air,"[5] are given for various surface conditions and velocities of rotation. For example, for a rough surface rotating at a velocity of 40 ft. per sec., the value for K is 3.32 B.t.u./sq. ft./hr./deg. fahr. difference in temperature.

Thus far we have attempted to show why it is not practical to apply the three fundamental formulas to brake heat-transmission problems as they exist in actual practice, first, because of a lack of accurate data on the value of various constants and, second, because these formulas apply only when the heat flow is steady. Our investigations have impressed us with the fact that nearly all formulas developed for the calculation of heat flow and temperature are very restricted in their application.

However, we can obtain some idea of the relative importance of conduction, radiation, and convection in the dissipation of heat from a brake drum by assuming hypothetical conditions for a brake drum which approach the actual under severe conditions on the road. For example, let us assume a brake drum 12 in. in diameter with a drum-back thickness of ⅛ in. and a radiating area on the outside equivalent to that of 1.25 sq. ft. of flat plate. Let us assume further an atmospheric temperature of 70 deg. fahr., a constant drum-ring temperature of 400 deg. fahr., and a temperature gradient in the drum back of 80 deg. fahr. per in., which latter condition corresponds closely to actual experience.

For heat conducted to the back, using a value for K of 30 in the formula

$$Q = K\ A\ (t_1 - t_2)/1,$$

we obtain a figure of 0.26 B.t.u. per sec.

For heat radiated from the ring, using a value of 0.95 for emissivity p in the formula,

$$Q = 0.162\ p\ A\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]$$

we obtain 0.25 B.t.u. per sec.

For heat carried away by convection, using a value of 3.32 for surface conductance, k, in the formula,

$$Q = k\ S\ O_m$$

we obtain 0.38 B.t.u. per sec.

Adding these three figures, we find that heat is dissipated from the drum ring at a rate of 0.89 B.t.u. per sec. This figure checks closely with experimental data. In our first example we showed that 1511 B.t.u. were generated in the four drums in 8 sec. This value would amount to an average for each drum of 47 B.t.u. per sec. Strictly speaking, these two figures are not comparable but they show in a general way how much faster heat is generated than it is dissipated.

Expressed in percentage, we show the heat dissipated per sec. as follows:

Conducted to the back	0.26 B.t.u.	29.2 per cent
Radiated from the ring	0.25 B.t.u.	28.1 per cent
Conducted to the air by convection	0.38 B.t.u.	42.7 per cent
	0.89 B.t.u.	100.0 per cent

Since heat is radiated in proportion to the fourth power of the absolute temperature, the percentage of heat dissipated by radiation would be greater at higher temperatures, but we conclude from these figures that, even under severe conditions, more brake heat is disposed of finally through convection than through either radiation or conduction. This statement refers, of course, to primary dissipation only, for it is obvious that all heat generated by the brakes eventually must be disposed of through convection and radiation.

Much experimental work has been done on the road and in the laboratory to determine what benefits might be gained by increasing the weight and surface of the drum ring beyond average practice. The most enlightening test on this subject which we have run was made on the dynamometer where, incidentally, we test under the most favorable conditions for comparisons of this character. A series of three 12-in. drums were built with variations confined to the drum ring, as follows:

	Drum Weight	Outside Area of Drum Ring
Drum No. 1	14.1 lb.	1.55 sq. ft.
Drum No. 2	16.4 lb.	1.62 sq. ft.
Drum No. 3	16.1 lb.	2.70 sq. ft.

For the purpose of studying the effect of varying mass and radiating area on peak temperatures, a thermocouple was placed in each drum ring at the center of and about 0.050 in. from the braking surface. The test was run both with and

[4] See "Heat Transmission," by William H. McAdams, p. 46, McGraw-Hill Book Co., 1933.

[5] See "Mechanical Engineers' Handbook," by Lionel L. Marks, Third Edition, p. 403, McGraw-Hill Book Co., 1930.

without a m.p.h. air blast in an attempt to approximate road conditions on front and rear drums. From the data obtained covering various speeds and loads the peak temperatures developed by making six consecutive 15 ft. per sec. per sec. stops from 60 and 70 m.p.h. at 1-min. intervals with a 149,000 ft-lb. load, were selected for comparison. These particular examples were chosen because, in our opinion, they show most clearly both recovery characteristics and thermal capacity.

By comparing Drum No. 1 with Drum No. 2 we found that increasing the mass of the drum ring 16 per cent and the area 4½ per cent reduced the peak temperature 3 per cent with the blower on and 8 per cent without the blower.

Comparing Drum No. 3 with Drum No. 1 we found that increasing the mass 14 per cent and the area 74 per cent reduced the peak temperatures 11 per cent with the blower and 10.6 per cent without the blower.

Differences obtained between the front and rear drum on a car were illustrated forcibly by observing that the maximum temperature of Drum No. 1 was reduced from 761 deg. fahr. to 557 deg. fahr., or 204 deg. fahr. by providing a 40 m.p.h. air blast. This drop in temperature amounting to 26.5 per cent demonstrates the difference between natural and forced convection.

Data recorded in road work back in 1933 give further proof of the value of unimpeded air circulation under conditions which gave even less contrast between front and rear drums than we find today. At that time, running at 40 m.p.h., it required 1/3 longer to cool rear drums from 700 deg. fahr. to 100 deg. fahr. than to cool front drums. With the car standing still it took the front drum 2½ times as long to cool through the same range as when it was moving at 40 m.p.h.

When wheel covers first were proposed, it was proved by repeated tests that they were a serious menace to brake efficiency. With conditions as serious as they are now, it seems logical as a future development to put openings in the wheels to permit greater circulation of air around the brake mechanisms. Experiments along this line have shown that both good and bad results are obtained when the brakes are shrouded too thoroughly, because with liberal wheel openings the direction of the wind becomes too great a factor in the effectiveness of the four assemblies. With the brakes well out in the air stream away from the centerline of the wheel, it should be practical to take advantage of liberal openings through the wheel.

Internal ventilation of brake assemblies offers another method for the dissipation of heat. The air scoops used by the winner of last year's Vanderbilt Cup race were an important factor in winning that race because the driving speed was determined to a large extent by the maintenance of brake efficiency through adequate cooling. Experimental work done on this problem indicates, however, that, in order to gain any real advantage, there must be a considerable amount of air directed across the braking surface. But, here again, care must be used to avoid making the direction of the wind too much of a factor in the distribution of ventilation to the four assemblies.

It is common practice nowadays to open up truck brakes for internal cooling, and there are a number of patents covering the internal ventilation of brakes by the use of pumps, exhaust ejectors, and by the translation of the car.

In summarizing this subject I am impressed with the high rate at which brake heat is generated and conducted through metal and with the slow rate at which the heat finally is dissipated.

The greater part of the heat stored in the hubs and wheels, because of their low temperatures, is disposed of finally by convection rather than by radiation. Unfortunately, however, these reservoirs of heat are surrounded by pockets of dead air which retard the dissipation of heat by convection.

Conductance to the air by convection is stimulated by the fact that the brake heat reservoirs revolve. However, our examples show that the forced convection furnished by the translation of the car is the most effective means available for the final disposition of brake heat. In order to avoid reaching the critical temperatures of the materials used, the heat reservoirs must be provided with enough air circulation to take care of the heat which is generated as much as fifty times as fast as it is dissipated.

We have much to learn in connection with brake heat-transmission problems. However, much can be accomplished by promoting an understanding of these problems and by suggesting compromises in appearance and comfort when there is an opportunity to contribute toward safety.

I have in mind, for example, bumper and fender heights; the relation between the centerline of wheels and brakes; designs for gravel deflectors used in rear fenders; and engine heat deflectors, which tend to break the vacuum behind front brake drums.

Brake engineers have been concentrating on the solution of problems within the brake assembly. It is imperative that they give more attention to the elimination of unnecessary restrictions to the flow of air to the brakes.

Combined Body and Chassis Construction

THERE is no doubt that the combined body and chassis construction will be adopted generally for quantity-production cars. European manufacturers have adopted this idea several years ago, and almost all new models are being built in this manner. Body engineers will have to step out and absorb some of the duties of the chassis engineers, and close give-and-take cooperation will, no doubt, give excellent results.

What are we gaining by this innovation? First of all we are reducing the number of individual units. We have one solidly welded unit instead of two, eliminating intricate fastening methods of body to frame and, incidentally, doing away with possible noise and squeaks from that source. We possibly can reduce overall height, increase ratio of inside dimensions to outside ones, and speed up assembly, with fewer parts to fit, and all these advantages would be capped by a considerable weight saving.

As to structural quality, there is no question that this construction gives maximum stiffness in all directions, especially with the now universally adopted steel roof. With this construction we have a most complete truss frame design, consisting of modern lines which completely divert the slightest suggestion of squareness of simple bridge or truss construction. It will be to the advantage of motor-car designers to study this rather new idea which probably will be adopted universally shortly.

This construction brings about a considerable change in our systematized assembly line; however, it is not a great stumbling block. It appears at first almost insurmountable but, after careful consideration, it seems to work out very nicely. It will, at the start, be annoying but, after one gets used to it, it will go as smoothly as the original line-up. A body and chassis unit will have to be worked out in such a manner that the assembly of engine suspension, axles, and so on, can be done from the outside.

Excerpts from the paper: "Body and Chassis Development," by Joseph Ledwinka, Edward G. Budd Mfg. Co., presented at the National Passenger Car Meeting of the Society, Detroit, Mich., Mar. 29, 1938.

Diesel Deposits as Influenced by Fuels and Operating Conditions

By J. R. MacGregor and W. V. Hanley
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FUEL deposition and ring-sticking tests are described which were performed in several single-cylinder and multicylinder service Diesel engines in the laboratory.

The development of an accelerated test method is outlined with special reference to the effects of engine variables on deposition. Decrease in load, speed, or jacket temperature or increase in altitude were found to increase fuel deposition. Increase in running time increased the exhaust deposits linearly but, within the combustion-chamber, equilibrium deposition was reached in a few hours of operation.

Marked differences were found among fuels in the single-cylinder test engine after 24 hr. of operation under the accelerated conditions. Fuels doped with different types of cetane-number improvers indicated that ignition quality is a factor in fuel deposition under certain operating conditions in some engines.

The results of the tests in one of the single-cylinder engines were compared with the results obtained in the other engines. Although they correlate closely with the relative fuel deposition in certain of these engines, they were not indicative of the results obtained in other engines which are relatively insensitive to fuel differences.

The tests demonstrated that the engine designer has much greater control over the quantity of fuel deposits formed than has either the fuel refiner or the engine operator.

MOST of the published information on the performance of fuels in high-speed Diesel engines has dealt with the initiation of combustion or the delay in initiation of combustion. Both of these phenomena have been found to correlate at least approximately with ease of starting, knocking, and formation of exhaust smoke. For this reason they have been and still are of considerable importance.

Less work has been done on the later phases of the combustion process which includes the rate of burning, duration of combustion, and completeness of burning. As in the case of the Otto-cycle engine, the study of the actual combustion process in a high-speed Diesel engine is fraught with many difficulties. Temperatures, pressures, and volumes all vary between wide limits, and the total interval of time involved in these changes must of necessity be measured in microseconds. For these reasons highly specialized instrumentation is required.

Among the methods which have been used to investigate the combustion process are the following: sodium-line reversal method of measuring temperature[1, 2, 3, 4]; spectroscopic analysis of radiation[5, 6]; Schlieren photography showing flame movement by difference in refraction of gases of different density[7, 8]; instantaneous sampling[9, 10]; ionization gaps indicating presence and location of flame[11, 12, 13]; inertialess pressure and rate of change of pressure indicators, radiometric measurement of flame temperature[14, 15]; photoelectric detection of flame travel[16]; high-speed flame photography[17, 18, 19, 20]; and quantitative measurement of fuel deposits[21].

In order to study the last phase of the combustion process four of these methods have been investigated in the laboratory of the Standard Oil Co. of Calif. using the Diesel engines shown in Fig. 1. Typical results from tests in these engines have been selected for the following presentation:

The first method made use of a precombustion-chamber type engine which was fitted with a glass window through which the flame illumination was transmitted to a phototube. By means of a thyratron relay connected to the phototube, a neon light mounted on the flywheel was used to indicate stroboscopically the beginning and end of inflammation. The second method was similar to the first except that the presence of flame was detected by an ionic gap. The third method made use of an engine indicator of a new type and cathode-ray oscillograph for studying the rate of pressure rise. The fourth method, to be described in detail, was the inferential method of testing a variety of fuels under various controlled engine conditions and studying the resulting engine deposits as an

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indication of the completeness of combustion. Since cases of stuck rings and valves and excessive engine deposits have been observed in the field in some engines under certain operating conditions, this last method has the advantage of yielding results which are indicative of service performance apparent to the engine operator.

Besides studying the combustion processes it was desired to develop an accelerated method of test in the laboratory service engine that would yield results which would correlate with deposits in other popular service engines. A service engine, rather than an experimental engine, was used in the laboratory in order to approximate field conditions more closely. It was considered permissible to alter the test conditions in the laboratory engine as required, to obtain a reproducible quantity of deposit sufficiently large to weigh accurately, provided that the conditions selected were encountered at some time in field operation.

After preliminary tests had indicated the type of engine most subject to the formation of deposits, the first step was to determine the effect of operating conditions in this engine on deposition, in order to select a set of conditions which would furnish reproducible results in a short time. It is known that engine deposits are influenced by both lubricating oil and fuel and, in order to exaggerate the influence of the fuel, it was desired to have the lubricating oil exert as small an influence as possible. For this reason two well-known and commonly used types of lubricants were tried in order to select the lubricant which would contribute the least to engine deposits. Table 1 shows the results of these lubricating-oil tests. It will be noted that, in the case of Fuel 1 which was a normal straight-run distillate, the total deposit with Lubricating Oil B was 156 per cent of that obtained with Lubricating Oil A. With Fuel 2, which was a different type of straight-run distillate, Lubricating Oil B formed 121 per cent of the deposit obtained with Lubricant A. Lubricating Oil A was, therefore, selected for all the subsequent fuel tests.

Table 1 – Comparison of Engine Deposits with Two Types of Lubricating Oils

Fuel	1	1	2	2
Lubricating Oil	A	B	A	B
Engine Deposit				
Combustion Chamber, gm.	4.40	6.26	2.94	3.86
Per cent	100	142	100	131
Nozzle, gm.	0.08	0.23	0.40	0.14
Per cent	100	288	100	35
Exhaust Valve, gm.	0.46	1.20	0.27	0.39
Per cent	100	262	100	144
Total, gm.	4.94	7.69	3.61	4.39
Per cent	100	156	100	121

Load also has been known to have an influence on engine deposits and tests therefore were carried out, covering a range from idle to three-quarters, to evaluate its effect in the test engine. The effect of load is illustrated by a comparison of Fig. 2 and Fig. 3 which shows the condition of the cylinder-head, exhaust valve, and injection nozzle at the completion of the idle and three-quarter-load runs respectively, and is evaluated in Fig. 4. As shown by Fig. 4 the total deposits at idle was approximately five times that at three-quarters load.

The effect of load is attributable partially to changes in engine temperature, and a range of the operating temperatures that are subject to control therefore was investigated. The temperatures controlled were intake air, jacket, and lubricating oil. Lowering the intake temperatures from 150 to 100 deg. fahr. increased the deposit in the combustion-chamber but had no effect on exhaust-valve deposit. These results are shown in detail in Table 2.

Table 2 – Effect of Intake-Air Temperature on Engine Deposits

Intake Air Temperature, deg. fahr.	150	100
Engine Deposits		
Combustion Chamber, gm.	2.76	4.86
Per cent	100	176
Nozzle, gm.	0.17	0.15
Per cent	100	88
Exhaust Valve, gm.	0.37	0.38
Per cent	100	102
Total, gm.	3.30	5.39
Per cent	100	163

The effect of jacket temperatures on deposits is shown in Table 3.

Table 3 – Effect of Jacket Temperature on Engine Deposits

Jacket Temperature, deg. fahr.	200	150	125
Engine Deposits			
Combustion Chamber, gm.	3.00	4.80	4.00
Per cent	100	160	133
Nozzle, gm.	0.21	0.15	0.11
Per cent	100	71	52
Exhaust Valve, gm.	0.29	0.36	0.44
Per cent	100	124	152
Exhaust Passage, gm.	3.36	4.47	4.89
Per cent	100	133	146
Total, gm.	6.86	9.78	9.44
Per cent	100	142	138

It will be noted that an increase in deposit except at the nozzle, resulted from a decrease in jacket temperature from 200 to 150 deg. fahr. However, when comparing results shown for 150 deg. fahr. with those of 125 deg. fahr., very little

[1] See S.A.E. TRANSACTIONS, Vol. 30, April, 1935, pp. 125-136: "Flame Temperatures Vary with Knock and Combustion-Chamber Position," by G. M. Rassweiler and Lloyd Withrow.

[2] See N.A.C.A. Technical Note No. 559, March, 1936: "Combustion-Engine Temperatures by the Sodium Line-Reversal Method," by M. J. Brevoort.

[3] See University of Illinois Bulletin No. 262, October, 1933: "Flame Temperatures in an Internal-Combustion Engine Measured by Sodium Line Reversal," by A. E. Hershey and R. F. Paton.

[4] See the *Journal of the American Chemical Society*, Vol. 53, March, 1931, pp. 869-883: "Flame Temperatures of Hydrocarbon Gases," by G. W. Jones, B. Lewis, J. B. Friauf, and G. Perrott.

[5] See *Industrial and Engineering Chemistry*, Vol. 18, May, 1926, pp. 528-531: "Ultraviolet Spectroscopy of Flames of Motor Fuels," by G. L. Clark and W. C. Thee.

[6] See *Industrial and Engineering Chemistry*, Vol. 23, July, 1931, pp. 769-776: "Spectroscopic Studies of Engine Combustion," by Lloyd Withrow and G. M. Rassweiler.

[7] See *Review of Scientific Instruments*, Vol. 2, September, 1931, pp. 522-531: "Applications of Schlieren Method of Photography," by D. B. Gawthrop.

[8] See N.A.C.A. Technical Report No. 588, 1937: "Fuel Spray and Flame Formation in a Compression-Ignition Engine Employing Air Flow," by A. M. Rothrock and C. D. Waldron.

[9] See N.A.C.A. Technical Note No. 454, March, 1933: "The N.A.C.A. Combustion-Chamber Gas-Sampling Valve and Some Preliminary Test Results," by J. A. Spanogle and E. C. Buckley.

[10] See *Industrial and Engineering Chemistry*, Vol. 22, September, 1930, pp. 945-951: "Following Combustion in the Gasoline Engine by Chemical Means," by L. Withrow, W. G. Lovell, and T. A. Boyd.

[11] See N.A.C.A. Technical Memo. No. 668, April, 1932: "Combustion Velocity of Benzine-Benzol-Air Mixtures in High-Speed Internal-Combustion Engines," by K. Schnauffer.

[12] See *Forschung auf dem Gebiete des Ingenieurwesens*, Vol. 6, No. 4, August, 1935, pp. 183-191: "Die Fortpflanzung der Verbrennung in Dieselmotor," by L. Breves.

[13] See the *Philosophical Magazine*, Vol. 22, 1911, pp. 775-791: "The Mobility of the Positive Ion in Flames," by S. G. Lusby.

[14] See *Industrial and Engineering Chemistry*, Vol. 24, August, 1932, pp. 867-870: "Flame Radiation and Temperature Measurements on an Internal-Combustion Engine," by A. E. Hershey.

[15] See N.A.C.A. Technical Report No. 486, 1934: "Infrared Radiations from Explosions in a Spark-Ignition Engine," by C. F. Marvin, Jr., F. R. Caldwell, and S. Steel.

[16] See S.A.E. TRANSACTIONS, Vol. 31, November, 1936, pp. 459-468: "Photo-Electric Combustion Analysis," by R. A. Rose, G. C. Wilson, and R. R. Benedict.

[17] See N.A.C.A. Technical Memo. No. 790, March, 1936: "Ignition and Flame Development in the Case of Diesel-Fuel Injection," by O. Holfelder.

[18] See N.A.C.A. Technical Note No. 496, April, 1934: "A Preliminary Motion-Picture Study of Combustion in a Compression Ignition-Engine," by E. C. Buckley and C. D. Waldron.

[19] See N.A.C.A. Technical Report No. 401, 1932: "Combustion in a High-Speed Compression-Ignition Engine," by A. M. Rothrock.

[20] See N.A.C.A. Technical Report No. 429, 1932, and N.A.C.A. Technical Note No. 389, 1931: "The N.A.C.A. Apparatus for Studying the Formation and Combustion of Fuel Sprays and the Results from Preliminary Tests," by A. M. Rothrock.

[21] See "Characteristics of Diesel Fuel Oil," by W. F. Joachim, presented before the Oil and Gas Power Meeting of the A.S.M.E., Tulsa, Okla., May 8, 1935.

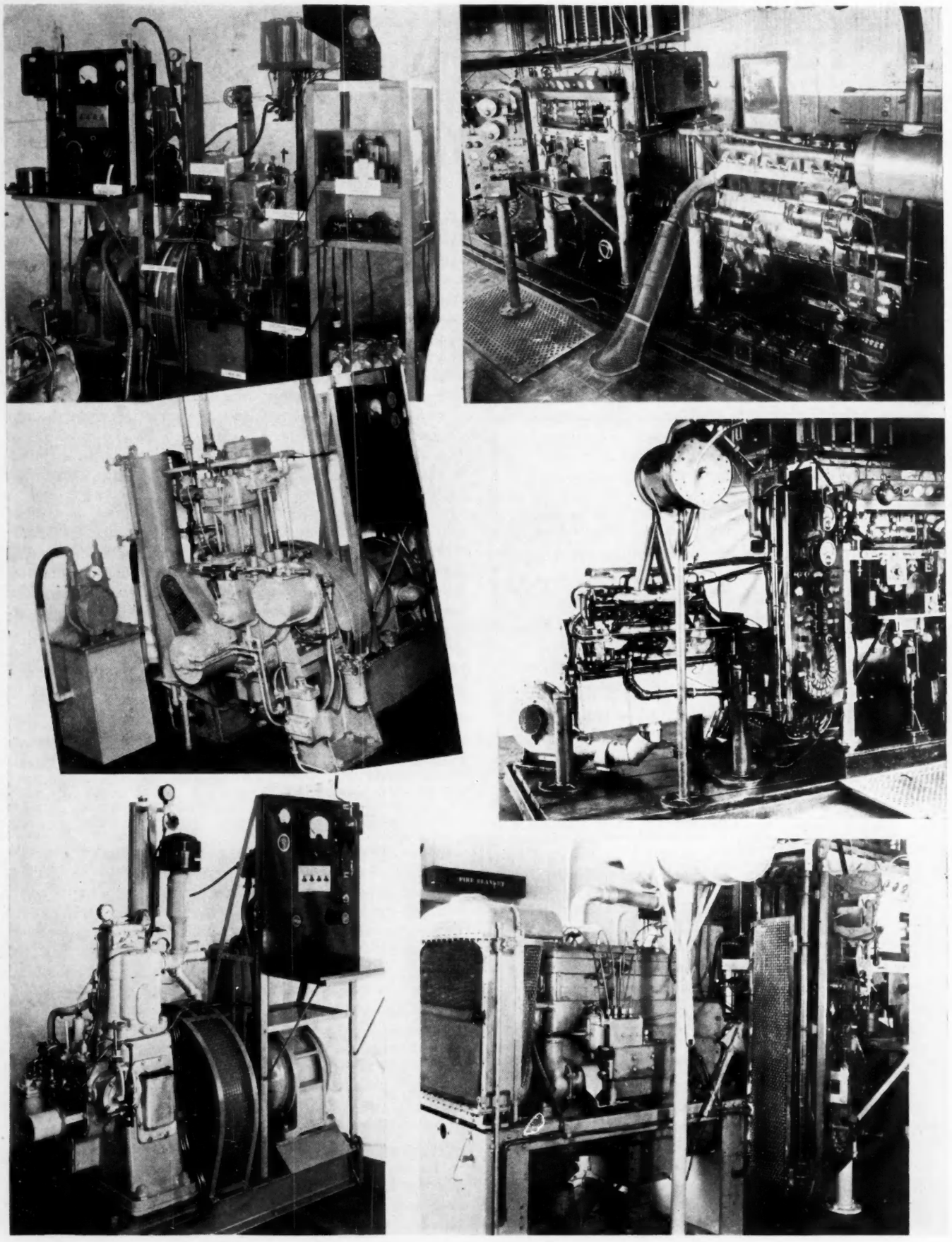


Fig. 1 – Diesel Engines Used in Laboratory Tests

change is indicated, the difference being within the range of experimental error. Likewise, a change in lubricating oil temperature from 130 deg. fahr. to 160 deg. fahr. was found to have no appreciable effect, the difference in engine deposits being less than the experimental error.

Excessive fuel quantities or over-rich mixtures are known also to increase deposits in Diesel engines. In normal sea-level operation the air-fuel ratio could not be reduced by means of increasing the fuel quantity without, at the same time, increasing the load and the combustion-chamber temperatures, both of which previously had been found to minimize the formations of deposits. Therefore, at idle load, the intake air was throttled to simulate altitude operation in an effort to obtain rich mixtures as well as low operating temperatures. A range of altitudes from sea-level to 7000 ft. was covered.

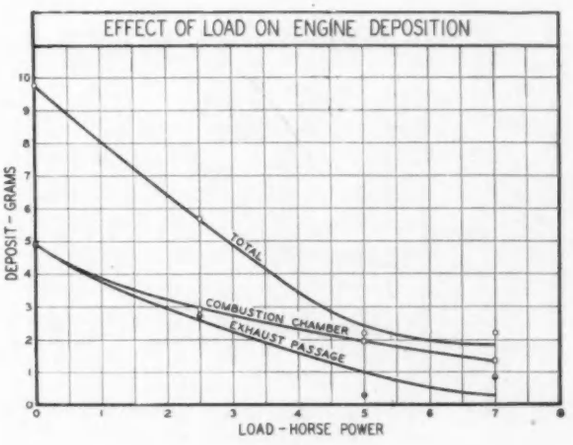


Fig. 4

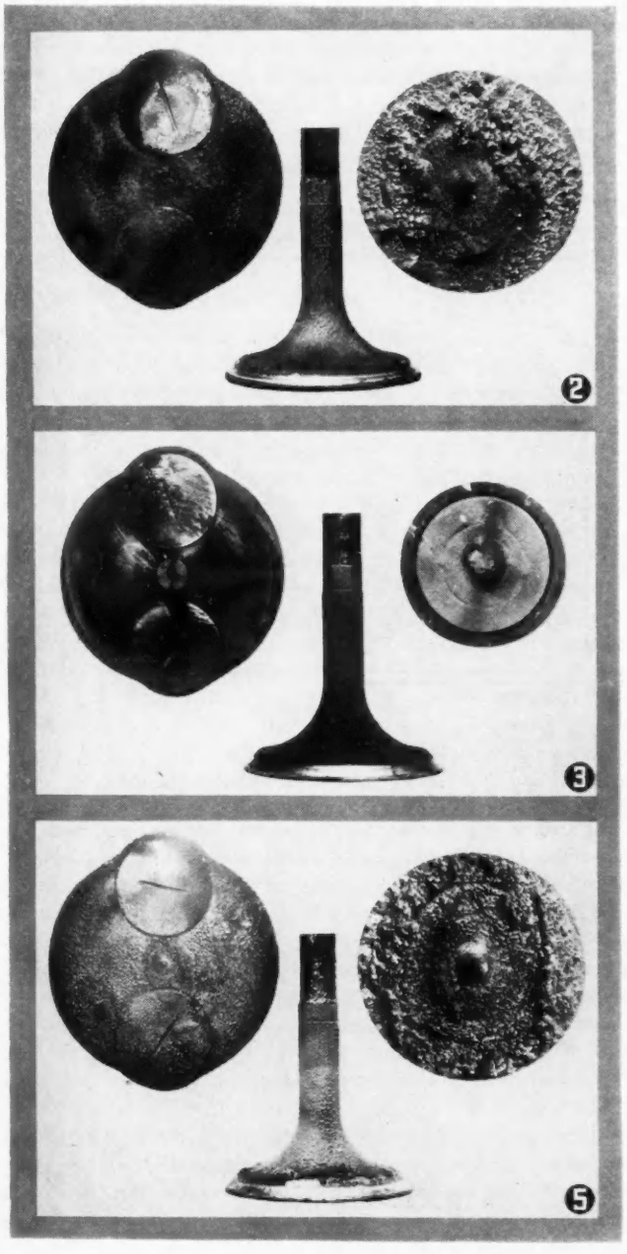


Fig. 2 – Deposits in a Direct-Injection Engine at Idle Load and Sea Level with a Typical Diesel Fuel

Fig. 3 – Deposits in a Direct-Injection Engine at Three-Fourths Load and Sea Level with a Typical Diesel Fuel

Fig. 5 – Deposits in a Direct-Injection Engine at Idle Load and 7000-Ft. Altitude with a Typical Diesel Fuel

The influence of this change is shown by a comparison of the engine parts after the 7000-ft. run in Fig. 5 with those of Fig. 2, previously mentioned. As shown by Fig. 6 the total engine deposit was approximately doubled by the simulated change of 7000 ft. in altitude. The fuel used in these tests did not cause the rings to stick under any operating conditions within the 24-hr. period, but ring-sticking was obtained in subsequent tests with certain fuels at the higher altitudes even though no ring-sticking occurred with those fuels at sea-level.

The foregoing tests indicated that large quantities of deposit could be obtained in a relatively short time by the proper choice of operating conditions. In order to determine the duration of tests required to yield significant comparisons and also as a means of extrapolating the data obtained from short-time tests, a series of runs was made with a total testing time varying from 24 to 300 hr. The results of these tests are shown in Fig. 7. The combustion-chamber deposit was substantially constant after 24 hr. while, in the exhaust passages, the deposit increased continuously with increase in running time. Apparently the high temperature and turbulence existing within the combustion-chamber caused the deposit to be removed as rapidly as it was formed after a certain point had been reached. That this point was reached before 24 hr. was verified by later tests with heavy residual-type fuels in which it was found that, unless sufficient combustion-chamber deposit was formed before 24 hr. to stop the engine, failure did not occur from this cause regardless of the length of run. In addition, these tests showed that, with residual-type fuels, and also with certain distillate fuels, the exhaust deposit was occasionally of sufficient magnitude to stop the engine. This condition occurred at a varying number of hours of running time beyond 24 hr.

From the results obtained in the preliminary investigation of operating variables it was possible to select a set of operating conditions for an accelerated test for Diesel fuel deposition. The following set of test conditions was selected:

Load	Idle
Speed	800 r.p.m.
Altitude	5000 ft.
Duration of Test	24 hr.
Jacket Temperature	150 deg. fahr.
Lubricating Oil Temperature	130 to 160 deg. fahr.

In the foregoing, the test speed of 800 r.p.m. was chosen as being the lowest possible with satisfactory governing. The altitude of 5000 ft. was determined from practical considerations which disclosed that this is the highest altitude at which Diesel engines are operated commonly. There may be some question as to the validity of tests made under the altitude idling conditions. However, there are numerous engines run-

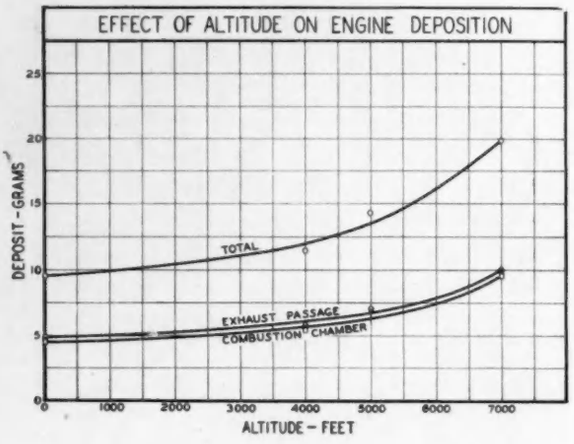


Fig. 6

ning in the field under these same operating conditions. It is known that, in several instances, tractors used in logging operations at altitudes of approximately 5000 ft. have been idled 8 hr. or more between shifts in order to overcome starting difficulties at the lower ambient temperatures encountered. It is also common practice for Diesel engines used in fishing boats to run at very light loads for long periods while trolling. It would be expected, therefore, that the results to be obtained in the laboratory service engine under the idling altitude conditions could be considered indicative of those actually obtained in the field in similar engines.

To determine if the accelerated test was sufficiently reproducible, a typical straight-run distillate fuel was retested at intervals over a four-month period. The results of these check tests are shown in Table 4.

Table 4 – Comparison of Check Tests Covering a Four-Month Period

	Deposit Weight, gm.			
	Combustion-Chamber	Nozzle	Exhaust Valve	Total
	7.22	0.27	0.53	14.45
	7.50	0.27	0.64	14.60
	7.81	0.13	0.63	15.06
	7.83	0.29	0.67	15.35
Average	7.59	0.24	0.62	14.86
Maximum Positive Deviation	0.24	0.05	0.05	0.49
Maximum Negative Deviation	0.37	0.11	0.09	0.41

Considering the magnitude of the deposits obtained, as shown by the average values, the reproducibility is considered satisfactory for this method of measurement. Further correlating tests were required, however, to determine if the differences among fuels of various physical properties, indicated by this laboratory test method, would be indicative of results obtained in other service engines under more normal operating conditions.

Fuel Properties and Deposits

Volatility is a common physical property which varies widely in marketed Diesel fuels. There have been differences of opinion as to the proper volatility range required for high-speed Diesel engines and, to determine the effect of volatility on engine deposition, various cuts of a normal fuel were selected. These special distillates were prepared to cover a range in boiling points wider than normally encountered in commercial fuels. That particular care was required in the distillation of such fuels was indicated by a test in which 1 per cent of residual-type Diesel fuel was added to a normal distillate. In this case the nozzle deposit was increased ten times and the exhaust-valve deposit was approximately doubled. This effect was avoided in the special fuels selected for the volatility test by the use of good fractionation. The distillation data for the fuels used are shown in Fig. 8 as well as the results obtained in the deposition tests. The best correlation of engine deposit with volatility is indicated by the average volatility as computed from the 20 per cent and 80 per cent A.S.T.M. distillation temperatures. The 50 per cent point provided slightly poorer correlation but, for convenience, affords a reasonably accurate criterion. The 10 per cent and 90 per cent temperatures showed no consistent relation to engine deposits.

Conradson Carbon has been used commonly as an indication of the burning qualities of Diesel fuels. An attempt, therefore, was made to correlate Conradson Carbon and engine deposits for a variety of fuels. As shown by Fig. 9, for normal distillate fuels, the Conradson Carbon values were so nearly alike and close to the minimum measurable, that this carbon-residue test failed to indicate the relative engine deposition. For residual-type fuels of the A.S.T.M. Classification No. 4 type and heavier, good agreement was indicated between Conradson Carbon and engine deposits.

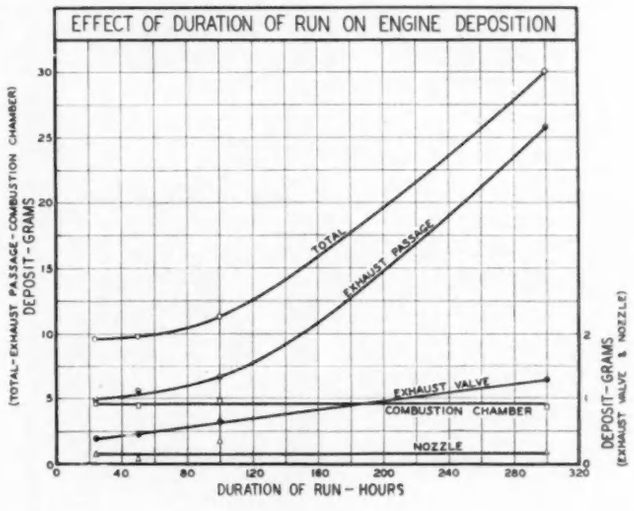


Fig. 7

The Conradson Carbon of the 10 per cent bottoms from the conventional A.S.T.M. gas oil distillation of burner fuels has been proposed as an indication of burning performance[22]. Since burner combustion may be considered somewhat analogous to Diesel-engine combustion, this test was applied to the fuels investigated for engine deposition. Fig. 9 shows that, for the normal distillate fuels or distillates with small amounts of contaminants, this carbon-residue test provided a fair indication of engine deposition. For residual fuels no correlation is apparent. Another carbon-residue test has been proposed in which the usual Conradson Carbon test procedure is followed except that, once the fuel is ignited, the heating flame is removed[21]. This method was investigated but the correlation with engine deposit for normal distillate fuels was found to be considerably poorer than the 10 per cent bottoms Conradson Carbon and, for the residual fuels, no correlation was found. Since poorer correlation was shown for all fuels, results by this method have not been included. A visual comparison of the deposits shown quantitatively in Fig. 9 may be obtained from Figs. 10, 11, and 12 which show, respectively, the deposits found with two widely different distillates and a residual fuel meeting A.S.T.M. Classification No. 4.

[22] See *National Petroleum News*, Aug. 12, 1936, pp. 41-49: "Characteristics of Fuel Oil," by E. B. Glendenning and W. A. Sullivan, Jr.

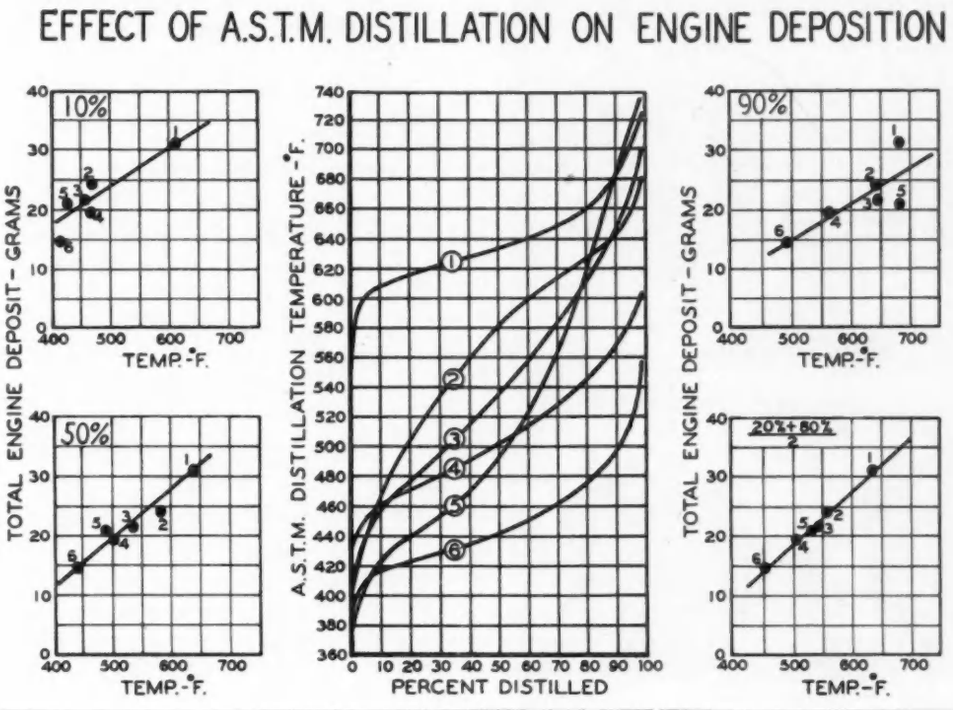


Fig. 8

There was some indication that the engine deposits with normal distillate fuels depended somewhat on the ignition quality. Tests were, therefore, made with fuels covering a range of cetane numbers. The relation shown in Fig. 13 was obtained by plotting engine deposition results versus cetane numbers as determined by the Magnetic Pickup Delay Method. A fair correlation was indicated for all the fuels tested including straight-run, cracked and doped fuels. Correlation with cetane number determined by a Critical Compression Pressure Method was found to be somewhat less exact. The results shown for doped fuels are of special interest. Two entirely different types of dopes were added to the same low-cetane fuel, and the engine deposits obtained were in close agreement with the improvement in cetane number. The deposits obtained with the low-cetane fuel were previously shown in Fig. 11 as one of the fuels also used in connection with the investigation of carbon residue. Fig. 14 illustrates the condition of the engine after running on the same fuel doped, and the comparison with Fig. 11 indicates the improvement obtained. A further indication of the effect of doping is furnished by a comparison of Fig. 14 with Fig. 10, the latter depicting the condition of the engine after running on the high-cetane number undoped fuel. The similarity between Figs. 14 and 10 is apparent.

Although ring-sticking was not as reproducible as the other phases of the tests, frequent ring-sticking occurred during the check tests with the low-cetane undoped fuel. A typical comparison of the pistons with the low- and high-cetane undoped and the doped fuel is shown in Fig. 15. The reduction in deposits with the doped fuels might have been the result of solvent action of the dopes on the deposit. However, the use of two widely different types of dopes with similar results is believed to have precluded this possibility.

An examination of Fig. 13 shows that the cracked fuels were intermediate to the extremes of the distillates. In this engine this condition appeared to be merely a function of cetane number, and the general conclusion that all cracked fuels cause greater engine deposits than straight-run fuels apparently is not justified.

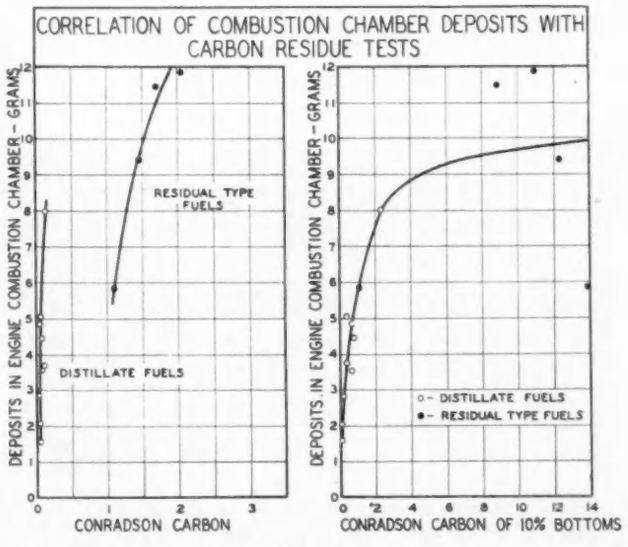


Fig. 9

Comparison with Other Service Engines

The direct-injection engine used in the laboratory tests was chosen because it was known to be critical to fuel-deposition characteristics. Tests were made, therefore, in three other popular makes of high-speed Diesel engines to determine the degree of correlation among the fuel-sensitive direct-injection engine and other service engines. Such tests, in addition to indicating the degree of correlation with the direct-injection test engine, also afforded a means of determining the relative deposition characteristics of the various makes of engines. The first engine investigated was of the precombustion-chamber type identified as Engine *B*. A variety of fuels, including straight-run and cracked distillates and residual types, was investigated. The correlation between the deposits in this engine and the test engine is shown in Fig. 16. Good correlation is indicated between the results in the two engines for the exhaust-valve deposits. Within the combustion-chamber fair correlation is indicated for the distillate fuels and poor

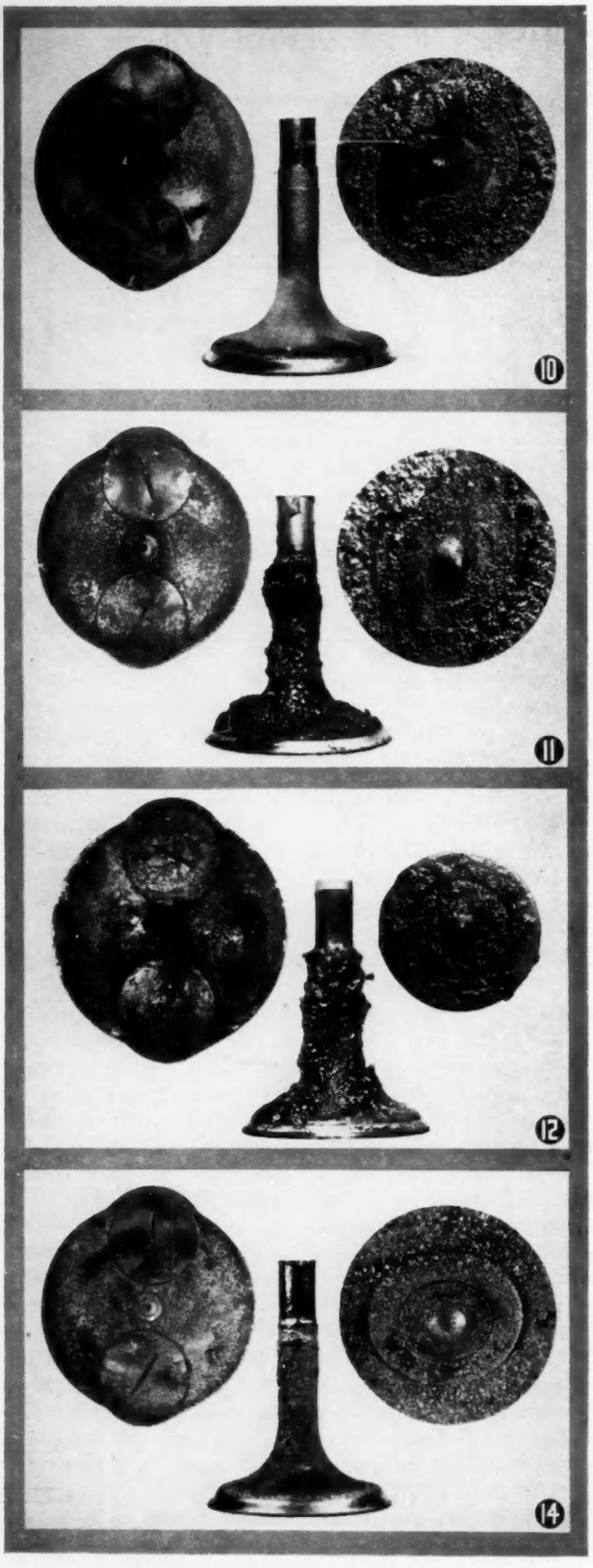


Fig. 10 – Deposits in a Direct-Injection Engine under Accelerated Conditions with Low-Carbon-Residue, High-Cetane Distillate Fuel

Fig. 11 – Deposits in a Direct-Injection Engine under Accelerated Conditions with High-Carbon-Residue, Low Cetane Distillate Fuel (Fuel *A*)

Fig. 12 – Deposits in a Direct-Injection Engine under Accelerated Conditions with No. 4 Residual-Type Fuel

Fig. 14 – Deposits in a Direct-Injection Engine under Accelerated Conditions with Low-Cetane Fuel *A* Plus Dope to Improve Cetane Number

correlation for the residual-type fuels. At the nozzle, although the correlation with distillates is good, no correlation exists for the residual fuels. This condition may be explained by the different types of nozzles employed. In the engine in which the nozzle deposits were greatest, a Bosch three-hole nozzle was used whereas, in the other engine, the Bosch pintle-type nozzle was employed. Similar operating conditions were used in the two engines, and it will be noted that the total combustion-chamber deposits are within the same range for the distillate fuels. For the residual fuels a marked increase in the deposit compared to the poorest distillate fuel was found in the direct injection Engine *A* whereas, in the pre-combustion-chamber Engine *B*, the deposit with the residual fuels was only slightly greater than for the poorest distillate fuels. The exhaust valve deposit in this Engine *B* was smaller than in the direct-injection engine for all fuels.

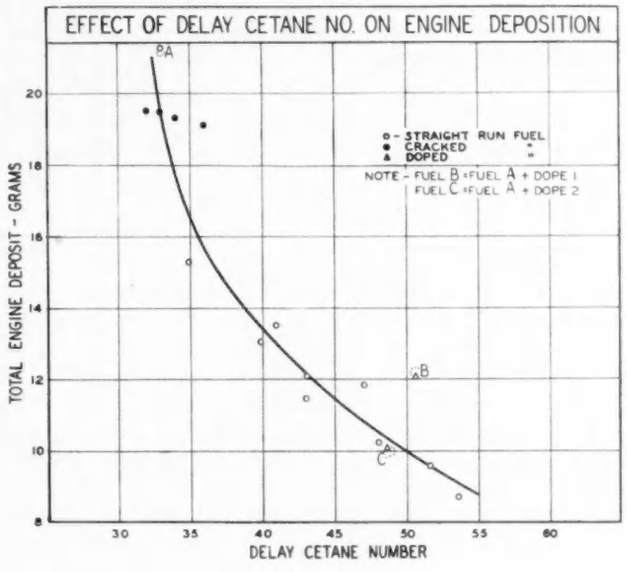


Fig. 13

The deposits formed in a popular type of precombustion Diesel engine used in tractors and identified as Engine *C*, were determined with the same two high- and low-cetane fuels used in the laboratory Engine *A*, depicted in Figs. 10 and 11. Fig. 17 shows the relative deposits with these two fuels in Engine *C* compared with the deposits of the same two fuels in the laboratory Engine *A*. One fuel formed less deposit than the other in both engines but, in Engine *C*, the total amount of deposit with either fuel was insignificant.

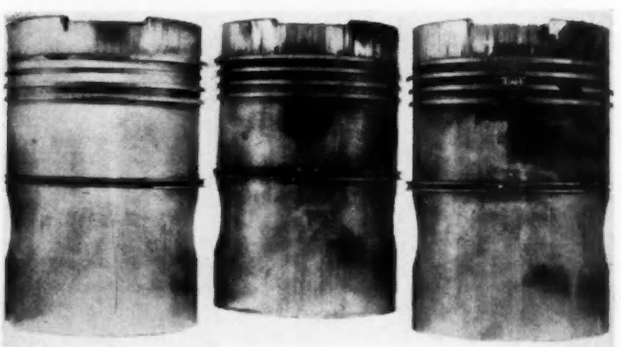


Fig. 15 – Direct-Injection Engine Pistons after Running under Accelerated Conditions with High-Cetane Fuel (*left*), Low-Cetane Fuel *A* Plus Dope (*center*), and Low-Cetane Fuel *A*

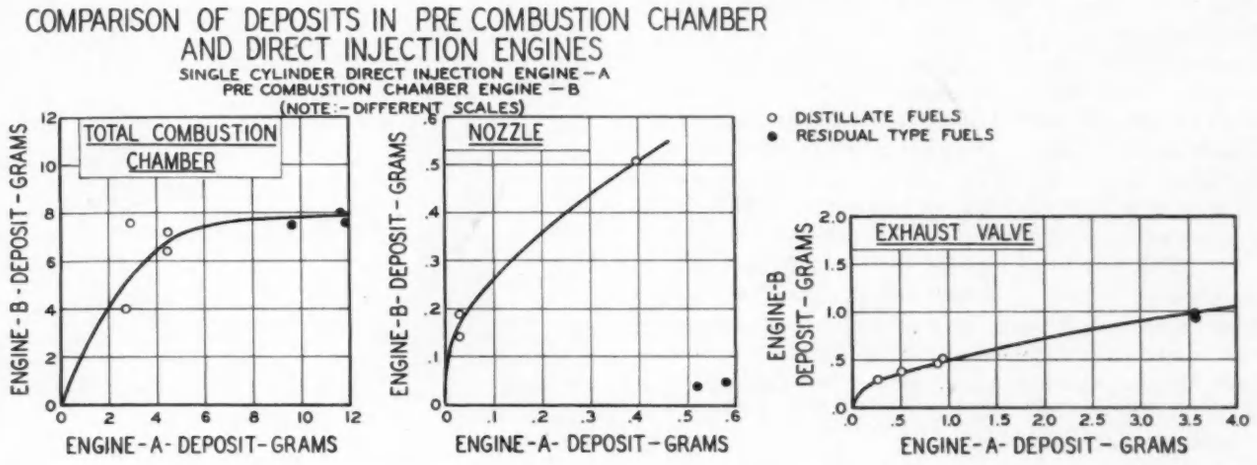


Fig. 16

The tests in Engine *C* were made incidental to another series of fuel tests in which it was necessary to operate at full load and this, at least in part, accounts for the small amount of deposit formed.

Deposition tests also were made in another popular make of tractor engine. This unit was of the precombustion-chamber type, identified as Engine *D* and operated under the following wide variety of conditions:

Continuous idling at 5000-ft. altitude.
Variable load at sea-level (alternate 5 min. idling, 10 min. full load).
Variable load at 5000 ft.
16 hr. idling followed by 8 hr. variable load at 5000 ft.

In all cases the duration of test was 100 hr. Temperatures and other adjustments were maintained in accordance with values as found in the field.

The effect of the various operating conditions on the resulting deposits when operating on the same fuels used in Engine *C* is shown graphically in Fig. 18. It is evident that in all tests the deposits were very slight, although the idling load at 5000 ft. confirmed the direct-injection engine tests in that several times more deposit was formed at altitude than with any other test condition. As previously shown by laboratory engine tests, exhaust deposits increased in a linear

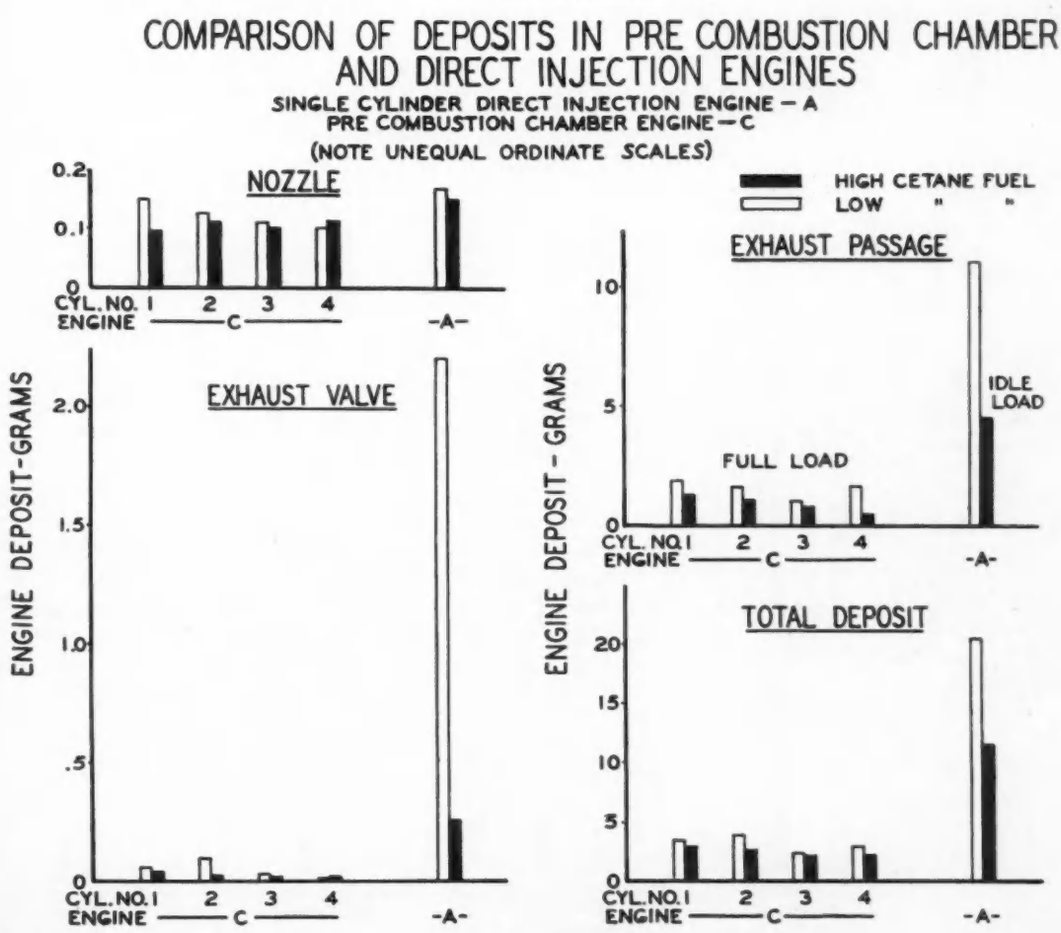


Fig. 17

relation with time. Therefore, the comparison of exhaust valves shown by Fig. 19 is of particular significance since it shows that the exhaust-valve deposit in the laboratory engine was some 20 times that in the precombustion-chamber Engine *D*, even though the latter engine was run for 100 hr. whereas the direct-injection Engine *A* was operated for only 24 hr.

In all tests in precombustion-chamber Engine *D* the deposits were so minute that there was no measurable difference between the two fuels. A comparison of the extreme differences between the deposits resulting from the use of the two fuels in direct injection Engine *A*, shown in Fig. 17 with the minor differences between deposits from the same fuels, run under similar conditions and shown in Fig. 18, illustrates the wide control that the engine designer has over fuel deposition not only in relative amounts between two fuels but also in the total deposition with any one fuel.

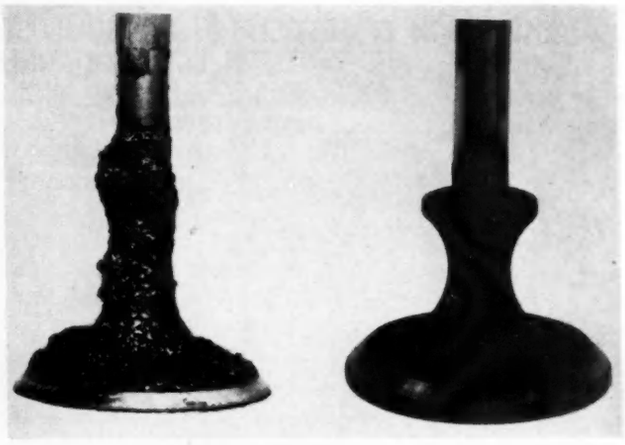


Fig. 19 – Comparison of Exhaust Valves from Direct-Injection Engine after 24 Hr. Running (*left*) and from Precombustion-Chamber Engine after 100 Hr. Running – Both Engines Were Operated under Accelerated Conditions

Conclusions

In summary it may be concluded that, in those Diesel engines which are critical to fuel deposition, a decrease in operating temperature and load or an increase in altitude increases the quantity of deposits formed. In these engines increasing the average volatility or the cetane number of the fuel decreases the deposit.

An approximation of the deposition tendencies of distillate fuels in engines which form appreciable amounts of deposits may be obtained from the Conradson Carbon determination of the 10 per cent bottoms remaining after 90 per cent has been distilled off in the conventional A.S.T.M. gas oil distillation. For fuels of this type the usual Conradson Carbon test fails to correlate with engine deposits. For residual-type fuels the Conradson Carbon test provides a convenient means of predicting engine deposition characteristics, whereas the Conradson Carbon of the A.S.T.M. distillation 10 per cent bottoms is valueless.

In those engines in which fuel deposition is slight no consistent difference between fuels and no correlation with physical properties of the fuels are apparent.

These tests covering a wide variety of engine designs, fuels, and operating conditions have demonstrated that the engine designer has greater control over the quantity of deposits formed in the engine than either the fuel refiner or engine operator, as indicated by the test results presented in which the ratios of deposits were 20:1 between extremes in engine design, 10:1 in operating conditions, and 3:1 between extremes in distillate fuels.

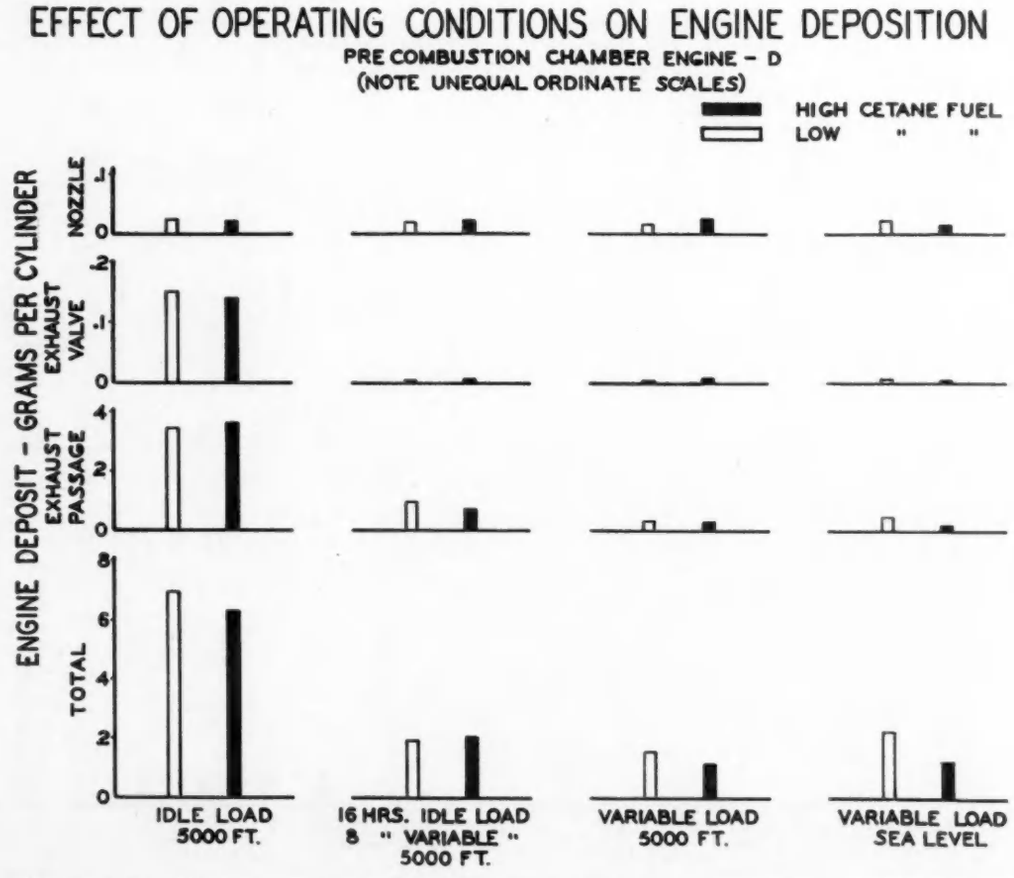


Fig. 18

Maintenance of a Scattered Fleet

By H. O. Mathews

Automotive Engineer, Public Utility Engineering & Service Corp.

MANY differences and difficulties are encountered in the maintenance of a scattered fleet. A scattered fleet consists of cars located in groups too small to warrant the establishment of a company garage and company employees to service and repair them. In our particular operation and for the purposes of discussion in this paper, this number of cars is considered to be less than 10 in any one location. Many fleet owners throughout the country are faced with this problem, and many have solved it satisfactorily to themselves. It is hoped that this paper will provoke discussion which may bring to light better methods of maintaining these scattered vehicles.

In my opinion the maintenance of this type of fleet must be tied closely to the dealers and the manufacturers of the particular vehicle which is to be serviced. The peculiar thing, and a natural one, is that the best dealers' service is in the cities and towns where company garages are maintained. I have secured a picture of some representative dealers' service stations in small towns.

(1) Although the appearance from the outside is fair, there is a dirt floor in all but the show room.

(2) A good agency in a town of 2200 population.

(3) An agency in a town of 1500. The condition of the inside of the garage is terrible.

(4) A service station in a town of 500.

These are fairly representative of the dealers' service stations in the parts of the country where this type of service is necessary. Certainly you would not want your private car serviced in some of these stations, and you would question the service received on a company vehicle.

Traveling Inspector Costly

It will be argued by many that the most economical maintenance of this type of fleet is by means of a traveling inspector who is a combination inspector, serviceman, and mechanic. This inspector spends his entire time inspecting and repairing scattered vehicles. To this plan I cannot agree. The first requirement for the inspector is a means of transportation, which will cost at least $50 per month. The next requirement is a complete set of tools, gages, and so on, in which the investment will be approximately $500. Next there is the time required for the inspector to travel from job to job unless, of course, the company is fortunate enough to have an 80-hr. per week man. This 80 hr. is not a guess but represents the actual working hours of some of our traveling inspectors. The average wage for the inspector would be approximately $200 per month, and his expenses while away from home would be at least $2.50 per day. The total cost of the traveling inspector would, therefore, be approximately $300 per month plus the depreciation on the tools and equipment that he uses. In order to make this summary clear, Table 1 has been prepared.

Table 1 – Cost of Traveling Inspector-Repairman

(1) Transportation (average per month)	$ 50.00
(2) Salary	200.00
(3) Expenses away from headquarters	50.00
(4) Tool expenses including depreciation	10.00
Total	$310.00
Number of Vehicles Inspected	100
Cost per vehicle	3.10

The amount of productive work which this inspector can do in a month warrants some discussion. In my opinion he should inspect and make the minor repairs on 100 scattered

Table 2 – Preventive Maintenance Plan Involving Dealer Service Developed by Manufacturer

July, 1936

GENERAL MOTORS FLEET USER'S PREVENTIVE MAINTENANCE ORDER

BUICK—CHEVROLET—OLDSMOBILE—PONTIAC PASSENGER CARS AND CHEVROLET TRUCKS

Operation No. 1—General Inspection and Lubrication Every 1000 Miles

Car No. ..... Dealer Repair Order No. ..... Date .....

Fleet User ..... Car Serial No. .....

Address ..... Engine Number .....

Operator ..... Mileage .....

Name of Dealer ..... Address .....

NOTICE: NO WORK OTHER THAN THAT LISTED HEREON IS TO BE PERFORMED WITHOUT SPECIFIC AUTHORIZATION

√—IF O. K. X—ADJUSTMENTS MADE O—REPAIRS DESIRABLE

1. LUBRICATE CAR—(BY EXPERIENCED MAN)
2. CHANGE ENGINE OIL—(if ordered). *Engine Hot*
3. CLEAN AIR CLEANER—(Fill oil bath type to proper level with SAE 50 OIL)

WHILE LUBRICATING VEHICLE—THOROUGHLY INSPECT THE FOLLOWING—NO CHARGE EXCEPT FOR EXTRA LUBRICANT USED. ANY WORK ORDERED PERFORMED AS A RESULT OF THIS INSPECTION WILL BE CHARGED FOR AT REGULAR FLAT RATE.

1. CHECK STEERING GEAR oil level. Add lubricant if necessary.
2. INSPECT WATER PUMP PACKING NUT adjustment, and HOSE CONNECTIONS. CHECK RADIATOR for leaks.
3. INSPECT FAN BELT adjustment and lubricate WATER PUMP BEARINGS.
4. CHECK all LIGHTS and HORN.
5. INSPECT BATTERY and fill to proper level.
6. CHECK CLUTCH PEDAL adjustment.
7. CHECK to see if GASOLINE GAUGE is recording.
8. CHECK REAR VIEW MIRROR.
9. CHECK STARTER to see if it operates properly.
10. CHECK CHOKE and set CARBURETOR for proper idling speed.
11. CHECK WINDSHIELD WIPER.
12. CHECK to see if OIL PRESSURE is up to standard.
13. CHECK GENERATOR charging rate.
14. CHECK STEERING GEAR for adjustment.
15. INSPECT for SPRING SHACKLE adjustment.
16. CHECK FRONT WHEEL BEARINGS for adjustment.
17. CHECK TIRE AIR PRESSURE and inflate as recommended.
18. CHECK TRANSMISSION oil level. Add lubricant if necessary. Extra charge for lubricant.
19. FILL UNIVERSAL JOINTS. (Chevrolet 1½-Ton Truck)
20. CHECK DIFFERENTIAL oil level. Add lubricant if necessary. Extra charge for lubricant. E. P. Lubricant for Oldsmobile.
21. CHECK KNEE ACTION oil level. Add lubricant if necessary. (Chev. Master and Pontiac)
22. CHECK REAR WHEEL BEARINGS for adjustment. (1936 Chevrolet 1½-Ton Trucks)
23. TEST EMERGENCY and SERVICE BRAKES during road test.
24. CHECK reading of HEAT INDICATOR.
25. INDICATE condition of FENDERS ..... BODY .....

PAINT ..... DECALCOMANIAS .....

This Inspection Indicates That The Following Repairs Are Desirable

Description of Suggested Repairs	Estimated Cost		
	Parts	Labor	Total

Service Manager

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1938.]

Table 3 – Preventive Maintenance Plan Involving Dealer Service Developed by Manufacturer

July, 1936

GENERAL MOTORS FLEET USER'S PREVENTIVE MAINTENANCE ORDER

BUICK — CHEVROLET — OLDSMOBILE — PONTIAC PASSENGER CARS AND CHEVROLET TRUCKS

Operation No. 2—General Inspection, Adjustment and Lubrication Every 5000 Miles

Car No..... Dealer Repair Order No..... Date.....
Fleet User..... Car Serial No.....
Address..... Engine Number.....
Operator..... Mileage.....
Name of Dealer..... Address.....

NOTICE: NO WORK OTHER THAN THAT LISTED HEREON IS TO BE PERFORMED WITHOUT SPECIFIC AUTHORIZATION

√—IF O. K. X—ADJUSTMENTS MADE O—REPAIRS DESIRABLE

A—TUNE ENGINE—Which Includes the Following Inspection and Adjustments.

1.	CHECK, CLEAN, and adjust SPARK PLUGS	
2.	CHECK, SQUARE, LUBRICATE and SET DISTRIBUTOR POINTS. FILL DISTRIBUTOR SHAFT GREASE CUP	
3.	CHECK and CLEAN DISTRIBUTOR CAP	
4.	INSPECT IGNITION WIRING and seat wires in coil and cap securely	
5.	SET OCTANE SELECTOR AT 0°—(Except Olds)	
6.	CLEAN sediment bulb on FUEL PUMP	
7.	TIGHTEN FUEL PUMP lines. (FUEL PUMP should be overhauled every 15,000 miles at extra charge)	
8.	ADJUST WATER PUMP PACKING NUT, and LUBRICATE PUMP BEARINGS	
9.	TIGHTEN HOSE CONNECTIONS and CHECK RADIATOR for leaks	
10.	CHECK and ADJUST FAN BELT	
11.	CLEAN and service AIR CLEANER—(Fill oil bath type to proper level with SAE 50 OIL) (CHECK CRANK CASE VENTILATOR unit—Pontiac and Olds)	
12.	INSPECT CARBURETOR and set float level and choke adjustment	
13.	ADJUST METERING ROD setting by using metering rod gauge—Chevrolet—1936 Olds—Pontiac (CHECK ANTI-PERCOLATOR—1936 Olds and Pontiac)	
14.	CHECK STARTER	
	The following operations should be performed with engine hot and running.	
15.	TIGHTEN CYLINDER HEAD BOLTS	
16.	TIGHTEN MANIFOLD BOLTS and EXHAUST PACKING NUTS	
17.	CHECK MANIFOLD HEAT CONTROL—(Adjust for season—Pontiac)	
18.	SET ENGINE IGNITION TIMING	
19.	ADJUST VALVE TAPPETS HOT—engine should be hot and running	
20.	ADJUST CARBURETOR. Use VACUUM GAUGE when possible. Adjust for proper idling speed—(check and set automatic throttle opening—Olds)	
21.	CHECK and ADJUST CLUTCH PEDAL for proper clearance. LUBRICATE THROW-OUT BEARINGS. (Except Pontiac and Olds)	
22.	CHECK reading of HEAT INDICATOR	
23.	CHECK to see if OIL PRESSURE is up to standard	
24.	CHECK GENERATOR and adjust charging rate	
25.	CHECK WINDSHIELD WIPER	
26.	CHECK to see if GASOLINE GAUGE IS RECORDING	
27.	CHECK MANUAL CHOKE—(except Olds)	
28.	CHECK ALL LIGHTS. (Does not include focusing)	

B—GENERAL TIGHTEN-UP—Which Includes the Following Inspection and Adjustments.

1.	CHECK and ADJUST STEERING GEAR	
2.	CHECK and ADJUST DRAG LINK and FRONT AXLE TIE ROD CONNECTIONS and PITMAN ARM	
3.	CHECK FRONT WHEEL BEARINGS. (Remove—clean and inspect. Lubricate—see Lubrication Chart)	
4.	TIGHTEN RADIATOR and FRONT FENDER SUPPORT BOLTS	
5.	TIGHTEN FRONT WHEEL HUB BOLTS	
6.	TIGHTEN FRONT BUMPER BOLTS	
7.	TIGHTEN FRONT ENGINE SUPPORT BOLTS—(Except Olds)	
8.	CHECK KING PINS	
9.	CHECK KNEE ACTION and RADIUS RODS for looseness. (Chevrolet and Pontiac)	
10.	ON CONVENTIONAL AXLE vehicle—INSPECT AND TIGHTEN FRONT SPRINGS—SHACKLES and U BOLTS	
11.	TIGHTEN ENGINE REAR MOUNTINGS	
12.	TIGHTEN RUNNING BOARD BOLTS	
13.	TIGHTEN FRONT FLOOR BOARDS	
14.	ADJUST BODY or CAB HOLD DOWN BOLTS	
15.	TIGHTEN UNIVERSAL JOINT BALL CAP BOLTS. (ADJUST IF NECESSARY.) (Except Olds)	
16.	TIGHTEN REAR FENDER BOLTS	
17.	TIGHTEN REAR WHEEL AXLE NUTS and HUB BOLTS	
18.	TIGHTEN MUFFLER SUPPORT BOLTS	
19.	TIGHTEN BATTERY CLAMP BOLTS	
20.	CHECK and ADJUST REAR SPRING SHACKLES and TIGHTEN REAR SPRING U BOLTS	
21.	CHECK REAR VIEW MIRROR	
22.	CHECK operation of WINDOW REGULATORS and DOOR LOCKS	
23.	CHECK TIRE air pressures and inflate as recommended	
24.	TEST EMERGENCY and SERVICE BRAKE adjustment during road test	

C—GENERAL LUBRICATION—Which Includes the Following Inspections. EXTRA CHARGE FOR LUBRICANT ADDED.

1.	LUBRICATE CAR—(BY EXPERIENCED MAN)	
2.	CHANGE ENGINE OIL—(if ordered). *Engine Hot*	
3.	CHECK and fill KNEE ACTION to proper level. (Chev. Master and Pontiac)	
4.	CHECK and fill TRANSMISSION and DIFFERENTIAL to proper level. (Change every 10,000 miles in addition to seasonal changes)	
5.	LUBRICATE REAR WHEEL BEARINGS—(1936 Chevrolet 1½ ton—10,000 miles at extra charge)	
6.	LUBRICATE all DOOR and TRUNK HINGES, BUMPERS, LOCKS and HANDLES	
7.	CHECK and fill STORAGE BATTERY. INSPECT and TIGHTEN TERMINALS	
8.	CHECK BRAKE MASTER CYLINDER for fluid level. Add fluid if necessary	
9.	FRONT SPRING COVERS SHOULD BE LUBRICATED each 10,000 miles at extra charge	
10.	REAR SPRING COVERS SHOULD BE LUBRICATED each 10,000 miles at extra charge	

THIS INSPECTION INDICATES THAT THE FOLLOWING REPAIRS ARE DESIRABLE

Description of Suggested Repairs	Estimated Cost		
	Parts	Labor	Total

.....
Service Manager

Table 4 – 5000-Mile Plan Developed by Another Manufacturer

5000 MILE GENERAL INSPECTION, ADJUSTMENT AND LUBRICATION

Dealer Repair Order No. . . . . Date .....

Car No .....
Engine No .....
Mileage .....
Fleet Owner .....
Name of Dealer .....

Operator.....
Address .....
Address .....

NOTE: No work other than that listed below is to be performed without specific authorization.

O. K. If O.K. X Adjustment made R. R. Repairs recommended

Gas Mileage Test at 30 M.P.H.
Miles per hour before adjustments_____
" " " after "_____
MOTOR Tune-up with Ford Lab.Test Set

1. Test compression and report condition of rings and valves, each cylinder .....
2. Clean, adjust and test spark plugs.....
3. Test Battery.....
4. Test battery connections, clean and tighten if required
5. Test primary circuit-replace or respace points if required
6. Test coil.....
7. Test high tension wires.....
8. Test distributor caps.....
9. Test terminal plates.....
10. Test distributor rotor.....
11. Test condensor.....
12. Test generator and adjust...
13. Test lights.....
14. Test horns.....
15. Test fuel pump and clean screen.....
16. Check fuel line for obstruction
17. Drain sediment from fuel pump
18. Check carburetor float for proper setting.....
19. Adjust carburetor to proper idle
20. Check distr. vacuum brake....
21. Check oil pressure gauge.....

Chassis Adjustments

1. Check brake and clutch pedals (pedal movement, adjust if necessary).....
2. Inflate tires to proper pressure
3. Adjust brakes.....

General Tightening-up

*1. Tighten cylinder head bolts
*2. Tighten manifold bolts.....
3. " fender and running board bolts.....
4. Tighten wheel hub bolt nuts
5. Check spring shackles.....
6. Tighten muffler brkt. bolts
7. Tighten front and rear spring U bolts and spring clips..
8. Check and adjust steering gear.....
9. Check steering connection rod.....
10. Check front wheels for toe-in.....
11. Check front wheel bearings, (remove-clean and inspect. Lubricate-see Lubrication)
12. Check spindles, spindle bolts, and bushings .....
13. Tighten body bolts.....
14. " Oil pan bolts.....
15. " and adjust motor mounts.....
16. Tighten bumper bolts.....
17. " battery clamp bolt
18. " front floor boards
19. " radiator bolts....
20. " " hose clamp bolts and check for leaks.....
21. Check water pumps.....
22. Adjust fan belt for proper tension.....
23. Check rear view mirror....
24. " operation of window regulators and door locks
25. Check windshield wiper...
26. " heat and fuel indicator

(*-Tighten with engine hot and running)

Lubrication

1. Check and change engine oil if necessary. (Extra charge for oil).....
2. Lubricate Universal joint with Universal Joint Grease.....
3. Lubricate with pressure gun gease all other pressure gun fittings(except springs)
4. Lubricate springs (use M-4628 in springs having lubrication fittings).....
5. Oil all parts designed in Lubrication chart.....
6. Wax hood lacings, door stricker plates, and dove tails.....
7. Lubricate door hinge pins.....
8. Replenish gear oil in steering gear.....
9. Drain, flush and refill transmission and rear axle with correct grade of gear oil for approaching season (extra charge for lubricate).....
10. Apply special "FORD" distributor grease to distributor cam.....
11. Lubricate and adjust front wheel bearings using short fiber sodium soap grease having high melting point.....
12. Replenish fluid in shock absorbers.....
13. Clean air cleaner and silencer.....

Repairs recommended are as listed below

Description of Repair	Estimated Cost Parts	Labor

. . . . . .
Service Manager

vehicles once each month or six weeks. Although this amount may be subject to criticism, I believe it will represent the average. The cost, therefore, is over $3 per vehicle per inspection. If much repair work is necessary the cost, of course, will be greater. This amount appears to be too high for economical maintenance of a fleet of 100 vehicles.

Another point against this plan is that our experience has been that the inspector is in the wrong place when trouble occurs and additional traveling time is necessary to get him to the job to make the repairs.

The foregoing data have been presented as an illustration of the cost of this particular plan of maintenance. Now let us look at another possibility. Some time ago one of the manufacturers developed a preventive maintenance plan which involves the dealers' service (Tables 2 and 3). This company is to be congratulated on this development. In my opinion it is a large step in the right direction. The trouble is that again this service is available in the larger cities where company garages are maintained, whereas the dealer in the small town is not familiar with the plan, is not interested in it, and is not prepared to carry it out. This statement may be challenged, but I am prepared to offer the proof for it. Table 4 indicates a plan developed by another manufacturer which is deserving of merit. This form is used on a 5000-mile basis on a fleet which has no company garages and must, therefore, depend upon dealer service for its complete maintenance.

1000-Mile Inspection

Table 5 is a 1000-mile inspection plan which has been developed for use in a fleet in which the cars are covered by a company inspection each 5000 miles or semi-annually, but the other work is performed at dealers' service stations.

From the charts it will be noted that much has been done to develop economical inspection methods for scattered fleets; yet the condition of many vehicles in various parts of the country indicates the need for further improvement in the methods.

These scattered fleets seem to fall into two classes: first, those which depend for all of their maintenance upon public garages and, second, those which receive their major inspections and work in a company garage at local or general headquarters. The problem of preventive maintenance is the same in both cases as it is merely a matter of who does the major work. It then reverts back to the local service station manager who is usually not interested in the problems of the fleet owner unless it means greater revenue to him.

Obtaining a Satisfactory Plan

It seems, therefore, that the job of providing a satisfactory preventive maintenance plan rests jointly with the owner, the dealer, and the manufacturer. It was with this thought that Table 5 was developed. The manufacturer has checked the labor required for normal checking and adjusting in addition to the lubrication, and reports that an average of 1¼ hr. will be sufficient to care for the items included on the form. The manufacturer further states that it will be necessary to sell the dealers involved on this plan, that is, they must be convinced that the revenue will be sufficient to cover the time involved and that it is to their best interest to help keep the maintenance costs at a minimum on these cars.

I believe that this plan is possible with the aid of the manufacturer, and several companies are developing similar plans at the present time. The cost of this plan would be an average of $2 per inspection which is considerably less than that of the traveling inspector. These inspection forms should be forwarded to the company headquarters for checking and for the authorization of additional work resulting from the inspection.

Table 5 – 1000-Mile Plan for Use in Conjunction with 5000-Mile Company Inspection

1000 MILE PREVENTIVE MAINTENANCE INSPECTION

Dealers Service Station

Each 1000 Miles or Thirty Days

Car No..... Make and Year of Car.....

Division..... Speedometer Reading.....

Driver..... Name of Dealer.....

The following items are to be checked and the necessary adjustments made by the Service Station unless otherwise noted each 1000 miles or thirty days.

1. Lubricate car using manufacturers' chart.
2. Check steering gear, oil level. Add lubricant if necessary.
3. Check transmission and differential oil level.
4. Inspect water pump packing, nut and hose connections, check radiator for leaks.
5. Inspect fan belt adjustment.
6. Check clutch pedal for proper adjustment.
7. Check generator charging rate.
8. Check front wheel bearings.
9. Check motor idling speed and carburetor adjustment.
10. Check steering gear for adjustment.
11. Check brake adjustments - make road test.
12. Check battery for water level, corrosion and tight connections.
13. Check tires for proper inflation, bad cuts, bruises, nails, etc.
14. Check water and Antifreeze in cooling system.
15. Check and report damage to body, fenders, paint, etc.

Suggested repairs needed..... Estimated Costs.....

....

....

....

Service Manager.....

We have attempted to make the driver responsible for this 1000-mile inspection, but this plan has not been successful. In the public-utility business the motor vehicle is considered to be the means of getting to the job. The driver is a workman and is usually responsible for the operation and maintenance of the plant in his territory and, therefore, the car expense is the least important and receives the least attention. We feel that a plan similar to this one should, therefore, be developed to make the maintenance of his car as easy as possible and yet provide for the lowest possible costs.

The major inspection and maintenance work may or may not be performed at the dealers' service station. The cost will be very little different on these major items, but a better job probably would be obtained in a company garage if there is one within the territory. The main difficulty with taking the car to headquarters is the loss of time, extra mileage and, perhaps, the need of a car for replacement purposes while the inspection is being completed. Arrangements can be made with the majority of dealers in the outlying territories to do the work in the evening, thus permitting full use of the vehicle for company business.

Conclusion

In conclusion I believe that: first, the manufacturer must educate the small-town dealers who are involved in fleet maintenance; second, the fleet operator must cooperate in this educational program to the end that the dealer understands the problem; and, third, the dealer must be paid a fair amount for the work performed. The driver must assume the responsibility for the regularity of the inspections, and no deviations from a standard inspection form should be allowed. If these suggestions are followed, I believe that better results and lower costs will be obtained.

Propeller Problems Imposed by Substratosphere Flight

By Carl F. Baker

Hamilton Standard Propellers, Division of United Aircraft Corp.

THIS paper points out that high-power substratosphere flight will require a close coordination of airplane, engine, and propeller designers in order to insure the most efficient conversion of power. If this coordination is not done, substantial penalties in propeller performance will be obtained.

Substratosphere flight with high powers is going to require propellers which are appreciably heavier than present ones. However, although weights will increase, it is expected that the upward trend in specific weights can be reduced.

THE demand for larger airplanes of the transport and bomber class for long-range operation and for faster small planes of the fighter class means not only increased powers with adequate propellers, but also, because of the increasing importance of obtaining the maximum overall efficiency, careful coordination between airplane, engine, and propeller designers to insure the best combination of the components.

Large multiengine and fast single-engine airplane design is requiring the development of powerplants having greatly increased capacities over what was considered high power a few years ago, and the demand is to have these powers available at high altitude. This concentration of large amounts of power in single units requires, of course, propellers of greatly increased power-absorption characteristics. Although propellers of the present sizes can, by increased blade angles of attack, absorb powers greatly in excess of those for which they are now employed, this increase is accomplished only at a great sacrifice in efficiency. The only way that increased powers can be absorbed satisfactorily is by providing corresponding increases in propeller blade area.

The basic efficiency of a propeller on any application is determined by the maximum diameter permitted by the aircraft structure and the gear ratio provided in the engine. If the airplane does not allow for a sufficiently large diameter, or the engine gear ratio is not correct, or both, a penalty is imposed on the airplane performance either through reduced propeller efficiency or through increased propeller weight, which penalty results from the use of a propeller of increased number of blades.

It is appreciated that there are basic structural and weight problems which must be considered by the airplane designer in providing accommodations for the propeller. Similarly, there are design and weight problems encountered by the powerplant engineers in providing proper propeller reduction gearing. However, the satisfactory solution to the propeller problem depends on the complete cooperation among the three engineering agencies so that the best compromise among the factors involved can be obtained.

In the early days of aviation very little consideration was given to the propeller in the design of an airplane. This condition was probably because the variations in the engine powers and airplane performances were very small and, hence, propeller requirements did not change appreciably from one airplane to another. This lack of consideration is indicated in the old designs by the fact that they most all provided for 9-ft. diameter propellers. This attitude has changed appreciably but, basically, it still persists to some extent.

The effect which such an attitude would have on the performance of airplanes currently being designed is indicated in Fig. 1. This figure illustrates the loss in propeller performance resulting from inadequate provision for propeller diameter in the airplane and from employment of the improper reduction-gear ratio in the engine. The airplane for this example was assumed to be of the transport class and to have a basic performance of 270 m.p.h. maximum speed at 15,000 ft. The design was considered to limit the diameter to 12 ft., 0 in. The engine rating was taken as 1500 hp. at 2400 r.p.m. at 15,000 ft., and the gear ratio was assumed to be 0.500:1.

Propellers were selected on the basis of the maximum speed condition so that the best diameters without tip speed loss would be obtained with the limiting conditions of diameter and gear ratio both in force, each in force, and neither in force. The blade designs were assumed to be of conventional shape and geometrically similar. To indicate the effect of employing different propeller types, four-bladed as well as three-bladed propellers were considered. Performances were determined for operation at the engine rating over a speed range of 150 to 270 m.p.h.

The improvement obtained by proper coordination of airplane, engine, and propeller design is readily apparent. An increase in thrust horsepower of about 14 per cent over the entire speed range is obtainable with the three-bladed type. The improvement with the four-bladed propellers is somewhat less as, for the restricted diameter case, a given diameter with this type always more closely approximates that required to absorb a given power than the same diameter with a three-bladed one. It will be also noted that, for the maximum-speed case, the performance with the best four-bladed propeller does not quite equal that of the three-bladed one and is substantially less at slower speeds. This effect is characteristic of an increase in the number of blades.

A comparison of the two types of propellers for the cases of restricted diameter with fixed and optional gear ratios shows that appreciable performance gains are available with

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., Mar. 10, 1938.]

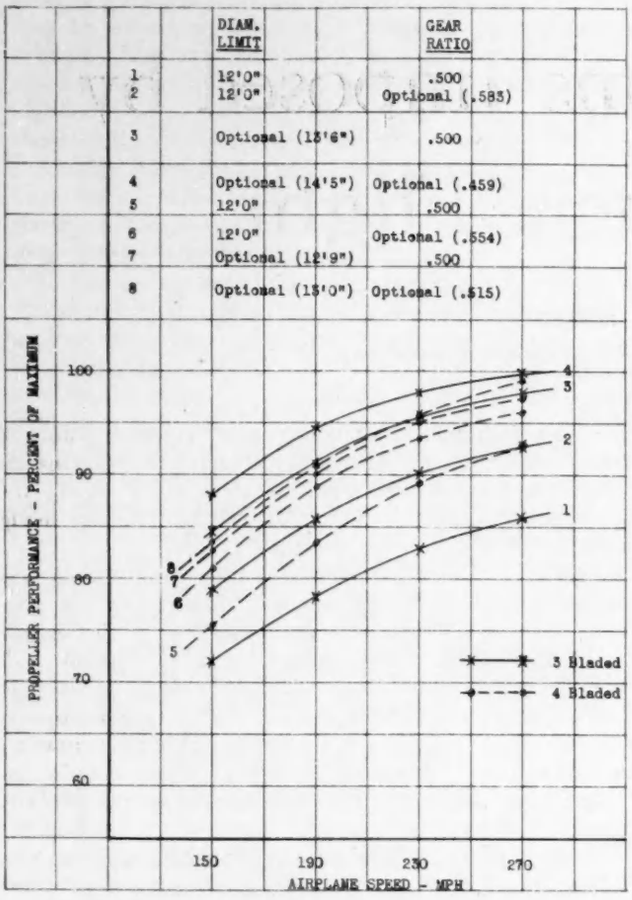


Fig. 1 – Propeller Performance at Various Airplane Speeds with Different Diameters and Gear Ratios

the four-bladed propeller. However, it must be remembered that changing from the three-bladed propeller to the four-bladed one means an increase in propeller weight on the order of 33 per cent. The effect on airplane performance of this weight increase must be balanced against the improved propeller efficiency.

This study emphasizes the fact that careful coordination of the divergent design factors affecting the propeller is more than ever before necessary in order to insure that the optimum performance will be obtained with airplanes of the future. With this thought in mind, we can now consider the effect of future aircraft performance demands on the propeller.

The requirements for propellers for substratosphere operation at high powers are divided into two categories by basic airplane types. The airplanes of the transport or bomber class will be able, because of their size, to accommodate relatively large diameter propellers. Single-engine airplanes of the fighter class, on the other hand, because of necessary landing-gear limitations, will require propellers of relatively small size. These two airplane classes require separate consideration.

The effect of increased power on propeller design is to require increased total blade area. Such an increase of area can be accomplished either by an increase in the propeller diameter for a given number of blades, or by an increase in the number of blades for a given diameter. If it were a matter only of providing propellers to absorb the powers at which engines of the future will be rated, the problem would be relatively simple since the increase in rated powers alone would not require a large increase in total blade area. However, two other factors enter the problem and complicate it exceedingly. One is the altitude effect and the other is the tip-speed effect.

As far as the propeller is concerned, increasing the engine rated altitude is equivalent, due to the reduction in air density, to a further increase in the power rating. For example, the size of a propeller required to absorb a given amount of power at 20,000 ft. is the same as that required to absorb approximately twice the same power at the same peripheral tip speed at sea level.

The effect of tip speed as the altitude is increased is such as to require an increase in the propeller diameter for a constant equivalent power in order that tip-speed losses may be avoided. The reason for this condition is that the critical tip speed from the propeller-efficiency standpoint is, roughly, the speed of sound. Since the speed of sound is a function of the temperature, decreasing with it, the critical tip speed occurs at a lower peripheral speed at altitude than it does at sea level. In order to keep it within reasonable limits as the altitude is increased, a lower gear ratio with a larger diameter is required.

Of the two methods of increasing propeller area, the first, that of increasing the diameter for a given number of blades, is preferable up to the point where the weights of the different propeller types are no longer comparable because, as the number of blades is increased, the basic efficiency of the propeller falls off. For propellers for transport airplanes which probably can provide satisfactorily for the larger diameters, this method can be used. However, for the single-engine type of airplane, because of the diameter limitations, it probably will be necessary to provide the increase in blade area by an increase in the number of blades, and to take the penalty of reduced efficiency.

Both of these methods of increasing total blade area mean, of course, increased propeller weights. Fig. 2 shows the weight trends as the diameter is increased for three different propeller types. The weights of three-bladed propellers up to a diameter of 14 ft. were based on present actual ones. The extrapolated weights were assumed to vary as the cube of the diameter. The weights of the four-bladed propellers were determined by adding a third to the three-blade weights, and those for the six-bladed propellers were taken as double those of the three-bladed propellers.

To indicate the weight variation with power requirements, a study was made determining diameters for each of the three propeller types for a typical application. Gear ratios and diameters were considered optional, and were selected to result in a constant tip speed for all cases. Airplanes having

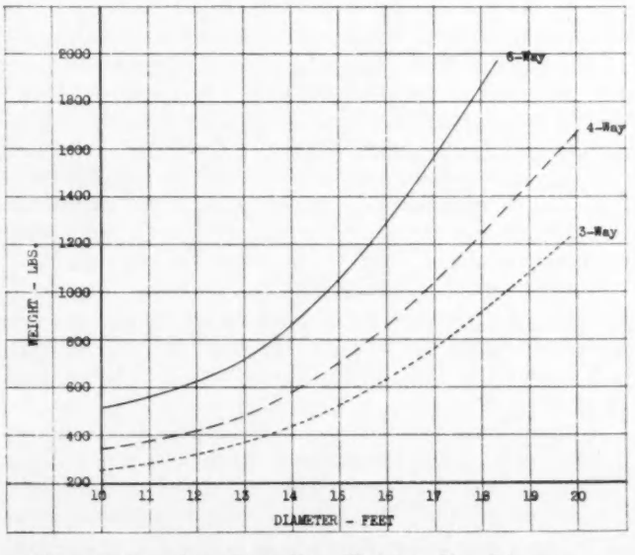


Fig. 2 – Complete Propeller Weights Vs. Diameters – Three-Blade, Four-Blade, and Six-Blade Propellers

maximum speeds of 300 m.p.h. at various equivalent sea-level powers (horsepower times density ratio) were assumed. Fig. 3 shows the diameters obtained.

Combining the data of Figs. 2 and 3 gives the variation of the weights against the horsepower. This variation is shown in Fig. 4.

It is apparent from this figure that, for increased powers at altitude, the weights for at least the propellers having the lesser number of blades become excessive.

The trend of actual weights, however, gives a distorted picture of the situation. To obtain a logical view of the trends, the weights should be considered from the standpoint of pounds per horsepower, as is done in the case of engines. Care must be taken, however, to employ the correct basis in the determination of specific weights. These weights should be evaluated on the basis of the equivalent sea-level horsepower and not on engine rating, as the equivalent sea-level horsepower represents the maximum power which a given propeller can absorb efficiently. If engine ratings are employed, an incorrect impression will be obtained. Fig. 5 shows the trend obtained when this method of specific weights

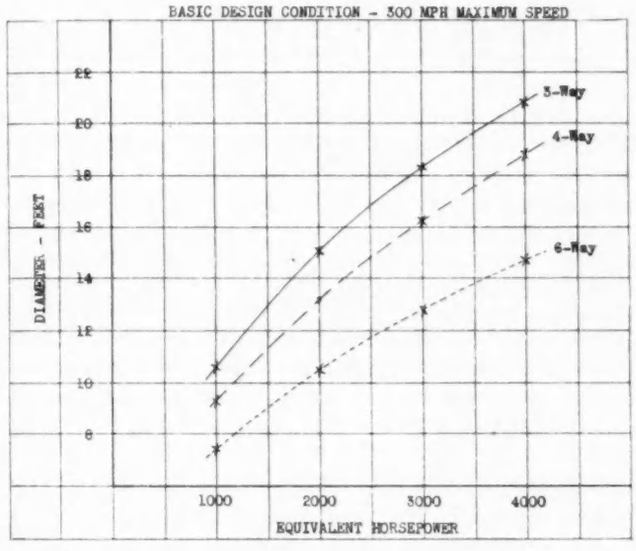


Fig. 3 – Diameters Required for Different Types of Propellers at Increasing Powers

expressed in pounds per rated brake horsepower is used, the ratings being selected arbitrarily in combination with altitude such that they would result in equivalent sea-level powers of 1000, 2000, 3000, and 4000 b.hp.; whereas Fig. 6 shows the correct trend obtained with specific weights expressed in pounds per equivalent brake horsepower. It is seen that, whereas actual weights increase rapidly, specific weights show only a small trend.

It is obvious from this study, however, that every effort should be made to reduce propeller weights, although they are not getting out of proportion as power requirements are increased.

In considering the possibilities of reducing weights it is, of course, necessary to keep in mind the safety factors involved. The present propeller designs have a specific weight of about 0.35 lb. per hp., and experience has indicated that they have a safe life of about 4000 to 5000 hr. If higher specific weights are employed, other things being equal, a longer life can be expected. For example, one installation currently operating has a specific weight of 0.45 lb. per hp. and is approaching 7000 hr.

In the problem of weight reduction, although the specific weight of 0.35 lb. per hp. has been found entirely adequate

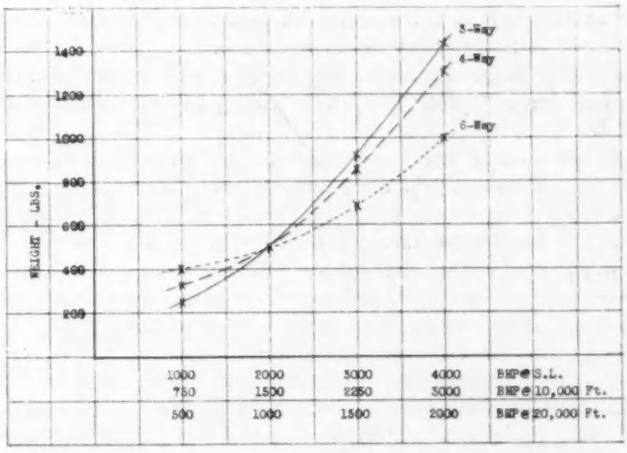


Fig. 4 – Propeller Weight Vs. Power Absorbed

for present designs, it should be remembered that, along with this problem, the question of maximum stress must be considered. As the diameter is increased, the propeller blade weight is, of course, necessarily increased. This increase means larger blade-shank sizes in order to maintain the same stresses. This increase in shank size is, in turn, reflected in increased hub sizes which, of course, mean further weight increases. This trend results in increasing specific weights, for the same degree of safety, as the diameter is increased.

In investigating ways of reducing propeller weights, attention should be given first to possible methods of isolating the propeller from severe vibration since this vibration induces high fatigue stresses. Such stresses are undesirable not only because fatigue strengths are considerably lower than strengths under steady load, but also because fatigue stresses are imposed on top of the steady stresses. If it were not necessary to consider severe vibration loads in propeller design, it would be possible to reduce specific weights by 1/4 to 1/3.

Insulation of the propeller from such loads probably could be accomplished through the use of an independent mounting of the propeller and a flexible drive combined with the proper damping. This method of obtaining a weight reduction is, of course, rather complex and requires radical changes in the conventional arrangement of propeller drive. The present thought is that the most practical method of reducing weights is through the use of lighter materials or different types of construction for the blades.

The possibilities of employing lighter materials, such as magnesium, depend on the ability of the metallurgists to produce a uniform product having acceptable properties. Considerable progress has been made in recent years so that development work can be undertaken with a degree of confidence in the basic materials not heretofore felt. The employment of hollow-steel construction for blades also is to be considered, although this type of construction complicates the situation considerably from the vibration standpoint due to the fact that, with the thin walls required, isolated portions of the blade may be subject to independent resonant frequencies. The use of plastics also should be given attention since recent developments indicate that satisfactory properties may be available in the near future. These methods possibly will result in reductions in specific weight on the order of 15 or 20 per cent.

It is believed that these steps in propeller design work will permit the satisfactory solution of the weight problem for increased powers at high altitude. It must be realized, however, that actual weights inevitably will increase somewhat as the powers are raised.

As previously mentioned, the basic performance of a pro-

peller falls off as the number of blades is increased. This effect is particularly severe for the take-off range of performance. For large airplanes, therefore, it will be desirable to employ large-diameter propellers having the least number of blades.

In the case of the small single-engine airplane with high power for substratosphere flight, another factor enters the problem. This is the effect of torque on airplane performance. It was found during a recent series of flight tests of a high-powered engine that torque effects, primarily at take-off, were quite pronounced, and caused definite swerving of the airplane. This torque was on the order of 5000 ft-lb. Although in this case the effects were probably particularly pronounced because the airplane was of rather early design and although they probably could be overcome in an airplane properly designed for the power, this experience indicates what may be expected when torques on the order of 11,000 to 13,000 ft-lb. are obtained. To overcome such a situation, the elimination of torque by the use of a counter-rotating tandem propeller has been considered. Such a propeller, of course, having multiple blades, could probably be used satisfactorily, since take-off with this type of airplane would not be critical.

From the performance standpoint, there is a question as to the effect of the blade interference with this type of propeller. However, recent test data indicate that, for at least flight operation, this effect is not too great. The design and weight problems involved with this type of propeller undoubtedly can be solved satisfactorily if further test work demonstrates its suitability from the performance standpoint.

Recent tests on a four-blade propeller of this type have indicated favorable efficiency values in comparison with the single rotating propellers. In order to evaluate the performance of this propeller a comparison, based on the data of the previously mentioned tests with three-blade and four-blade single rotating propellers, was made for a particular application.

The airplane and engine combination taken for this comparison was assumed to have the following characteristics:

Top speed at 20,000 ft. – 350 m.p.h.

Speed of best climb at sea level – 150 m.p.h.

Cruising condition – taken on propeller load curve at 60 per cent power.

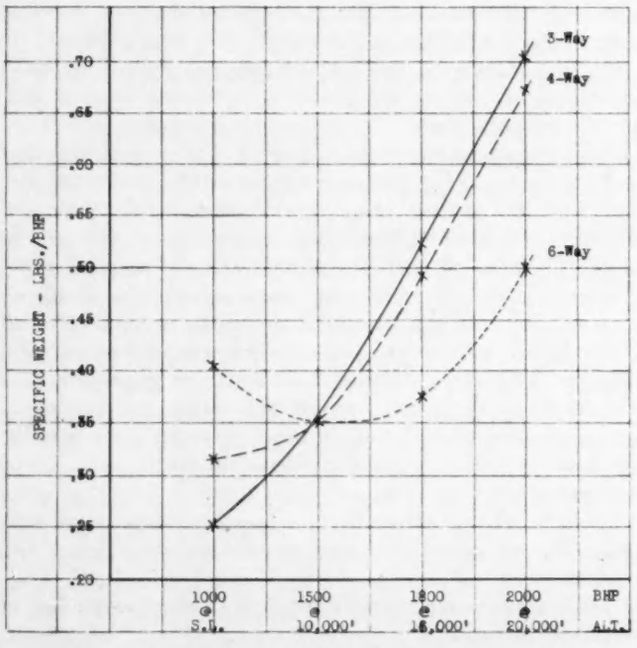


Fig. 5 – Propeller Specific Weight Vs. Engine Rating

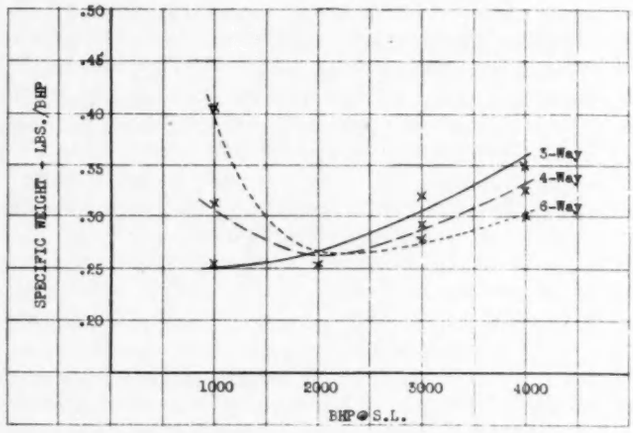


Fig. 6 – Propeller Specific Weight Vs. Power Absorbed

Engine altitude rating – 2000 hp. – 2800 r.p.m. – 20.000 ft.

Engine take-off rating – 2300 hp. – 3000 r.p.m. – sea level.

Propeller reduction-gear ratio – optional.

The propeller diameter and gear ratio, in each case, were selected to give maximum efficiency at top speed with an effective tip speed of 1000 ft. per sec. In all cases, blade designs were considered to be of conventional shape and geometrically similar.

The first comparison was made on the basis of no torque limitation. Table 1 gives the figures for this case. The diameters obtained are probably satisfactory for the four-bladed and tandem propellers but too large for the three-bladed propeller. The performance favors the tandem propeller in both top speed and cruising, with a slight loss being obtained for climb. Considering a torque limit of 5000 ft-lb., the three-bladed and four-bladed propellers are unsuitable. Although the torque for the tandem propeller is indicated in the table as zero, this is not exactly the case. With a constant difference between blade angles of the forward and rear components, the torque is zero for only one condition of power and rotational speed, but the variation is so comparatively slight that the torque can be considered negligible for all conditions.

Table 1

Propeller	3-Blade	4-Blade	Tandem
Diameter, ft.	15.8	12.6	11.45
Reduction-gear ratio	0.339	0.422	0.464
Effective tip speed at maximum airplane speed, ft. per sec.	1000	1000	1000
Propeller efficiency at maximum speed	0.789	0.785	0.810
Propeller efficiency at cruising	0.789	0.785	0.810
Propeller efficiency at sea-level climb	0.749	0.725	0.710
Torque at take-off, ft-lb.	11900	9570	0.000

If a comparison is made on the basis of a 5000 ft-lb. torque limit for take-off, the gear ratio will be fixed by this condition. Table 2 indicates the performances for this case. The performance penalty due to excessive tip speeds obtained through limiting the torque eliminates the single rotating propellers from consideration, although the diameters are satisfactory from the airplane design standpoint.

Table 2

Propeller	3-Blade	4-Blade	Tandem
Diameter, ft.	9.5	8.7	11.45
Reduction-gear ratio	0.807	0.807	0.464
Effective tip speed at maximum airplane speed, ft. per sec.	1340	1250	1000
Propeller efficiency at maximum speed	0.622	0.667	0.810
Propeller efficiency at cruising	0.766	0.770	0.810
Propeller efficiency at sea-level climb	0.568	0.619	0.710
Torque at take-off, ft-lb.	5000	5000	0.000

The effect of taking a compromise on the three-blade and four-blade performance in an effort to reduce the undesirable

(*Continued on page* 300)

Propeller Factors Tending to Limit Aircraft-Engine Powers

By G. T. Lampton
Lycoming Division, Aviation Mfg. Corp.

THERE is a strong trend toward increasing airplane gross weights inasmuch as larger sizes simultaneously result in structural economy and aerodynamic improvement. The concurrent demand for larger powerplants may be limited by the propeller since, at constant tip speed and velocity, its weight is proportional to the 3/2 power of the horsepower.

By assuming a specific engine-weight curve for increasing powers and adding the propeller weight, specific-weight curves for the powerplant group are obtained. These curves show minima in the vicinity of 2500 and 6000 hp. for 250 and 450 m.p.h. design speeds, respectively, at sea level. Multiblade propellers and very high design speeds tend to reduce the weight penalty, so there is no ultimate limit to engine powers.

ENGINE powers, gross weights and airplane performances are now increasing at such a rapid rate that it has become essential to stop and examine these trends for practical limits. Airplane, engine, and propeller engineers are equally involved with commercial personnel whose duty it is to examine designs, tools, and equipment for obsolescence. The rapid increase in engine ratings with improved fuels and generally higher critical altitudes has the propeller industry at a disadvantage. In addition to this condition, we are now up against large increases in displacement which require propellers in sizes well in advance of the art. If present trends persist, propellers previously considered in the Zeppelin category shortly will be standard equipment on pursuit ships.

The design pressure forcing increased powerplant sizes is the natural result of increased gross weights which, in turn, simultaneously improve operating costs per unit payload and the airplane design conditions. Boulton[1] shows the following advantages for increasing the gross weight of flying boats and the same reasoning is applicable to landplanes:

(1) The structural and powerplant weights definitely are a decreasing percentage of the gross.

(2) The span loading is increased, so L/D maximum occurs at a higher cruising speed.

(3) Scale effect reduces the parasite drag.

(4) The percentage of structural parasite drag is reduced with substantial improvement in L/D.

(5) The span load distribution is improved.

In any case we should expect weights to increase until the engines can be submerged in the wing.

While increasing the total power with gross weight, it is obviously desirable to keep the number of engine installations to the safe minimum for maintenance purposes, to improve the span load distribution, to reduce yawing moments in case of failure, and to realize the economies possible by reduction of engine controls, engine instruments, plumbing, cowling, and so on.

Aside from these arguments we might note that propellers occupy about half of the span of today's four-motored planes. If we increase the gross weight at constant wing loading and power loading, the span goes up only as the cube root of the weight. If engines of greater size are not available we would very soon run out of span on which to install them and be forced to tandem nacelles, which arrangements have many objectionable features. Important disadvantages will have to be adduced to arrest these strong trends toward increasing gross weights and powerplant sizes.

Of the propeller factors, let us consider first critical tip speed, since it will yield most readily to analysis. The latitude in this respect which propeller engineers first enjoyed when gearing was adopted generally has now been used up by increased ratings, critical altitudes, and overspeeding for take-off. From the vector diagram on Fig. 1 it is immediately apparent that the airplane velocity and an arbitrarily chosen tip speed determine the design pitch ratio V/ND. This limitation will be more severe for military craft which may have to operate at full throttle than for commercial long-range operations where cruising engine speeds may be reduced substantially. To avoid excessive diameters, weights,

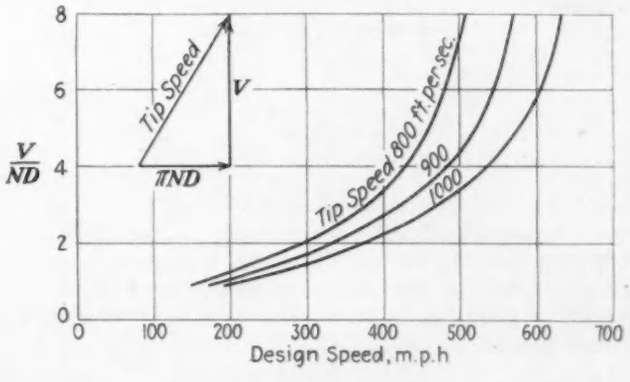


Fig. 1 – Tip Speeds

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 10, 1938.]

[1] See the *Journal of the Aeronautical Sciences*, Vol. 4, April, 1937, pp. 254-257: "Aerodynamic Factors in the Design of Long-Range Flying Boats," by B. C. Boulton.

and pitches, it will be necessary for engine manufacturers to have each model available with several different gear ratios to obtain high tip speeds at any design velocity. A tip speed of 900 ft. per sec. was selected for all the following calculations.

At present our transports and bombers are operating in the vicinity of 200-300 m.p.h., and the propeller problems are not too difficult. With increased powers and substratosphere operation, however, there will be a rapid advance on the 400 m.p.h. zone, which must be accompanied by increased pitches and, eventually, by a loss in efficiency.

Fig. 2 shows the envelope of efficiency curves as a function of V/ND and has the velocities corresponding to 900 ft. per sec. tip speed spotted on it. In previous years the design pitch ratios were usually one or less, and reduction gearing resulted in substantial improvement in propeller efficiencies. Modern conditions are right at the peak of the curve, so increased velocities will reduce the efficiency. Although the rate of the down slope is uncertain, this characteristic is exhibited by practically all model and full-scale propeller tests. We certainly cannot expect a propeller approximating the feathered condition to be very efficient.

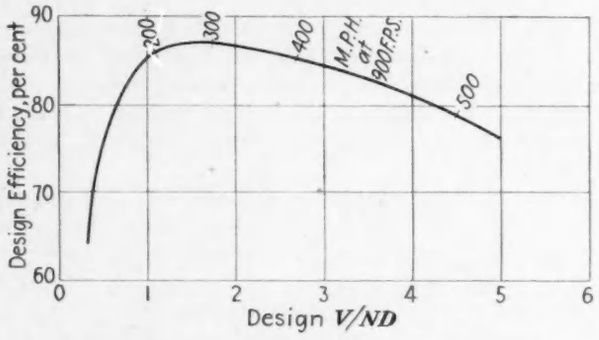


Fig. 2 – Efficiency Vs. V/ND at 900 Ft. per Sec.

Fig. 3 compares the required pitch distribution for a conventional blade with design V/ND of 1.0 and 4.5, which, at 900 ft. per sec. corresponds to 185 and 500 m.p.h., respectively. Note that the latter propeller has blade angles of 55 deg. at the tip and 80 deg. at ¼ radius. Naturally, such angles are so high as to have excessive torque components, and the Drzewiecki efficiency suffers as shown in Fig. 4. Weighting the Drzewiecki efficiencies with the air-load distribution, we arrive at an efficiency of approximately 79 per cent at 500 m.p.h. (Fig. 2), or 8 per cent below optimum. This result is independent of power and the Froude factor because it is predicated on constant tip speed. At any velocity, the product nD is constant, V/ND is fixed and, for given solidity, the power absorbed is proportional to the diameter squared.

$$\text{Hp.} \propto n^3D^5$$
$$nD = C$$
$$n^3 = \frac{C}{D^3}$$
$$\text{Hp.} \propto D^2$$

Since the disc area varies likewise, the slipstream condition and momentum efficiency are constant. It should be noted, however, that the air loading of the high-pitch propeller is much greater over the inboard portion of the blade and will require especially clean nacelle treatment to avoid substantial interference.

[2] See the *Journal of the Aeronautical Sciences*, Vol. 5, December, 1937, pp. 37-52: "Propellers for Aircraft Engines of High Power Output," by Frank W. Caldwell.

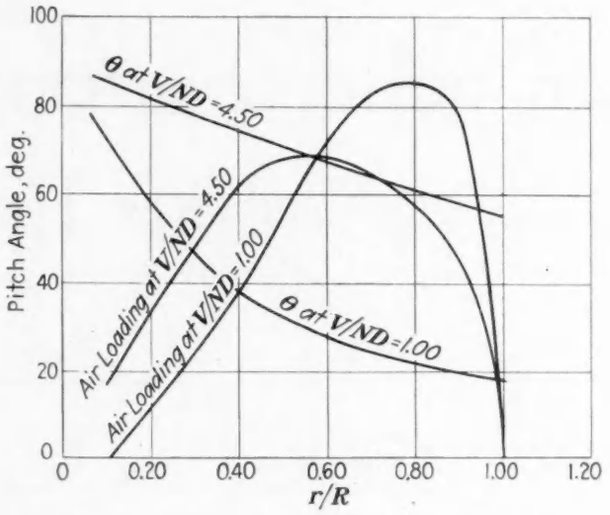


Fig. 3 – Comparison $V/ND = 1.0$ and 4.5 – Tip Speed, 900 Ft. per Sec.

The foregoing phenomena really involve only the geometry of the propeller, primarily controlled by true air speed, and are not affected by engine factors except as they raise velocities. Since 500 m.p.h. is just around the corner for the high-speed categories at altitude, the behavior of this class of propeller in slow-speed regimes would bear considerable investigation. At 600 m.p.h. (900 ft. per sec.) local velocities around the structure will equal that of sound and determine our ultimate speed. Note from Figs. 1 and 2 that the pitch ratios required will be about 6, and our approach on this boundary will be handicapped by poor propeller efficiency. Another minor observation from Fig. 2 is that high b.m.e.p. cruising at slow propeller speeds may result in a loss of efficiency offsetting the improvement in specific fuel consumption in cases where the design pitches are high.

The real propeller limitation on engine sizes, if any, resides in the increasing weight of the propeller and, therefore, tip speeds will be maintained high and constant with increasing power. Since blade stresses are proportional to the square of the tip speed, we must abandon the previous practice of thinning out geared blades and maintain geometrical similarity. Thus, the propeller weight will be proportional to the cube of the diameter with sufficient precision for our purposes. Fig. 5 shows the weight of multibladed propellers on this assumption, and is in close agreement with Caldwell[2].

Fig. 6 shows the diameters required for three-, four-, and six-bladed propellers for 250 and 450 m.p.h. airplanes against the size of the unit powerplant, constructed partly from full-

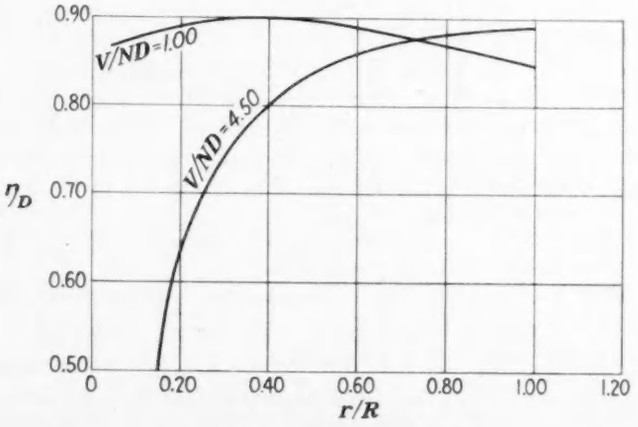


Fig. 4 – Drzewiecki Efficiencies

scale characteristics. Since all propellers will operate at the same tip speed, air speed, altitude and power, the diameters are inversely proportional to the square root of the number of blades or the solidity. In controllable propellers it seems preferable to increase the number of blades rather than the blade width because the centrifugal pitch reducing moment, and therefore either control forces or weight of counterbalances, are proportional roughly to the cube of the width.

With given design conditions the diameter required is proportional to the square root of the horsepower, and the propeller weight is therefore proportional to $(hp.)^{3/2}$. Engines now being developed will require diameters of approximately 20 ft. with weights of about 1000 lb., and the foregoing exponent indicates that these weights will increase very rapidly. To investigate the basic importance of propeller weight it is necessary to study its effect on the airplane. A convenient method of attack is to assume an air speed and tip speed, estimate the complete powerplant weight plus the propeller, and convert it to specific weight in lb. per hp. We can then examine the curve of specific weight vs. hp. for the optimum size.

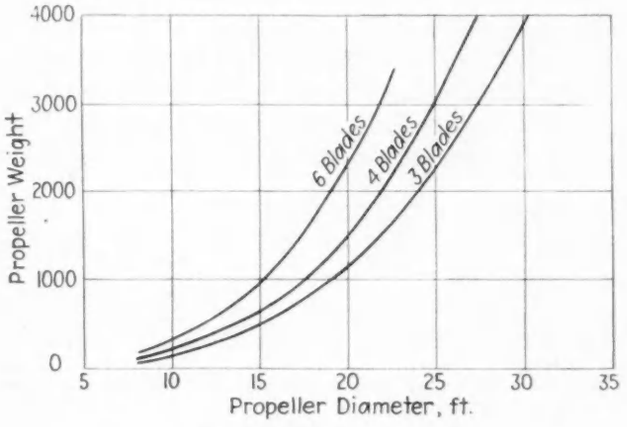


Fig. 5 – Propeller Weight

The basic assumption will be that of the specific weight of the engine itself as shown on the lower curve of Fig. 7. Since we are dealing only with the merit of increasing engine sizes, it is necessary to disregard any reduction in specific weight due to improved fuels, materials, design advances, and so on, and attempt to evaluate only the economies which can be achieved by increasing power. With increasing size a relatively more compact engine should be obtainable, and design economy in accessory drives, and so on, can be achieved. A reduction in the basic weight of the engine of 0.05 lb. per hp. does not seem unreasonable when going from 1000 to 2000 hp. In the larger sizes an increment of 1000 hp., of course, will result in smaller savings. As a credit for reducing the number of nacelles by increasing engine sizes I have also assumed a fixed weight of 115 lb. for engine instruments, engine controls, and so on, which is naturally to the advantage of the larger engines. This weight being only a small percentage of the total powerplant, this assumption is relatively unimportant.

At constant conditions and tip speed, letting B, D, and W be the number of blades, diameter, and propeller weight, respectively, we have

$$\text{Hp.} \propto BD^2$$

$$D \propto \frac{1}{B^{1/2}}$$

$$W \propto BD^3 \propto \frac{1}{B^{1/2}}$$

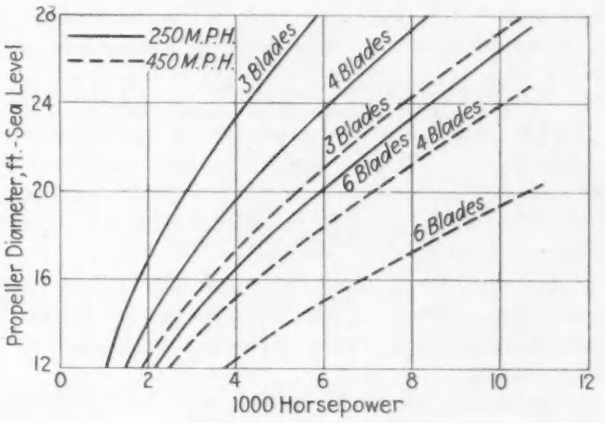


Fig. 6 – Multibladed Propeller Diameters

Thus, increasing the number of blades reduces the propeller weight.

Fig. 7 shows that the reduction in engine weight largely offsets the increased propeller weight. In the case of comparatively slow airplanes the propellers are larger, and the optimum powerplant rating at sea level is only slightly above 2000 hp. The peak is not sharp, however, so substantially larger powers can be used in combination with a plurality of blades. This arrangement has its disadvantages in that the slower planes commonly will have critical take-off and initial climb criteria.

The faster planes with smaller propellers have an optimum rating of 6000 hp. at sea level, but once more the curves are so flat that design convenience will probably govern the size.

The assumed specific engine weight curve corresponds roughly to short time ratings, so Fig. 7 is applicable to military type of craft which require full-throttle operation, rather than those classes of ships for which the propeller would be selected strictly for cruising at small fractions of power. In this case the specific engine weight at cruising would be

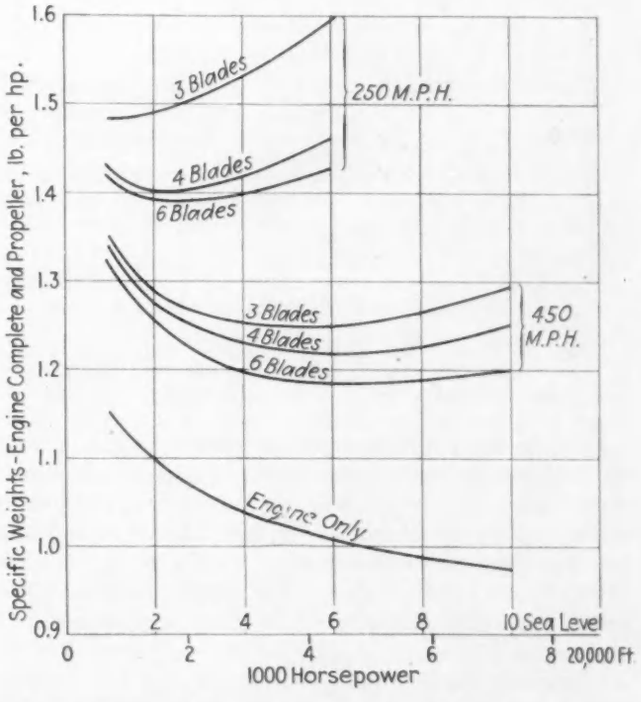


Fig. 7 – Powerplant Group – Specific Weight Vs. Power – Tip Speed, 900 Ft. per Sec.

much greater, the propeller weight of less importance, and the minimum specifics would occur at substantially higher powers.

The abscissa scale for 20,000 ft. critical altitude was obtained by applying the square root of the relative density to the horsepower. Thus we have the original propeller diameter and weight, bearing in mind the 900 ft. per sec. tip speed. The heavier altitude propeller, of course, causes the optimum size to occur at lower power.

If we added in a structural weight about equal to the power group to obtain a "specific weight empty," the minima would be sharper, but would involve the arrangement of the entire airplane.

By multiplying the specific weight of the powerplant group by the propeller efficiency we can obtain "specific thrust horsepowers," as shown in Figs. 8, 9, and 10 for the design speed

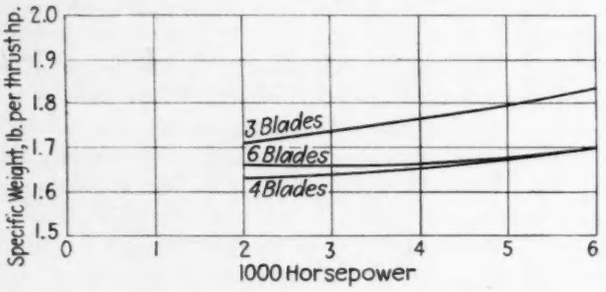


Fig. 8 – Thrust Horsepower – Specific Weight at Design V/ND – 250 M.P.H., 900 Ft. per Sec. Tip Speed

of 250 m.p.h., best climb at 65 per cent V, and initial climb at 30 per cent V, respectively. For commercial operations where the payload may be governed by safety in the initial climb, an excessive number of blades is obviously undesirable.

In general, the application of efficiency reduces the slope of the specific-weight curves, being independent of horsepower and, thereby, allows more latitude in the selection of engine size.

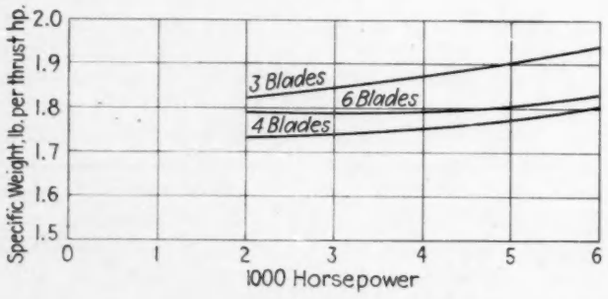


Fig. 9 – Thrust Horsepower – Specific Weight at Best Climb – 250 M.P.H., 900 Ft. per Sec. Tip Speed

It is obviously impossible to determine a sharp boundary for engine sizes from generalities and assumptions such as those advanced here. In the last analysis the airplane designer will have to select powerplants of a size which will give him the number of independent units dictated by safety considerations, and weigh possible weight penalties against aerodynamic improvement, maintenance requirements, and so on.

Increasing velocities will permit larger engines and, therefore, larger gross weights, which establishes a cycle with further increase of speed. The optima shown on Fig. 7 are

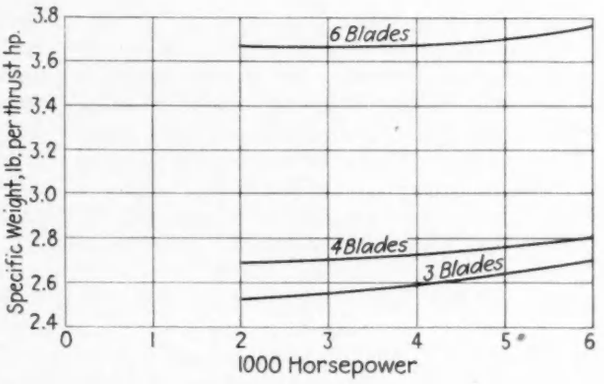


Fig. 10 – Thrust Horsepower – Specific Weight at Take-off and Initial Climb – 250 M.P.H., 900 Ft. per Sec. Tip Speed

artificial because they were derived for definite velocities. Actually, there appears to be no limit to size other than the courage of the industry.

Discussion

Necessity for Propeller-Weight Reduction Stressed

– T. P. Wright

Curtiss-Wright Corp.

DURING the past two years several papers have appeared expressing considerable concern over the prospect of reaching a definite limit in aircraft-engine size, brought about by increasing propeller weight. Commentators have feared that propeller weights in proportion to gross weights (and with accompanying decrease in payload) would go up so rapidly that the power of the individual power unit could not be increased beyond horsepowers currently being rapidly realized.

It is therefore interesting and important to hear from the propeller engineers themselves concerning this important point and it is indeed encouraging to find them with an optimistic attitude on the situation. It is important that, with increasing size of aircraft, the number of power units be kept to a minimum, as brought out by Mr. Lampton. The operating cost of an airplane will, in general, increase per unit of payload as the number of power units increase.

I feel that there is some question, however, concerning one fundamental assumption which Mr. Lampton makes, that is, that there will be a reduction in the unit weight of the engine itself as powers increase by virtue of such larger size and which decrease will serve as a balance for the increase in propeller weight accompanying it. It does appear likely that the powerplant unit installation weight, aside from the engine itself, will decrease with increasing power, but hardly sufficiently to offset the increase in propeller weight entirely unless something is done in the propeller itself to alleviate the operation of the weight increase indicated by application of the formula

$$\text{Weight} = (\text{Hp.})^{3/2}$$

It is therefore believed that propeller engineers should continue to apply their best efforts to the problem of reducing specific propeller weight by actual designs considerations. More discussion of this point is believed to be in order and it is hoped that future meetings will bring out ways and means which will tend toward propeller-weight decrease, perhaps including analyses showing the change in the exponent in the preceding formula when using hollow rather than solid blades.

Mr. Lampton also mentions the need for engine manufacturers to have each engine model available with several different gear ratios in order to obtain high tip speeds at any design velocity, with 900 ft. per sec. as a limit. This statement would automatically seem to point toward the need soon of providing the engine itself with a clutch so that two gear ratios may be selected to cover flight regimes corresponding to widely differing V/ND.

The point that Mr. Lampton raises concerning decrease in propeller efficiency as airplane speeds exceed 300 m.p.h. is interesting as representing a limitation in the magnitude of the ultimate speeds that aircraft may hope to attain, additional to compressibility effects, emphasized recently by experiments conducted by the N.A.C.A.

Aerodynamic Considerations Affecting Propellers for Large Engines

By George W. Brady

Curtiss Propeller Division, Curtiss-Wright Corp.

FROM the aerodynamic standpoint, propellers for engines of several times present powers will operate just as efficiently as those for smaller powers. For long-range aircraft operating at altitudes of 20,000 to 30,000 ft. three-blade propellers increased somewhat in diameter over what is current practice are definitely indicated although, for medium-range types, the diameters of present practice scaled up by the square root of the take-off power ratio appear satisfactory.

As alternatives to the increased diameters for the long-range high-altitude aircraft, the three-blade propeller of about 15 per cent smaller diameter with a two-speed reduction gear will give equal range but a poorer take-off, or the four-blade propeller of the same diameter as the smaller three-blade design will give very close to equal range without a two-speed reduction gear and slightly poorer take-off.

The importance of pitch distribution designed for operating speeds is indicated, as is the necessity for keeping blade-shank sections faired when exposed to the airstream particularly in liquid-cooled installations. Propellers which can be feathered in case of powerplant failures are considered essential to long-range aircraft from the standpoint of safety and increased performance on the remaining engines.

THERE has been a considerable amount of discussion in recent months concerning the possibility that the maximum size of a single aircraft powerplant may be limited by the means used to transform its power into forward thrust. The object of this study is to determine if such a limit is imposed by the conventional means used to obtain this thrust – the aircraft propeller – from the aerodynamic standpoint.

Various considerations other than aerodynamic enter into the whole picture, of course, of which the weight of propellers for the higher-powered units is probably of greatest concern. It is not intended to minimize the importance of this consideration or the difficulty of keeping propeller weight down, but it is believed that, if weight is the only factor that stands in the way of the higher-powered units and if it is established that they are suitable from every other standpoint, methods will be found to overcome this difficulty.

In this connection it may be pointed out that the autogyro rotor is very similar to the aircraft propeller in general form and that successful rotors have been built having three and four blades and diameters of 30 to 40 ft. or more. Weight of these rotors is understood to be substantially less than 500 lb. As will be shown later in the paper, a 30-ft. diameter three-blade propeller is suitable for about 5000 hp. It is true that the gyro-rotor construction is not suitable directly for propellers since rotors are not subjected to the same magnitude of torques and thrusts as are propellers of equivalent diameter but, even assuming a two- or three-fold increase in weight for this purpose, the weight per horsepower for this general type of construction would not be out of line with present propeller weights of the order of 0.35 lb. per hp. Hence, even though conventional construction may run to too great unit weights for economical operation, some different type will be developed such as the use of hollow or fabricated blades or perhaps along the lines of insulation of the propeller from engine vibration as mentioned by Caldwell[1]. This latter method would appear to be a joint engine-aircraft-propeller manufacturers' problem.

It also is believed that the weight problem, even with conventional construction using solid aluminum-alloy or hollow-steel blades, is not so serious but that substantial increases over present sizes may be made without running into excessive weight. The prediction of weight of large propellers is usually based on a variation with diameter cubed starting with the weight of propellers of about 12-ft. diameter as a base point. Inasmuch as, with this size of propeller, the blade weight is about half of the total propeller weight and it is only the blade weight that varies as the diameter cubed, hub weight increasing more nearly in direct proportion to diameter, the increase in weight for a 30 per cent increase in diameter over the basic 12-ft. size is only 76 per cent or as the diameter ratio to the 2.15 power instead of the cube. In the case of the propeller having two-thirds the weight in the blades, the figures for 30 per cent in diameter become 90 per cent and the 2.45 power respectively.

Analysis by this method indicates that, for an increase in power of 100 per cent which requires an increase in diameter of 40 per cent over that for which the basic 12-ft. propeller is suitable, the weight per horsepower for three-blade propellers of conventional construction will increase less than 10 per cent

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., Mar. 10, 1938.]

[1] See the *Journal of the Aeronautical Sciences*, Vol. 5, December, 1937, pp. 37-52: "Propellers for Aircraft Engines of High Output," by F. W. Caldwell.

or from about 0.35 lb. per hp. for 1000-hp. units to 0.38 lb. per hp. for 2000-hp. units.

Another factor of special importance from the airplane-designer's viewpoint and which sometimes limits propeller diameters is that of clearance of the propeller tips either with the ground, the fuselage, or adjoining propellers. This factor dictates the use of the smallest diameter propeller which is aerodynamically suitable. Later in the paper will be shown probable diameters for certain powers and a comparison will be made with the way in which physical dimensions of aircraft to use these powers may be expected to increase, from which data it is indicated that, as powers go up, clearances will not be a limiting factor.

Various other considerations of a mechanical nature, such as vibration characteristics, power required to change pitch, and so on, affect the propellers of the larger diameters, but they are more in the nature of detail design problems and hence are not so fundamental as propeller aerodynamics. The problem of making full use of engine power under all of the many different flight conditions is a very difficult one and one which cannot be solved without a certain amount of compromising between various conditions unless propellers of variable diameters and with variable pitch distributions driven by engines with variable gears are considered. This study is limited, however, to propellers having blades of fixed diameter with fixed pitch distribution rotatable about their axes in a controllable hub and, in general, to a single ratio of engine-crankshaft-to-propeller speed, although variation of propeller r.p.m. by a two-speed reduction gear is alternately considered. Some compromise in propeller selection, therefore, will be required.

Propeller Selection

In selecting a propeller for a given airplane-engine combination, it will be necessary to determine what factors of its performance require most emphasis. Except for types which operate over a limited power range, it is not possible to select one propeller which will give the very best performance for all flight conditions. However, by choosing the best propeller for one or two conditions, it usually is possible for this propeller to give reasonably good performance for a majority of the other conditions.

The major items to be determined by aerodynamic analysis are: 1. diameter, 2. gear ratio, 3. number of blades, 4. blade width, 5. blade plan form, 6. blade pitch distribution, and 7. blade shank fairing.

Of these items the first four are determined on the basis of the best compromise between the various operating conditions, the fifth and sixth generally are selected for the operating conditions on which most emphasis is laid, and the seventh is usually taken as it comes although, as will be shown later, it is extremely important for certain types of installations.

Flight conditions on which propeller selection will be based depend upon the type of aircraft. For the principal types the following flight conditions are listed in order of importance:

Long-Range Types
1. Take-off.
2. Cruising at low power for maximum range.

Medium-Range Types
1. Cruising at medium power.
2. Take-off.

Short-Range High-Performance Types
1. High speed.
2. Climb.
3. Take-off.

For the purpose of this paper, a comprehensive study was

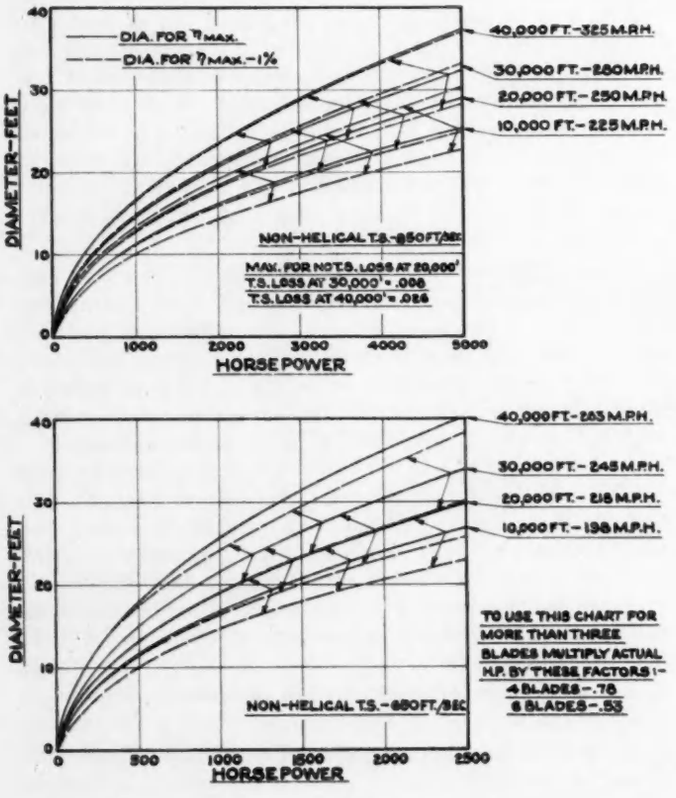


Fig. 1

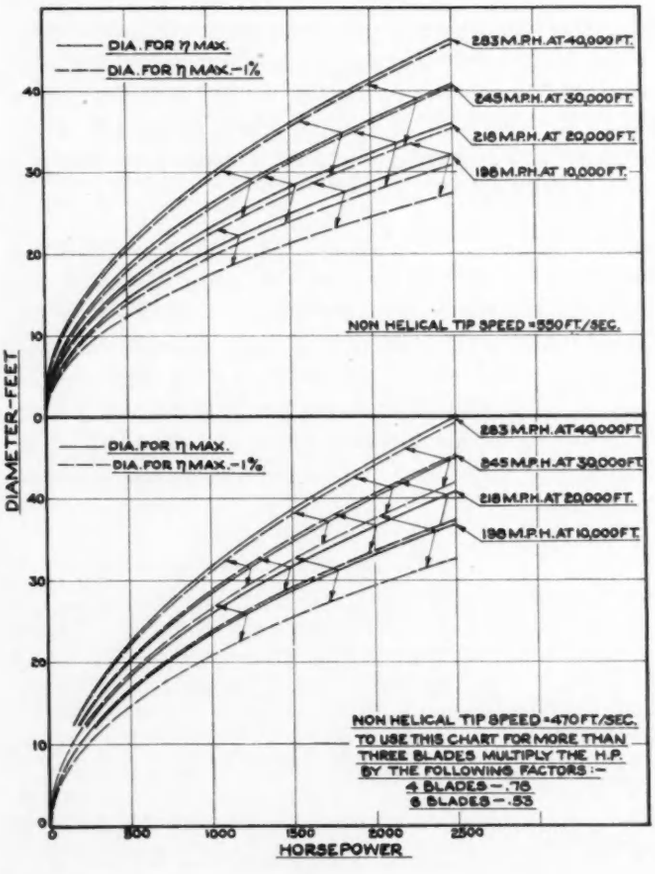


Fig. 2

Figs. 1 and 2 – Charts Showing the Range of Diameters Which Will Give, for Any Power, an Efficiency within 1 Per Cent of Maximum Efficiency

made of the effect of variations in propeller diameter, number of blades, and engine reduction gear ratio. Blade-width change gives results similar to change in the number of blades, and its effect may be approximated from the data for number of blades. Proper plan form is important, but it is assumed that a satisfactory shape is used in the propellers considered. Pitch distribution and effect of shank fairing are treated.

Diameter, Number of Blades, Gear Ratio

To cover all possible ranges of aircraft power, speeds, operating altitudes, and other variables in a single series of charts is impossible. It seems advisable to fix one variable at least and to determine the best compromise propeller with that variable fixed and for any given set of other conditions. Of the variables which should be considered constant for purposes of study, horsepower appears to be the most logical since, once the analysis is completed for one power, it may be made applicable to any other power by the simple process of multiplying the diameter found for the chosen power by the square root of the ratio of the new to the old power. The new rotational speeds may be obtained by multiplying by the reciprocal of the diameter ratio, thus holding tip speed constant. This relation may be seen readily by consideration of the formula for power coefficient:

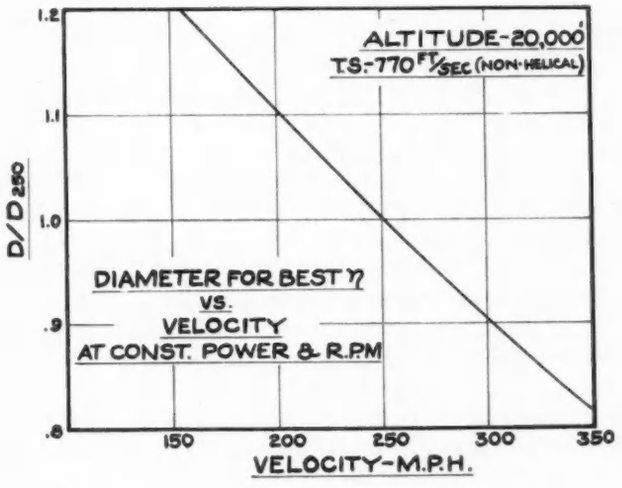


Fig. 3 – Corrective Factor for Diameter with Speed

$$C_p = \frac{P}{\rho n^3 D^5} = \frac{P}{\rho (nD)^3 D^2}$$

from which it is evident that if nD or tip speed is held constant:

$$D \propto \sqrt{P}$$

and since $$n \propto \frac{1}{D}$$

we may write $$n \propto \sqrt{\frac{1}{P}}$$

Data are presented in two groups, the first covering take-off and the second, for high speed and cruising flight. Take-off calculations are made on the basis of an arbitrarily selected horsepower of 2500, approximately double the powers in current use. Diameter for any other desired power may be obtained from the simple relationship between power and diameter keeping tip speed (πnD) constant. Engine or propeller speeds as such are not considered, but gear ratios giving certain tip speeds are assumed.

The cruising and high-speed analysis is made on a slightly different basis. Since, for maximum speed or range it is necessary that the propeller efficiency be at the maximum, a series of charts are presented, Figs. 1 and 2, showing the range of diameters which will give, for any power, an efficiency within 1 per cent of the maximum efficiency. One chart for each of several tip speeds is used and, on each chart, curves representing the range of diameters for even 10,000-ft. altitudes up to 40,000 ft. are drawn. The charts are calculated for an airplane having an assumed maximum speed of 250 m.p.h. at 20,000 ft. Speed at all other altitudes varies from the assumed speed inversely as the cube root of the density ratio. The chart for 850 ft. per sec. tip speed is based on maximum speed, and the three cruising charts are based on speeds obtained when cruising on 40 per cent power. In order to provide a quick method of approximating the change in diameter for other speeds, a corrective factor for diameter is plotted in Fig. 3.

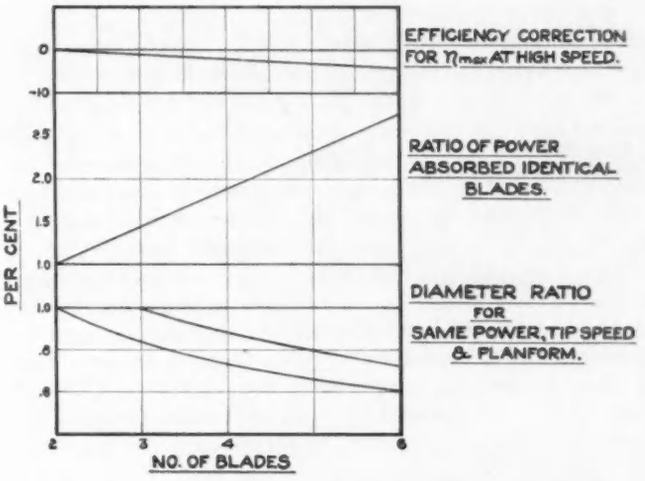


Fig. 4 – Factors for Applying Three-Blade-Propeller Cruising and High-Speed Charts to Four-Blade and Six-Blade Propellers

Propellers having three, four, and six blades are considered in the study although cruising and high-speed charts are plotted directly only for three-blade propellers. They may be applied to four and six blades by multiplying the power by the factor shown and considering that the efficiency is reduced below the three-blade efficiency by the factor shown in Fig. 4.

Two-blade propellers are not considered because their use in the larger sizes is not considered advisable due, to some extent, to the larger diameter required but principally because of the fact that, when yawed, two-blade propellers give rise to a periodic couple of considerable magnitude of frequency twice propeller speed, whereas propellers of three or more blades are steady at any moderate angle of yaw.

Take-Off Thrust

The take-off data, shown in Figs. 5, 6, and 7, are obtained from the C_T/C_p versus C'_p at static condition curves of Fig. 8 for three-, four-, and six-blade propellers. These fundamental data for the four- and six-blade propellers were obtained by modifying a working curve of a three-blade propeller developed from electric-whirl and other test data by analysis of the limited data of four- and six-blade models or full-scale propeller tests that are available. Absolute accuracy of the four- and six-blade curves is not claimed, but they check comparative results of available tests quite closely.

Static thrust is not, of course, a complete index to thrust at all stages of take-off, but it is quite representative so long as

comparison is being made between propellers of the same airfoil section, whether RAF-6, Clark Y, or some other type. In this range of ordinarily used diameters it is true that the greater the static thrust, the greater the thrust throughout the take-off run. The particular set of curves shown in Fig. 5 are for a 1000 ft. per sec. tip speed at take-off r.p.m. and are plotted as static thrust against horsepower for various diameters and number of blades for medium-width blades of a blade-width to diameter ratio at 0.75 tip radius of 0.06:1.

This chart indicates that there is a maximum static thrust per horsepower which can be obtained with a given number of blades and which is a constant value with horsepower. However, this maximum unit thrust requires diameters considerably larger than are practical from other standpoints so that the data are replotted on Fig. 6 as diameter versus horsepower for thrusts 90 per cent, 85 per cent, and 80 per cent of the envelope lines.

Both of the charts show the distinct superiority of the three-blade type over either the four- or six-blade propellers where maximum static thrust is required. The static thrust per horsepower at the diameters giving 90 per cent of the maximum gives a static thrust of 4.5 lb. per hp. for the three-blade, 4.05 lb. per hp. for the four-blade and 3.4 lb. per hp. for the six-blade propeller, a loss over the three-blade propeller of 10 per cent for the four-blade and 24 per cent for the six-blade propeller. Diameters are seen to be 87 per cent and 71 per cent respectively of those for the three-blade propeller.

It is also shown clearly in Fig. 5 that, for a given diameter, there is a certain power beyond which very little gain in static thrust is derived no matter how much additional power is put into the propeller. For the 10-ft. three-bladed

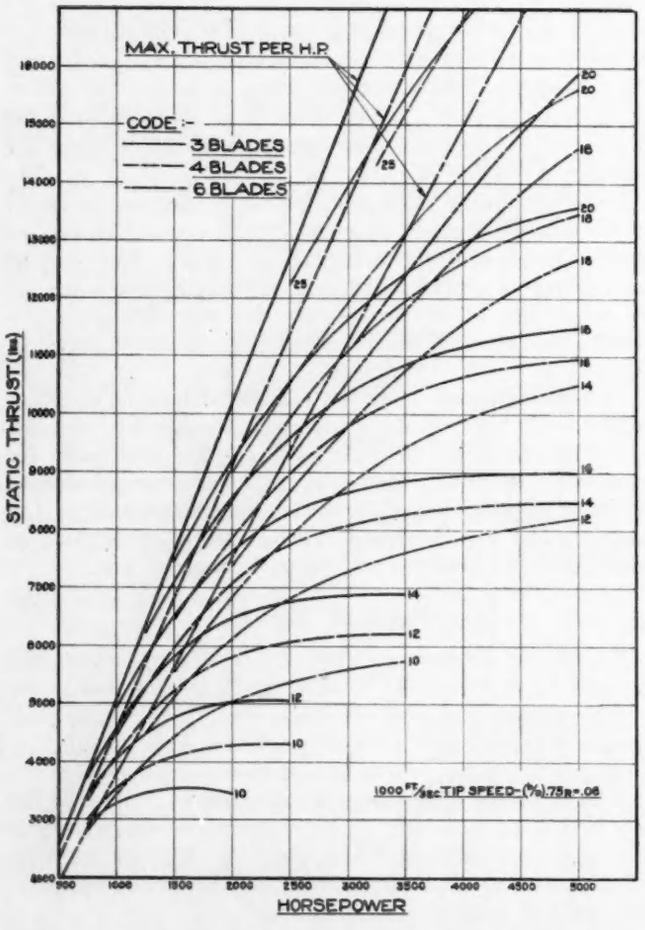


Fig. 5 – Static Thrust for Three-, Four-, and Six-Bladed Propellers of Various Diameters

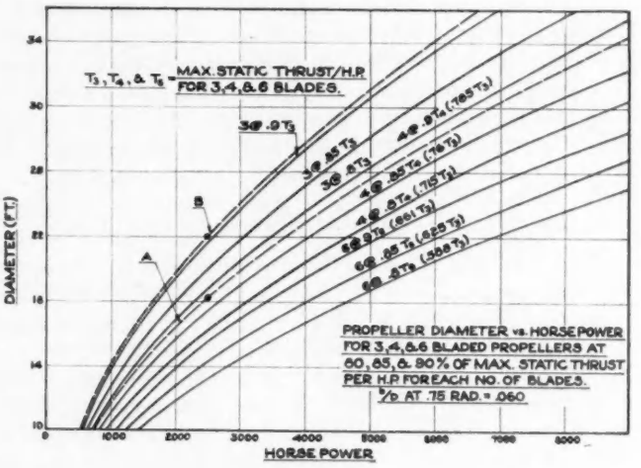


Fig. 6 – Propeller Diameter Vs. Horsepower for Three-, Four-, and Six-Bladed Propellers at Various Percentages of Maximum Static Thrust per Hp.

propeller, an increase in power of 20 per cent beyond 1000 hp., results in an increase in static thrust of only 4.5 per cent. For this case a propeller of more than three blades is indicated definitely, static thrust for the same diameters and power for a four-blade propeller being 14 per cent greater. No advantage at this point is seen in going to six blades as the static thrust is very nearly the same as for the four-blade propeller but, if this power is again increased by 20 per cent or more, the six-blade propeller will give superior static thrust.

Although the definitely lower maximum static thrust of the six-blade propeller rules it out of consideration for aircraft with high power loadings, the relative advantages of a four-blade and three-blade propeller should be considered further. Fig. 7 shows thrust versus diameter at several different tip speeds for a take-off rating of 2500 hp. The upper part of the figure is for static conditions, whereas the lower is for a V/ND corresponding to 80 m.p.h. Actual flight test data on thrust at V/ND's toward the end of the take-off run are meager so that the lower curves are based on wind-tunnel measurements and the absolute values are not vouched for. However, it is believed that they are at least relatively accurate. The first thing noted is the comparatively large diameter required to obtain maximum static thrust. The point A is placed on the chart to show the diameter which would result if the 2500-hp. propeller is chosen on the basis of propellers representative of current practice. The diameter of 18.2 ft. is scaled up from an 11.5-ft. diameter propeller for a take-off power of 1000 hp. operating at 920 ft. per sec. tip speed.

The location of point A indicates how far below the peak it is sometimes necessary to operate. This compromise propeller gives a static thrust of 8400 lb. as against a possible figure of better than 11,000 lb., or an improvement of 31 per cent. Diameter for this maximum is at least 22 ft. in three blades.

In order to determine how much of this 31 per cent improvement in static thrust will carry through to the end of the take-off run, it may be noted in the lower part of the chart that the maximum thrust at take-off speed is 8200 lb. as against 7700 lb. for the compromise propellers, an increase of 6.5 per cent.

The four-blade propeller does not appear to offer any appreciable improvement in take-off. It is noted, however, that, compared with an 18-ft. three-blader, the four-blade propeller in 16.5 ft. gives almost identical results. Above 18 ft. diam-

eter the three-blade propeller gives superior thrust which cannot be equaled by the four-blade propeller in an equal or smaller diameter.

Cruising and High-Speed Efficiencies

From the take-off standpoint, a three-blade propeller with a diameter of about 22 ft. operating at 900 ft. per sec. tip speed will give static thrust 13 per cent higher than either an 18-ft. three-blade or a 16.5-ft. four-blade propeller operating at 1000 ft. per sec. tip speed, and the thrust at 80 m.p.h. will be 6.5 per cent greater than with either of the smaller propellers. The 18-ft. four-blade propeller is so nearly equivalent to the 18-ft. three-blade for take-off thrust that it offers no advantage from this standpoint.

Looking at the high-speed and cruising charts of Figs. 1 and 2, it is noted that the diameter required for best efficiency varies with altitude. However, one very important fact is shown by the method of presentation. This is the fact that there is a narrow range of diameters when a single propeller will give within 1 per cent of the maximum efficiency at any altitude between 10,000 and 30,000 ft.

On the basis of the 2500 hp. for take-off rating, the diameter for high speed will be taken assuming 2100 hp. as the normal rating. Diameters for cruising at 40 per cent of normal power at 55, 65 and 75 per cent of normal r.p.m. are

[2] See S.A.E. TRANSACTIONS, Vol. 32, September, 1937, pp. 400-404, 427: "The Determination of Ratings for Transport Aircraft Engines," by R. F. Gagg.

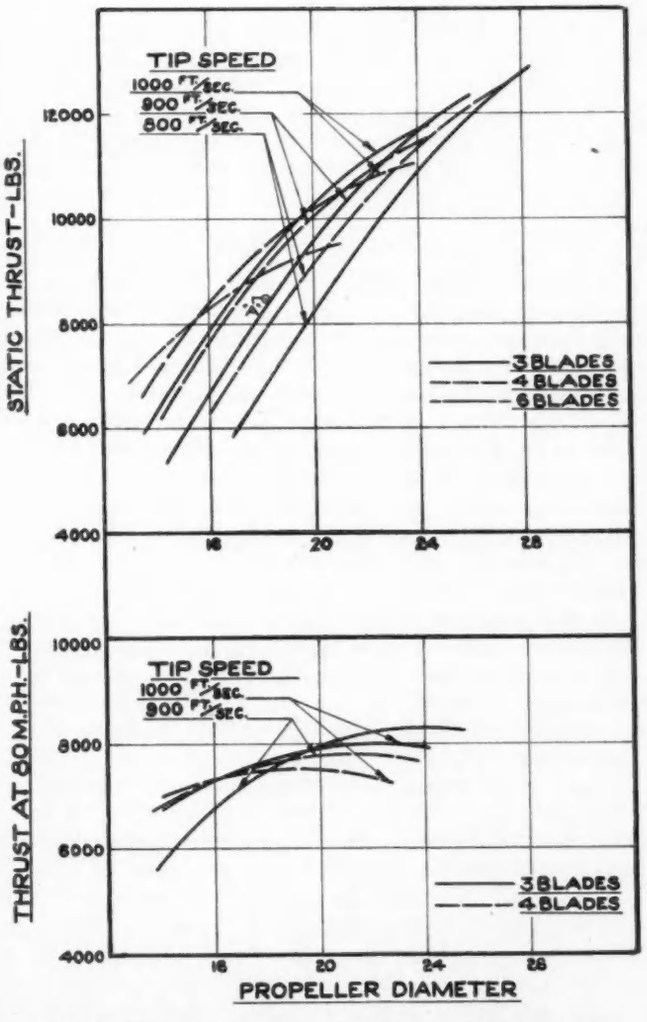


Fig. 7 – Thrust Vs. Diameter at Different Tip Speeds for a Take-Off Rating of 2500 Hp.

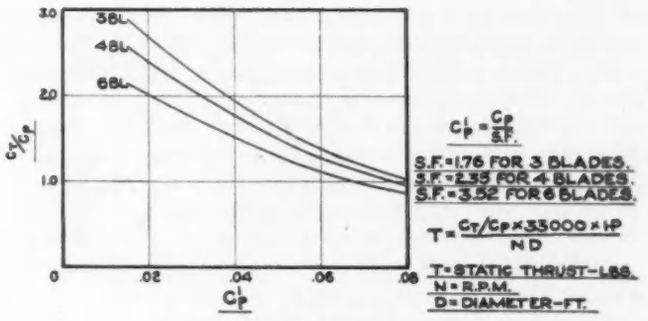


Fig. 8 – C_T/C_P for Static Thrust

obtained from Figs. 1 and 2 from both three- and four-blade propellers. Data may be summarized as follows:

	Diameter, ft.	
	Three-Blade	Four-Blade
High Speed	19.0	16.8
Cruising, 40 per cent power 55 per cent r.p.m.	24.0	21.0
Cruising, 40 per cent power 65 per cent r.p.m.	21.0	18.5
Cruising, 40 per cent power 75 per cent r.p.m.	17.5	15.5

For long-range high-altitude cruising it is believed that the 40 per cent of rated power just used is representative. For this condition, from data given by Gagg[2], best fuel economy is obtained at about 55 per cent of rated engine speed. At the same 40 per cent power but at 75 per cent of rated engine speed, fuel consumption is approximately 4 per cent greater than at 55 per cent of engine speed.

It appears desirable, therefore, to use the large-diameter propeller in order to maintain maximum propeller efficiency at low r.p.m. for maximum fuel economy. Since this same larger diameter gives greater take-off thrust and, hence, would permit larger loads being taken off, it appears doubly attractive. A final decision, of course, would have to balance the increased load and range against the added propeller weight.

As an alternative to the use of the larger propeller, it might be possible to improve engine characteristics so that the low fuel consumption now obtained at the low r.p.m. also can be obtained at the higher r.p.m., thus permitting cruising with the smaller propeller at best economy. This improvement may not be possible of accomplishment, however, due to the inherent characteristics of the engine whereby it is the reduction in friction losses at the lower crankshaft speeds that is principally responsible for the gain in economy at lower r.p.m.

The other alternative is to use a two-speed reduction gear on the engine so that, for cruising at the assumed 40 per cent power, it will be possible for the engine to operate at 55 per cent rated speed while the propeller is turning at a speed corresponding to 65 to 75 per cent rated r.p.m.

Actual decision to use a two-speed reduction gear would depend upon the comparison of a 4 per cent increase in economy and the added weight of the gear. It seems probable that it will be of advantage only for very long-range aircraft operating at altitudes over 20,000 ft.

Three Vs. Four Blades

The four-blade propeller might be a compromise which would eliminate the necessity of the variable gear at least at the 20,000- to 30,000-ft. altitudes, although even with an 18-ft. four-blade propeller it will be necessary to cruise at 65 per cent of rated r.p.m. With the 1 per cent loss in efficiency between three and four blades and the 2 per cent increase in fuel consumption due to the higher r.p.m., a total increased fuel

consumption of 3 per cent would result, or almost all the saving it was hoped to make would be cancelled. Hence the three-blade propeller either of the large diameter of about 22 ft. or of a smaller diameter of 18 to 19 ft. with a variable gear appears to be the most desirable. From the static-thrust analysis made earlier it appears probable that the airplane could take off with about 8 per cent more gross weight with the 22-ft. propeller and, hence, this propeller might be desirable. The added weight caused by the extra diameter would be partially compensated for at least by the elimination of the two-speed reduction gear.

Rating the several different propellers discussed gives the following order of merit:

1. 22-ft., three-blade – single-speed reduction gear.
2. 18-ft., three-blade – two-speed reduction gear.
3. 18-ft., four-blade – single-speed reduction gear.
4. 16.5-ft., four-blade – two-speed reduction gear.

Best Diameter

It is realized that the best diameter is considerably larger than would be obtained on the basis of current practice. Spotting on Fig. 6 an 11 ft. 6 in. diameter propeller such as is used in a number of installations on engines rated at 1000 hp. for take-off and extending a curve, *A*, through this point and parallel to the family of curves, it is seen that, for the 2500 hp. for take-off engine used in this study, a diameter of slightly over 18 ft. is indicated.

From the standpoint of the propeller designer, the smaller the diameter can be kept, the better, since propellers of 18-ft. diameter can be developed more readily than those of 22-ft. diameter. For the medium-range aircraft the 18-ft. three-blade propeller is probably satisfactory. However, from the standpoint of the ultimate in performance, the three-blade propeller of about 22-ft. diameter appears definitely necessary. This diameter is located as a point on curve *B* of Fig. 6 for purposes of comparison.

On Fig. 9 are shown curves indicating the range of gear ratios necessary for the larger powered engines. The upper curve for each engine r.p.m. represents the case of the 18-ft. propeller for 2500 take-off hp., whereas the lower curve is the ratio necessary for the 22-ft. propeller. High-speed, high-performance types of aircraft require propellers of smaller diameter than those discussed here, and gear ratios for this type will be higher than shown on Fig. 9.

Application of the results of this study to engines with other ratings may be made by means of the curves *A* and *B*

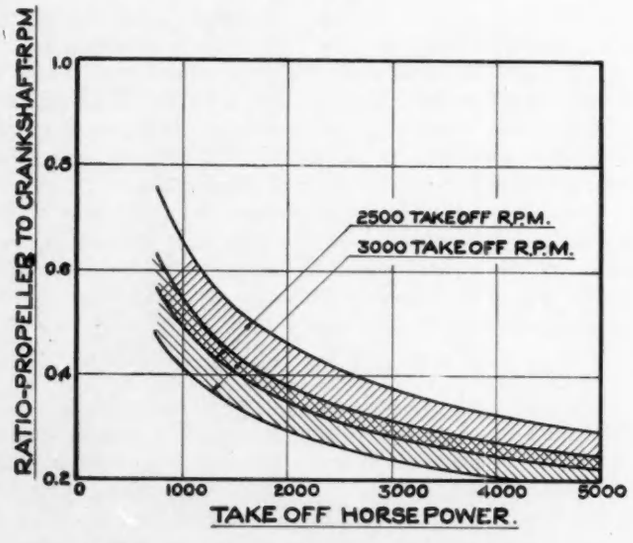


Fig. 9 – Range of Gear Ratios Necessary for the Larger Powered Engines

on Fig. 6. The upper curve, *B*, indicates the diameters for operating conditions equivalent to the 22-ft. propeller for 2500 take-off hp., whereas the lower curve, *A*, represents the compromise diameter. For airplanes having speeds other than that assumed for the example, allowance can be made by means of the correction factor of Fig. 3.

In case the variable reduction gear is adopted for use, it should be mentioned that it will be essential to provide a safety means whereby it is impossible to increase engine speed when in the high gear to an amount exceeding the rated propeller r.p.m. This safety device might take the form of an inter-connection between governor control and gearshift lever or it might be done more simply by driving the governor from the propeller shaft.

As byproducts of the foregoing analysis, two considerations develop which it is hoped may be of general interest. One is that a much clearer picture of the application of any given engine to a specific airplane design problem could be obtained if engines were rated in terms of propeller-shaft r.p.m. rather than crankshaft r.p.m. From the standpoint of airplane performance, it makes no difference if an engine of a given power is rated at 1500 or 5000 crankshaft r.p.m. so long as the propeller r.p.m. is the same. The other consideration is that a desirable figure of merit on which to base engine and propeller selection and which could be made use of more generally for application to heavily loaded aircraft is lb. propeller or engine-propeller weight per lb. take-off thrust rather than, or in addition to, the usual lb. per hp. This figure would give a definite index as to the propeller or engine-propeller combination which is most efficient in taking off large loads. Take-off thrust should be a weighted value of the thrust over the range of take-off speeds, possibly the mean between static thrust and thrust at 80 m.p.h. as calculated previously.

Of the propeller factors to be determined by aerodynamic analysis, diameter, number of blades, and gear ratio have been discussed in detail and blade width and plan form briefly. Consideration of pitch distribution and shank fairing remains.

Pitch Distribution

Accurate test data on the effect of pitch distribution on propeller characteristics is very meager. Flight test experience with Curtiss single-piece propellers indicates, however, the importance of this factor, although the results usually have been tied up with a variation in shank fairing. It has long been standard practice with the Curtiss type of forged single-piece propellers to manufacture the blade with a constant pitch from the 24-in. station to the tip rather than to use a compromise pitch distribution. At the same time, of course, the single-piece propeller permits a section of approximately airfoil shape all the way into the hub so that lower drag and better cooling in radial-engine installations at least may be expected.

As both pitch and shank fairing are different among propellers used in comparative flight tests, it is difficult to know exactly how much to attribute to each factor of high-speed gains of the order of 1 to 2 per cent and equivalent improvements in take-off. Improvement due to the combined effect is definite, however.

The same situation holds true with regard to tests with controllable propellers except that, for this type, the pitch distribution is definitely the more important factor. Calculations made by the modified-blade-element theory indicate that a 6 per cent improvement in maximum efficiency is possible in a blade with pitch designed for a V/ND of 1.6 as compared with one designed for a V/ND of 0.8 but working at V/ND of 1.6. Such flight tests results as are available, although not entirely consistent, tend to confirm this figure. Static thrust measurements between blades of the same plan form but having pitch distribution as plotted as *A* and *B* in

Fig. 10 showed that the higher pitch blade is practically as good for take-off as the lower pitch blade. There is some evidence, however, to indicate that, for climbing speeds, there may be a slight difference in favor of the low-pitch blade. The gain in high-speed and cruising efficiency, however, is enough definitely to indicate the advantage of using a pitch distribution designed for the operating V/ND. Also in Fig. 10 is plotted pitch distribution of a third blade such as would be required for operating V/ND's of the order of 2.4.

Shank Fairing

As mentioned previously, tests results on differences in shank fairing have been tied in with changes in pitch distributions so that specific results are not available. However, from improved engine cooling in radial-engine installations obtained by bringing the airfoil section in closer to the hub, it is indicated that an improved airflow is obtained which, in the majority of cases, will result in lower drag.

Strut sections having a fineness ratio of 1.5 have a drag of 10 per cent that of a circular section. Blade shank sections having such an aspect ratio if set at the correct angle so as to be in line with the relative wind will reduce propeller drag even though they may not contribute to thrust.

Current practice on Curtiss controllables is to bring the blade section down as far toward the shank as structural considerations will permit so that the section with a b/h or fineness ratio of 1.5 is from 16 per cent to 19 per cent of tip radius.

For larger diameter propellers it is doubtful if the h/b point can be brought in appreciably closer. For radial-engine installations, this point is probably not particularly important but, for in-line engines or engines with extension-shaft drives, the shank fairing must be designed carefully so that the length of high-drag shank will not be excessive, and the propeller spinner proportions must be chosen to enclose as much unfaired shank as possible.

Some calculations made on the basis of propellers of about 15-ft. diameter installed on a clean nose such as may be obtained with liquid-cooled engines with spinners of 24-in. diameter show that the drag of the shanks of a three-blade propeller having blades of the Curtiss type is of the order of 10 hp. at 225 m.p.h. at 10,000 ft. and 24 hp. at 350 m.p.h. For blades having round sections out beyond the 12-in. station, these figures would increase appreciably, reaching values of the order of 100 hp. at 300 m.p.h. at 10,000 ft. for round shanks 5 ft. in diameter extending 10 in. from the spinner. The necessity for proper blade shank shapes for this type of installation is very evident.

Propeller Torque

For the case of the large engines installed in small aircraft of the single- or two-seater type having a single engine and propeller, the matter of torque reaction is of interest. Considering the problem by means of dimensional theory, propeller diameters will go up approximately as the square root of the increasing power and, hence, for the same tip speed, r.p.m.'s must decrease in proportion to diameter. The effect is for torque to increase as the 3/2 power of the horsepower or $Q \backsim P^{3/2}$.

Fortunately, assuming that, as the power goes up, the airplane gross weight will also increase and that the wing and power loadings will remain approximately constant, it is possible to determine the relation between the power of lateral and directional control and the increased horsepower. L_a, the rolling moment, which may be considered as a good measure of lateral control, will increase with the wing area and the span or as the cube of the span. Since the span varies as the square root of the weight or, assuming constant wing and power loading, as the square root of the engine power, it is seen readily that the lateral control effectiveness will increase as the 3/2 power of the horsepower or $L_a \backsim P^{3/2}$.

Hence, it appears that the increased propeller torque with the larger engines will be taken care of automatically by increases in power of lateral and directional control due to increased area and linear dimensions.

If power loading is decreased and wing loading increased, however, there will be a tendency for the relation between torque and aileron effectiveness to develop in the wrong direction, but it is not believed that prospective changes in loadings will be so great as to cause the airplane designer very great concern from the propeller-torque standpoint even with three-blade propellers. With four- or six-blade propellers, the diameter for equivalent performance will be lower than for the three-blade and, hence, the r.p.m. will be stepped up to hold the same tip speed so that the torque for the same power will be less.

Feathering

The necessity for providing feathering of the propellers in case of engine failure on multiengine airplanes is now very generally accepted because of the performance advantages resulting from feathering and also the enhancement of safety after a powerplant failure. Mechanically, feathering is of value in stopping windmilling of the propeller in a dead engine with consequent elimination of vibration and disturbed air flow which, in some cases, can be dangerously severe. Aerodynamically the performance is increased appreciably by feathering over either a windmilling propeller or a braked one. Values obtained in flight as reported by McCoy[3] for two types of two-engine airplanes show improved climb and ceiling and an appreciable increase in single-engine speed. The reported speed improvement in single-engine flight with propeller feathered over propeller windmilling in high pitch is from 5 to 8.9 m.p.h. which represents a power saving of 11 to 15 per cent. This saving may be made use of either by flying at the same power with increased speed or by reducing power to fly at the same speed as for a windmilling propeller, thus increasing reliability of the remaining engine and also increasing range. One additional characteristic observed in single-engine flight with propeller feathered is the improved handling quality and ease of control.

For airplanes of four or more engines, feathering is of still more importance due to the high air speed which may be maintained with only a single engine out of commission, thus giving rise to an excessively high windmilling speed causing great drag and heavy vibration.

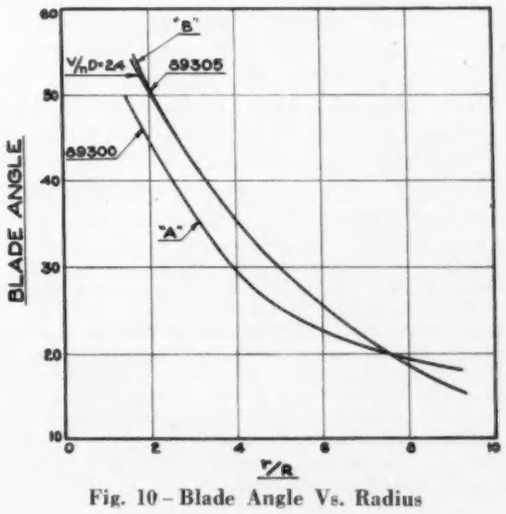


Fig. 10 – Blade Angle Vs. Radius

[3] See the *Journal of the Aeronautical Sciences*, May, 1938, pp. 253-259: "Full-Feathering Propellers," by H. M. McCoy.

Propeller Clearance

As mentioned earlier in the paper, maintaining adequate propeller tip clearance with increasing diameters affects the airplane designer quite seriously. It appears probable that power loadings will remain about the same and that wing loadings will increase slightly with airplanes of higher gross weight; hence airplane linear dimensions will go up slightly slower than the square root of the ratio of gross weights and, consequently, slightly less than the square root of the ratio of the power. Since propeller diameters, keeping the number of blades constant, go up directly as the square root of the power ratio, clearances in percentage of diameter will decrease slightly so long as geometrical similarity is maintained. Clearances in feet will probably remain the same.

Assuming that airplanes of 50,000 lb. gross weight require four 12-ft. diameter propellers driven by 1000 hp. for take-off engines and have a propeller tip ground clearance of 2 ft., then the airplane of 200,000 lb. gross weight will require four 4000-hp. engines with about 24-ft. diameter propellers and, assuming an increase in wing loading between the two of 10 per cent, the propeller tip clearance will decrease by about 5 per cent in terms of propeller diameter but will increase in absolute value to 3 ft. 9 in.

However, if the propeller diameter is increased for the long-range type of aircraft to obtain maximum performance as outlined earlier in this paper, clearances again will be changed. Diameters for the 4000-hp. engines for propellers designed for the maximum take-off and range will be about 28 ft. so that clearance will be reduced by 2 ft. to 1 ft. 9 in.

Without predicting the form that aircraft will assume with increasing size, it is impossible to say positively that propeller diameter will not be a limiting factor to powerplant size but, since airplane linear dimensions increase very nearly in direct proportion to propeller diameter, it does not appear likely that difficulty will be encountered.

In conclusion it may be stated that, from the aerodynamic standpoint, propellers of the type considered having blades of fixed diameter and of fixed-pitch distribution but rotatable about their axes in a controllable hub can be designed to operate efficiently for engines of any powers. As operating altitudes increase, it is essential that the propeller be selected for the most important flight conditions but one propeller, if properly selected, will give within 1 per cent of the maximum efficiency for altitudes 10,000 ft. above or below the design altitude. For maximum performance with these large propellers, it will be necessary that operation of the engine-propeller combination be controlled very carefully so that the predicted performances can be achieved.

Discussion

Stresses Need for Shank Fairing

– T. P. Wright

Curtiss-Wright Corp.

MR. BRADY also is optimistic that propeller engineers will solve the problem of propeller design so that it will not be permitted to act as a limitation to increasing the size of the individual powerplant. One encouraging point brought out in his paper in this regard is the fact that the hub weight will not go up so rapidly as the blade weight which fact, coupled with the beneficial effects of hollow material, will make possible the design of propellers of even large sizes at reasonable weights.

Mr. Brady brings out very definitely the probable need for designing engines with two-speed gears and means of shifting from one to the other. He indicates that the increased weight involved must be considered in determining the benefits of such procedure.

I was impressed with his discussion concerning the redistribution of air loading of propellers along their radii under different speed conditions, indicating the greater importance of inboard portions as the speed increases. This point would seem to indicate some change in plan form, particularly when considering another point raised by Mr. Brady, that is, the need for fairing the shank inwards towards the hub as much as possible, preferably into a spinner on certain installations. To emphasize the importance of Mr. Brady's contentions in this regard, I would like to mention information I received in England a year and a half ago which indicated that, when trying out two different propellers differing from each other chiefly in regard to this shank fairing, and on a very high-speed airplane, it was found that the speed increase due to the shank fairing was 11 m.p.h.

I feel that Mr. Brady lays proper stress on the importance of full feathering, both from the standpoint of enhancement of safety in the event of powerplant failure and because of improved performance.

Propeller Problems Imposed by Substratosphere Flight

(*Continued from page* 288)

torque by permitting the tip speed to go to 1100 ft. per sec. is shown in Table 3. Although diameters and torques are reasonable, with the possible exception of the three-bladed propeller, the performances of the single rotating propellers are still inferior to that of the tandem propeller.

Table 3

Propeller	3-Blade	4-Blade	Tandem
Diameter, ft.	14.0	11.53	11.45
Gear ratio	0.429	0.518	0.464
Efficiency tip speed at maximum airplane speed, ft. per sec.	1100	1100	1000
Propeller efficiency at maximum speed	0.756	0.752	0.810
Propeller efficiency at cruising	0.789	0.785	0.810
Propeller efficiency at sea-level climb	0.708	0.700	0.710
Torque at take-off, ft-lb.	9370	7750	0.000

Although this comparison strongly favors the tandem propeller, it must be remembered that the performance indicated is based on only one model test and is, therefore, not as reliable as those indicated for the single rotating propellers which are based on a large number of tests. Furthermore, the effect of tip speed for the tandem propeller is not definitely known since the tests did not cover sufficiently high rotational speeds and, although the same correction for tip speed was applied for both the single-rotating and tandem propellers, it may very well be that it should be more drastic for the tandem.

No evaluation of take-off thrusts was possible in the comparison because of insufficient data. It is quite probable that the blade interference effects are sufficiently severe to make the tandem propeller definitely inferior to the single-rotating ones in this phase of performance.

The performance comparison indicates that the counter-rotating propeller should be considered definitely for the single-engine airplane, particularly where torque is a problem. However, the practicability of such a design depends on the satisfactory solution of operational, as well as weight problems. Before this phase of the work is undertaken, however, further testing must be accomplished to investigate all phases of performance completely.

To sum up: high-power substratosphere flight will require the close coordination of airplane, engine, and propeller designers in order to insure the most efficient conversion of power. This type of flight is going to require propellers which are appreciably heavier than present ones. Although weights will increase, it is expected that the upward trend in specific weights can be reduced appreciably. Counter-rotating propellers appear to have some possibilities where torque is a factor and where sacrifices can be taken on take-off performance.

Why Drain Crankcases and When

By G. A. Round

Socony-Vacuum Oil Co., Inc.

REPORTS on fleet operations from many different sections of the country show a wide variation in crankcase draining practice ranging from what we would consider entirely too frequent changes to no draining at all. Some have claimed that the use of efficient filters has eliminated any need for draining except perhaps with seasonal oil changes; others have learned, at no small expense, that this condition is not always so and that apparently, under some operating conditions, draining at very short intervals is imperative.

Since those who are having trouble do not advertise it except to their suppliers and those who feel they are saving a good deal by never draining are rather vociferous about it, a fair number have been led to believe that crankcase draining is a fallacious idea. Motor-car and filter advertising have fostered this opinion to a considerable degree so the daily mail shows a fair percentage of inquiries on the subject, asking when and sometimes why. The answers given here represent an attempt to present the facts as the author sees them today.

Why Ever Change?

If engines were lubricated by the "all-loss" method of twenty-five years ago, there would be no need of draining, the oil passed over the bearings once – or perhaps a few times – and then thrown away or burned up as fast as it came in. There was not much chance for contaminants to build up and cause trouble. Today a few quarts are put in the crankcase and, more often than not, they are expected to remain there indefinitely. Oil is bought (and sold) to last rather than to lubricate. At least most of the emphasis is on economy rather than on overall results.

Under such conditions, there is ample opportunity for lubricating oil to become contaminated progressively with materials which may be formed or brought into the engine or produced in the oil as a result of use. These contaminants are:

1. Water
2. Soot
3. Lead Salts
4. Fuel Ends
5. Dust
6. Metals
7. Oil Decomposition Products

Water Conditions Are Serious

In the author's opinion, water in the crankcase represents the most dangerous type of contaminant. It can freeze and break an oil pump; it can emulsify with the oil and clog the lubricating system with disastrous results; it can mess up the inside of an engine so that nothing short of a tear-down will make it run right; combined with permanent anti-freeze solutions, it can gum rings, pistons, and valves in short order. And every time there is a fairly prolonged spell of cold weather, it does all of these things in a surprisingly large number of fleets in spite of all the warnings that have been published.

One would think by now that most everyone would know how and why this trouble happens but we still hear the question, why?, and so we repeat the answer.

Barring cylinder-head gasket and block leaks, water gets into the crankcase through condensation of the steam formed by fuel combustion. For every pound of gasoline burned, slightly over a pound of water is formed as steam by the combination of the hydrogen in the fuel with the oxygen in the intake air. If the cylinder walls are cold, some of the steam condenses on them and is carried to the crankcase by the rings. Some steam undoubtedly gets there by direct blowby and may or may not condense, depending upon the degree of saturation of the crankcase atmosphere.

Where vehicles are operated on relatively long routes or under sufficiently severe service to maintain high jacket temperatures and good crankcase ventilation, water accumulations are rarely troublesome. Thermostats, radiator shutters, and adequate air flow through the crankcase have all been very helpful in reducing condensation and in eliminating the corrosion troubles which, before the advent of these devices, were a common cold-weather occurrence when high-sulphur fuels were used.

In those delivery services where short runs, many stops, and much idling are the rule, there is trouble every winter from water and sludge. The engines never get really warm – cab heaters do not help this condition – crankcase ventilation with ordinary systems is just about nil and combustion is decidedly on the rich side.

What Winter Sludges Consist of

Under such conditions water accumulates at an astonishing rate. For example, during the severe winter of three years ago, it was necessary to drain the crankcases of a fleet of city milk-delivery trucks every third day – every 54 miles – to prevent tie-ups!

What happens in some cases is illustrated well by the analysis given in the following table:

A Typical Water Condition Oil Used 262 Miles

Used Oil Analysis		Crankcase Sludge	
Viscosity at 130 deg. fahr.	106 sec.	Oil – Soluble in P.E.	57.0 per cent
A.S.T.M. Dilution	9.60 per cent	Oxidized Oil, Soaps, and so on	0.5
Insoluble Material	1.03	Carbonaceous Matter and Metals	5.5
Oxidized Oil – Soaps, and so on	0.11	Water	37.0
Ash – Iron and Lead	0.44	Ash	2.0
Water	None	In Ash – Lead and Iron	—

The used oil is rather light in body due to dilution; it contains considerable foreign material consisting of blowby carbon (soot), lead salts from the fuel, and iron resulting from rusting and wear. There is only a trace of oxidized material from the oil; it has been in service too short a time at too low temperatures for appreciable chemical change to take place.

Strangely enough there is no water in the main body of the oil. What has happened is that it has formed an emulsion – mayonnaise dressing – with a portion of the oil, carbonaceous matter (soot), lead salts and traces of oxidized oil acting as binders to hold the two liquids together. This mixture is shown by the second analysis in the preceding table.

The resulting mixture – sludge – settled and centrifuged out of the main body of the oil, blocking the oil screen and causing a bearing failure. The major portion of the oil remained fairly

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clean, which condition emphasizes the importance of getting representative samples when studying complaints, and the larger the samples the better.

What Can Be Done About It

With present designs, some trouble from water accumulations is almost inevitable during any prolonged period of severe cold, especially in intermittent service. The use of high-temperature thermostats (with permanent anti-freeze solutions) is a help. Radiator covers also aid because they keep some cold air off the crankcase, but the oil reservoir is usually unprotected in any case. All crankcase ventilators should be checked to see that they are free and clear. Recently we found a fleet where the air cleaners in the breather intakes had never been cleaned, although the carburetor air cleaners had been serviced regularly. Getting air through the crankcases cleared up this trouble case.

So far we have had no experience with the package car designed for steam cooling. This design should help and indicates that the designers realize the problems connected with light-duty service in cold weather.

Many have asked if something cannot be done through the oil to eliminate such troubles. We believe not. If oils are made so that they will emulsify readily with water, the entire mass of lubricant is likely to thicken to a point where it will not circulate – in fact this emulsification sometimes happens when the right combination of oil, water, and crankcase dirt occurs. In addition, the water in the oil may freeze and shut off circulation.

Oils for steam-turbine service are highly resistant to emulsification, but experience with them in fleet operation as well as in the laboratory shows that they give no better service than high-grade well-refined motor oils. The reason is that soot and lead salts are powerful emulsifying agents which hold oil and water together. Hence, the real problem is to prevent water accumulation.

Filters Not Effective

Will filters help? In our experience they are practically useless and perhaps a handicap because they give the operator a false sense of security. The reason for their ineffectiveness is their small capacity in comparison with the amount of water and sludge to be handled.

Just a few weeks ago a passenger car was fitted with a highly efficient crankcase ventilating system having a trap to catch everything coming out of the ventilator. This car was driven a distance of 150 miles in intermittent service during very cold weather, after which approximately ½ pt. of water was taken from the trap. Obviously, no filter could be expected to cope for long with such a situation.

The analysis shown in the following table indicates what erratic results may be encountered in a single fleet, with and without filters. Here two units of identical design are operating under similar service conditions, the filter-equipped unit probably running colder. Obviously, the device was not taking care of the trouble and we would not expect it to – which is not a blanket condemnation of filters. They have their place.

	Unit A With Filter	Unit B No Filter
Mileage since draining	500	1500
Viscosity at 210 Deg. Fahr., sec.	57	55
A.S.T.M. Dilution, per cent	12.0	9.6
Insoluble Material	0.60	0.21
Oxidized Oil, Soaps, and so on	0.10	0.12
Water	21.5	None
Ash	0.23	0.52
In Ash – Lead and Iron		

It is the author's belief that trouble from water accumulations, except those due to direct leaks, can be overcome by providing adequate ventilation under idling, low-speed, and light-load conditions. Means of doing this ventilating are now on test, and the reports thus far are highly encouraging. This work is important because the conditions which promote water accumulations also induce high wear while the damage and loss of time resulting from winter sludge trouble often are large expense items.

Soot

As already mentioned, soot is the product of rich mixtures, especially in cold operation. Some is always formed during the starting and warm-up periods but, with normal operation, there should be little of it, and only a very small amount should reach the crankcase.

We know of no evidence that it causes wear or other damage except as it may act as an emulsifying agent to cause the winter sludge troubles already described. Ordinarily, an efficient filter will remove it from the oil as fast as it reaches the crankcase.

Soot, of course, will discolor oil and the lighter colored the oil, the more marked will be its effect. In this connection it is interesting to note that rapid discoloration of oil often indicates poor condition of the rings. In other words, with efficient rings, the oil tends to stay clean. Once blowby starts, more soot reaches the crankcase, the piston temperature goes up, oil oxidation is more rapid, and a whole chain of conditions promoting rapid contamination of the oil supply is formed. To a certain extent, therefore, oil condition may be considered a barometer of engine condition and, when piston-rings are nearing the end of their useful life, that fact often will be indicated by oil analysis.

Lead Salts

A common complaint these days is that crankcase oil turns gray or brown after a short period of use. This discoloration affects all types of oils including the darker colored ones, although the light-colored oils – as always – show discoloration more readily. (A black spot always looks worse on a white shirt.)

This condition is due to the presence in the oil of lead salts from the combustion of leaded fuels. It is common experience to find combustion-chamber deposits light gray or even nearly white in color due to the presence of these compounds so it is perfectly obvious that a small quantity must reach the crankcase. Every analysis of crankcase oil from engines using leaded gasoline shows this to be a fact, as indicated by the data in the following table:

	Sample A	Sample B
Color	Grayish-Tan	Brownish-Gray
Viscosity at 130 Deg. Fahr., sec.	127	208
A.S.T.M. Dilution, per cent	4.8	Trace
Total Insolubles, per cent	1.28	0.14
Oxidized Oil	0.11	0.03
Water	0.10	None
Ash	0.53	0.35
In Ash		
Lead Oxide	76.3	89.1
Silica	8.3	—
Iron Oxide	15.2	11.7
Service	500 Miles Truck Trailer	100 Hr. Stationary Power Unit

Sample A is quite dirty, containing a relatively large amount of carbonaceous material as well as considerable ash. There is more dilution than might be expected in heavy-duty service, so the combination of fuel and carbon indicates inefficient combustion. The presence of a fairly large amount of lead is indicated clearly. Rarely do we see as much silica – road dust – but this material is accounted for in part by the unit operating in the dust area of the Middle West. Sample B is fairly clean but contains enough lead to discolor it. This unit is obviously operating very efficiently.

Careful study of many of these samples shows that, as a

general rule, some of the lead salts are sufficiently coarse to settle out of the oil if the sample is left standing. These salts ordinarily are taken out by oil filters. However, some of the lead compounds are so finely divided that they cannot be filtered even with laboratory equipment, and these compounds are largely responsible for the brownish color of the used oil. Some of the lead also may be dissolved in the oil, as indicated by Sample B, the percentage of ash being higher than the total insoluble material.

There is no evidence that the lead does any harm other than adding to the amount of crankcase dirt and aiding in emulsion formation when water is present. It does, however, alarm those who have not happened previously to notice the discoloration. Since most oil companies have issued bulletins on the subject, more should know about it and the complaints therefore would decrease.

Dilution

With modern engine designs, dilution of the oil – per se – rarely causes trouble. In fact, most passenger-car manufacturers recommend diluting S.A.E. 10 motor oil up to 10 per cent or more with kerosene to assure easy cranking at temperatures below —10 deg. fahr. There are points for and against this practice but they need not be discussed here.

There are several reasons why fuel-thinned oil can be used in present-day engines: First, we have oil pumps with adequate capacity to keep the bearings full even with hot badly diluted oil. Second, the bearing clearances have been reduced so that end leakage is lessened greatly and, what is more important, the load-carrying ability of the oil film is increased – reducing clearance has exactly the same effect as increasing viscosity. Third, modern piston design and ring equipment are less sensitive to oil-viscosity variations – at least in the range where there is likely to be complaint. Even the lightest oils give good mileage, at least in passenger cars.

On the other hand, if bearing clearances are relatively large and if the oil pump has insufficient capacity to keep the bearings full and maintain proper oil pressures, failures may occur from lack of lubrication or from bearings pounding out. This was the condition not so many years ago. Then too, diluted oil usually burned up fast and sometimes led to complaints of poor economy. By contrast, sometimes the fuel ends accumulated fast enough to cause the oil level to increase.

One thing is certain: close clearances and low-viscosity oil put a premium on oil cleanliness. Dirt and dust can aggravate wear under these conditions and this situation is where filters are distinctly helpful.

Dilution conditions should not be disregarded, however, because they are a definite indication of something wrong, and present possibilities for trouble elsewhere than in the crankcase. Cold operation, leaking, fuel-pump diaphragms, defective automatic chokes, and carburetors out of adjustment are things to look for if dilution becomes apparent.

Road Dust

During the dust storms of recent years, there was ample proof that engines could be ruined in short order by the conditions which prevailed, even when equipped with air cleaners. Granted that these were extreme cases, the fact remains that it has proved wise to use large and highly efficient air cleaners when operating under normal conditions in the Middle West and where hard surfaced roads are the exception not the rule.

Here in the East the air is relatively clean but, even so, it has been shown that from 1 to 4 tons of dust per sq. mile fall each day. Hence, it is good judgment to service air cleaners regularly. Since none of them are 100 per cent efficient, it is obvious that some cylinder and ring wear may be attributed to dust.

How much bearing wear is caused by dust is difficult to say but, in any event, most of this dirt must pass the rings and cause wear there before it reaches the crankcase. Hence the carburetor air cleaner must be depended upon very largely for protection from this contaminant. Oil filters will help, of course, particularly with material coming in through the breather. Here again we would like to emphasize the need of regular cleaning of air cleaners in breather intakes. Our experience has shown repeatedly that they are sadly neglected.

Metals

Crankcase-oil analyses almost invariably show the presence of metals, chiefly lead and iron. Where aluminum pistons, copper-lead bearings, and so on, are employed, traces of these metals will also appear in the ash.

The effect of lead compounds in the oil already has been discussed. How much wear is caused by the other metals is difficult to say; we are more inclined to feel that their presence in the oil is the result of wear, not the cause of it. A good reason for believing this is the fact that iron is usually present as iron soaps rather than as metallic iron. Iron oxide from rusting is also present.

Some investigators have attempted to make a direct correlation between wear rate and amount of iron in the used oil. The theory seems reasonable, but the problem of measuring wear in well-lubricated engines is exceedingly difficult, so we would hesitate to say that direct comparisons were practicable even though we believe that a high iron content in the oil – along with water, excess dilution, or dust – is an indication that something is wrong and that wear is taking place as a result.

With these facts in mind we do not believe that any one method of keeping metals out of the oil is effective. A filter will undoubtedly remove suspended but not dissolved metals. Some of its limitations in other directions already have been discussed. The important thing is to study all conditions and to correct each one which appears to be incorrect.

Types of Oil Decomposition Products

When oil is subjected to high temperatures (over 250 deg. fahr.) in the presence of air, it tends to oxidize. This action is accelerated enormously by the presence of metals which act as catalysts. If temperatures are high enough, the oil also will crack.

Cracking produces carbonaceous insoluble materials. Oxidation results in the formation of a wide variety of compounds ranging from acids soluble in the oil to insoluble asphaltic materials and lacquers. The exact nature of these materials is difficult to determine, and the type or types formed in any given case depend upon a great many factors including engine design, operating temperatures, character of oil, and length of time it is used.

When the word "acid" is used, there is an involuntary feeling on the part of many that corrosion inevitably will occur. This reaction has been inferred in connection with the use of acids in oil refining as well as their development in service. However, actual corrosion troubles have been small in number and the facts regarding them are as follows:

Reference to the 1936 report of the A.P.I. Automotive Survey Committee will show that serious corrosion trouble was confined to one make of car using cadmium-silver bearings. It developed only in sustained high-speed driving; it occurred with practically all high-grade oils normally considered most suitable for automotive use and was undoubtedly due to abnormal piston-head temperatures. It did not occur with other makes using the same bearings and oils. When the engine temperatures were brought down to normal, the trouble disappeared. Since 1935, such difficulties have been rare and confined to unusual operating conditions.

In fear of trouble from this source, many have been led to

watch the neutralization number or acid value of used oil. They have also laid special emphasis on low values for new oils. In both ideas, they are likely to be misled because the type of acid is more important than the amount. This point is shown by the results of engine corrosion-test runs given in the following table. This particular unit is especially designed to cause severe oil oxidation and operates at a very high temperature with glycol cooling. At regular intervals, the oil is checked for acidity and the test bearings weighed and examined to determine their condition.

Oil Acidity and Corrosion

	Engine Results vs. Neutralization Number					Neutralization Number					
						Bearing Weight Loss					Corrosion
Hr. Run	10	20	40	60	80	10	20	40	60	80	Observed
Oil A	—	1.8	1.4	1.7	—	—	0.16	0.47	—	—	Slight at 60 hr.
Oil B	0.3	1.0	—	—	—	—	0.68	—	—	—	Marked at 20 hr.
Oil C	—	0.1	1.3	3.0	4.6	—	0.20	0.24	0.27	0.67	Marked at 80 hr.
Oil B	0.6	1.7	—	—	—	—	1.8	—	—	—	Bad at 20 hr.

The figures in the table show that, with Oil *A*, there was only slight corrosion with a neutralization number of 1.7; with Oil *B*, it was bad with a neutralization number of 1.0; whereas, with Oil *C*, there was none until the value went above 3.0. At this time, however, this particular oil had thickened so badly from oxidation that it could scarcely be drained from the engine. A check run on Oil *B* shows corrosion to be bad with a much lower value. Obviously, used-oil neutralization numbers alone are of little significance.

With regard to new oils, ordinarily the neutralization number will be very low, but some of the addition agents used as lacquer solvents, catalyst poisoners, extreme-pressure builders, and so on, raise the neutralization number considerably. Hence, it is important to know the characteristics of the oil and the service for which it is intended before basing any conclusions regarding its suitability on neutralization numbers or, in fact, on any other special tests.

Preventing Corrosion Troubles

The active acids from oxidation appear to be harmful only to the cadmium alloys, copper-lead and hardened lead bearings. Tin-base babbitts are not attacked, neither are steel, aluminum, or other engine metals. Since high temperatures and certain metals alone are involved, it is obvious that the condition is controllable through design. However, as loads are stepped up, there is more need for the harder bearing metals and more chance for trouble. Hence, other means for preventing corrosion are desirable and have been provided by some of the major oil companies. This development was described last year by Dr. George Maverick in his paper on addition agents for motor oils[1]. The treatment of these bearings with indium as described recently by Mr. Smart of Pontiac also is highly effective.

Will filters help? Some are advertised as containing neutralizing materials. Probably they would be of some benefit in borderline cases, but we doubt if there would be enough active material present to cope with really severe oxidation conditions.

Temperature Limits for Oils

In addition to forming acids of various types, oil oxidation also produces a wide variety of materials commonly called sludge which range from partially oil-soluble plastic materials to coke-like resins. Under extreme service conditions in certain engines, high-temperature oxidation produces a granular carbon-like material, most of which is too coarse to go through the oil screen and be picked up by the filter. Instead, it piles up until there is enough to block flow to the pump, and sometimes adheres to the sides of the oil reservoir in layers.

This material is soluble in chloroform, showing that it is an oxidation product rather than carbon. It develops with a wide variety of oils, but some merely thicken to a tarry condition when used in such service.

Conditions of this sort can be corrected only by lowering temperatures – particularly at the piston – by design, or by reduced loading. Engine temperatures cannot be increased without limit in the belief that the resulting troubles always can be controlled through improvements in oil. We are dealing with hydrocarbons subject to chemical change at an accelerating rate with increasing temperatures so, although better oils doubtless will be developed, the law of diminishing returns will act to nullify their value more rapidly as higher temperatures are reached. In our opinion we are not far from the limit today.

Normal oxidation products are picked up by oil filters which are decidedly helpful in keeping oil clean, provided they are of adequate capacity and serviced regularly. The coarser the material, within limits, the more easily it is handled, whereas soft gelatinous materials clog filters rather quickly. However, with high-grade oils, much less oxidation occurs and engines and filters stay clean much longer.

Where Filters Are Harmful

Under certain conditions of temperature, an insoluble lacquer or varnish-like material may be produced by oxidation. This material forms a thin coating on connecting-rods, valves, and piston reliefs, particularly in aircraft and Diesel engines. Such deposits may interfere seriously with engine performance.

To prevent these formations, particularly in Diesels, addition agents of various types have been added to engine oils. Unfortunately, however, they are removed by certain types of filters, especially those containing neutralizing agents. In fact, such filters have an unfavorable effect on addition agents generally; hence, they should not be used with the special oils which have been developed to meet the peculiar requirements of some types of engines.

When Should Crankcase Oil Be Changed?

In the author's opinion, there is only one sure answer to this question, namely: "It depends upon many things." In view of what has been said previously, perhaps there is better appreciation of why this is so. However, it may be well to summarize.

With cold operation and water actually collecting, very frequent draining may be imperative to prevent serious damage. Oil cost is negligible in comparison with the expense of repairs and loss of service of the vehicle resulting from failure to drain often enough under this condition. The same may be said of bad dust conditions.

The presence of lead deposits and soot is not especially objectionable, and ordinary filters will keep the accumulations to a safe figure if properly serviced. The same holds true for metals in the oil.

The possibility for trouble from oil oxidation depends upon the service, the engine, and the oil and only a study of the individual case will enable one to say when to drain. The same unit in different services often will present entirely different problems and require different oils, different draining practice, and filter changes.

After all, lubricating oil represents a relatively small item spent as insurance against wear, depreciation, service troubles, and loss of operating time. As usual the premium is a very small percentage of the potential value of the policy. Hence, it is decidedly worth while to study all factors before deciding what to do in any individual operation.

[1] See "Modern Trends in Modern Oils," by Dr. George Maverick, presented at the Metropolitan Section Meeting of the Society, New York, N. Y., Mar. 8, 1937.

Surface Finish Related to Wear in Internal-Combustion Engines

By Kirke W. Connor
Micromatic Hone Corp.

UNTIL recently factual investigation of wear, lubrication, and maintenance of operating tolerances in internal-combustion engines have been handicapped by the lack of factory-type equipment which would permit standardized measurement in all conditions of manufacture and service. The combined characteristics of surface character and finish in bearing and operating parts may now be determined practically and efficiently by means of the new portable Profilometer, combined with photographic or microscopic study.

Newly developed honing tools and new tool actuations now accomplish a large amount of stock removal rapidly by the hone abrading process. These new tools remove all stock which has been deformed or disturbed in previous processing under conditions of actuation which produce increased accuracy, controlled surface character, and final surface finish in which roughness is held within one or two micro-inches in some parts. In present installations, they have permitted reductions in the total number of processing operations, increased production, and effected other substantial economies in processing.

The adoption of this combined new equipment indicates new opportunities for expansion of existing investigating technique; promises greater correlation of factual data on the causes of wear, and increased efficiency in production processing.

AN entirely new vision of quality improvement in the final finish and the production efficiency of producing internal-combustion engine cylinder bores and other engine parts is indicated in the recent adoption of controlled surface finish by two of the leading motor-car manufacturers and by one manufacturer in the industrial-equipment field producing replaceable sleeves. This project comprises the accomplishment and maintenance in regular production of surface finishes in which the running average of the height of the irregularities may be held uniformly less than 5 micro-inches (millionths of an inch) for engine cylinder bores; from 5 to 8 micro-inches on pistons; and from 1 to 2 micro-inches on some parts such as crankcase main bearings, crankshaft main and throw bearings, crankcase camshaft bearings, camshaft bearings, valve stems, piston-pins, mushroom heads on valve tappets, wheel hub bearings, brake drums, and so on. The significance of this accomplishment immediately suggests the importance of utilizing new equipment now available to re-study the whole subject of wear and bearing tolerances in internal-combustion engines.

[This paper was presented at the National Tractor Meeting of the Society, Milwaukee, Wis., April 15, 1938.]

[1] A comprehensive bibliography of this paper, classified under headings of "Measurement of Surface Finish," "Machining," "Lubrication and Oil Consumption," and "Wear," may be had upon request to S.A.E. headquarters.

Engine Wear

Wear of engine bearing and operating parts should be considered in two classifications, namely: initial wear and operating wear. Continued study is establishing the fact that initial wear and operating wear are affected directly by the relative surface character of the assembled operating parts. Wear of any classification obviously must start from some condition of relative surface roughness or smoothness. It is agreed generally that the rate of wear must vary in relation to the kind of materials and the combined processing used; the design of the parts; and their interrelated ability to resist corrosion, to minimize erosion and abrasion, and to support uninterrupted lubrication at all times. The combined characteristics of the surface finish, therefore, would appear to be of major importance in any consideration of the causes of wear; yet it is interesting to note that, in reviewing the papers included in the bibliography of this paper[1], the majority of the authors make little or no mention of the surface character and finish of the parts under discussion.

Surface Generation

The ability to control the generation of final surface finish approximating the existing standard for an optical flat has been made possible by the development of new hydraulic honing equipment for cylinder bores, new mechanical tool actuations of abrasive on external surfaces, and a remarkable instrument for measuring qualitative irregularities in finished surfaces. This new honing equipment comprises greater control of unit pressures by means of balanced hydraulic actuations and, as now being used in high production, makes possible greater control of surface character and finish, increases production, and effects desirable economies in processing.

Of major importance in this development is the fact that these newly developed hydraulically actuated honing tools have advanced definitely into the scope of *heavy stock removal* under minimum conditions of molecular and structural dis-

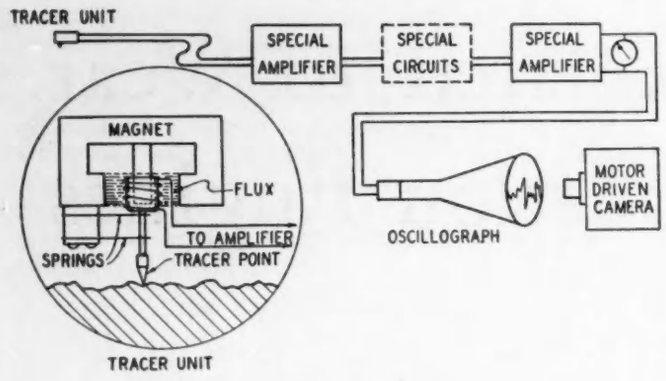


Fig. 1 – Detail of Profilometer

turbance of the metals processed, while generating substantially increased accuracy and surface finish.

The Profilometer

The reader undoubtedly will remember the Profilograph instrument which was developed several years ago; also, that this instrument was restricted, because of design and operating factors, to laboratory usage.

Until very recently, no portable type of factory equipment has been available with which to measure and maintain final surface finish as obtained in daily production. The Profilometer was developed and introduced to meet this requirement, and the fundamental elements of the instrument are shown in Fig. 1. The tracer method is used, the tracer support being arranged with a tiny coil which operates in the field of a permanent magnet. Movements of the tracer are thus translated into electrical voltages proportional to the velocity of the varying direction of tracer movement. These voltages are amplified and caused to produce "vertical" and "horizontal" magnifications of the displacement of the spot of a cathode-ray oscillograph, which magnifications are proportional to the displacements of the tracer point.

The profilometer circuit may be adjusted to separate different types of irregularities that exist simultaneously on a surface, that is, roughness and waviness. The readings provided are in micro-inches – root mean square, and represent a running average of the height of the surface irregularities. This automatic-averaging feature is one of the principal advantages of the Profilometer. Pictorial records of surface profiles may be made by suitable synchronization of tracer speed to film speed. For visual observation a suitable oscillograph "sweep" can be used.

As shown in Fig. 2, the instrument is portable and self-contained in a convenient wood case, weighs approximately 50 lb., and is designed and constructed for use at the production line, as well as for laboratory purposes.

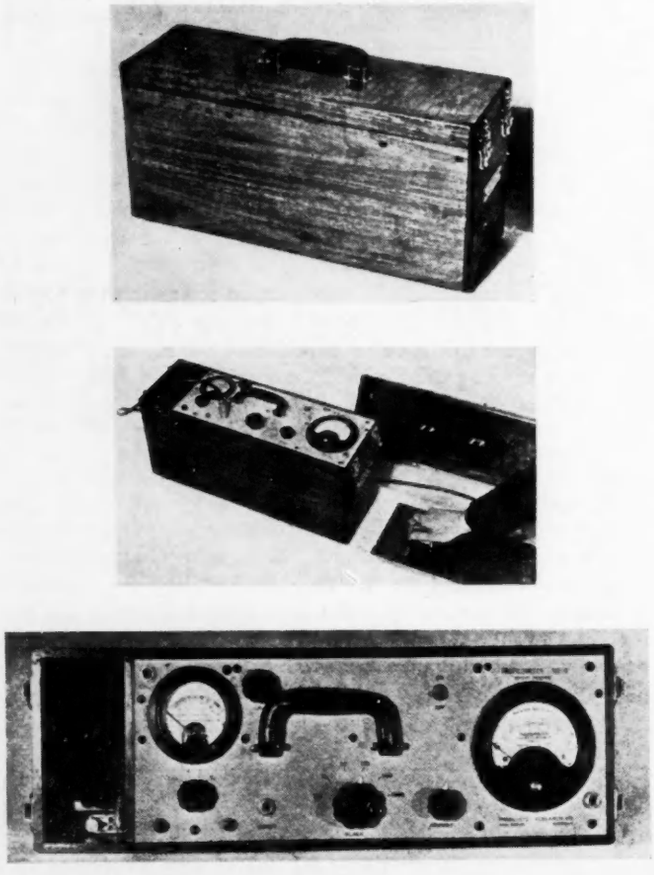


Fig. 2 – The Portable Profilometer

Cylinder-Bore Processing

An interesting adoption of the new hydraulic honing equipment previously mentioned is now being used by one manufacturer in high production of cast-iron cylinder bores cast-in-block, holding selective, controlled, uniform finish as represented by the profilograms shown in Fig. 3. The photomicrographs shown on the left were made at seven magnifications from respectively the same specimens.

Standard processing in this plant consists of rough-boring, using four-blade piloted bars with stellite-tipped cutters, removing 0.120 in. on the diameter, and it will be noted that the profilometer reading for this surface ranges from 150 to 200 micro-in.

The second operation is semi-finish boring, using three-blade stud bars with Carboloy-tipped cutters, removing 0.020 in. on the diameter, for which the profilometer reading is approximately 150 micro-in., or not materially smoother than in the rough-boring operation.

The third operation comprises precision-boring with single-point Carboloy-tipped cutters, removing 0.015 in. on the diameter, with a profilometer reading for surface roughness of 60 micro-in. In the final operation, hydraulically actuated honing tools are used to remove approximately 0.0015 to 0.002 in. on the diameter within accuracy of 0.0003 to 0.0005 in., and with surface finish profilometer reading of 8 micro-in.

Reference to the photomicrographs accompanying each of these profilograms provides an interesting study of surface character produced by these processing operations, and reveals the comparative uniformity of cutting action which has been maintained through the various operations, resulting in minimum percussion or burnishing action in process.

Oil Consumption

Surface roughness as related to oil consumption is shown in Fig. 4. It is interesting to observe that, in this engine, a cylinder finish showing an average roughness of 13 micro-in. consumed 1 qt. of oil during 300 miles of operation, whereas a finish showing average roughness of 9 micro-in. consumed one quart of oil in 1500 miles of operation.

Structural Deformation

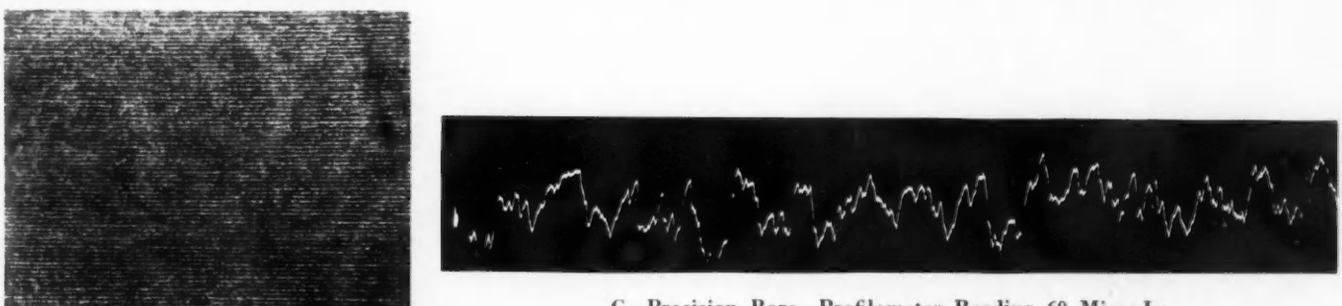
Increasing attention is being given to the effects of machining metal as related to deformation of structure. It has been indicated that the action of boring, broaching, reaming, rolling, or similar processes, produces more or less molecular and structural disturbance of the surfaces processed. The disturbance thus created may, and usually does, persist throughout successive machining processes because of the pressures



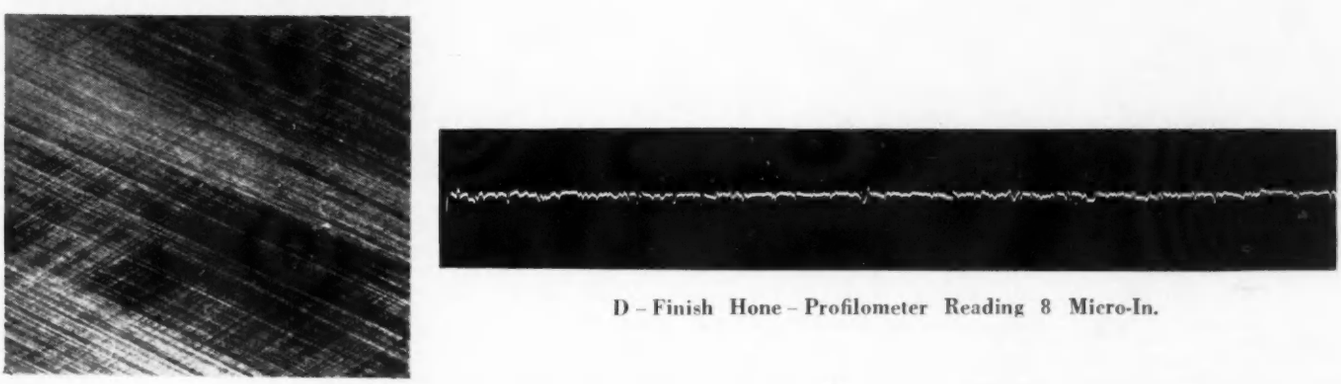
A – Rough Bore – Profilometer Reading 150 to 200 Micro-In.



B – Semi-Finish Bore – Profilometer Reading 150 Micro-In.

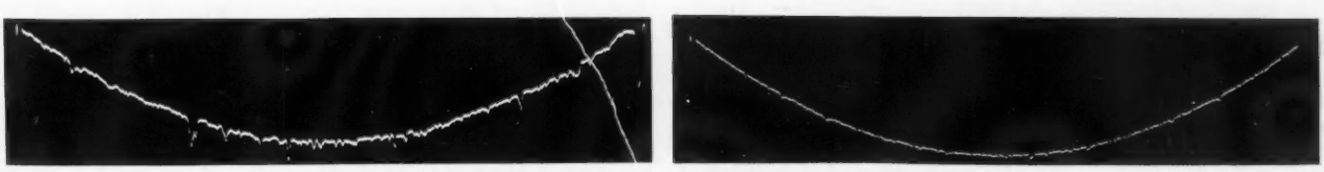


C – Precision Bore – Profilometer Reading 60 Micro-In.



D – Finish Hone – Profilometer Reading 8 Micro-In.

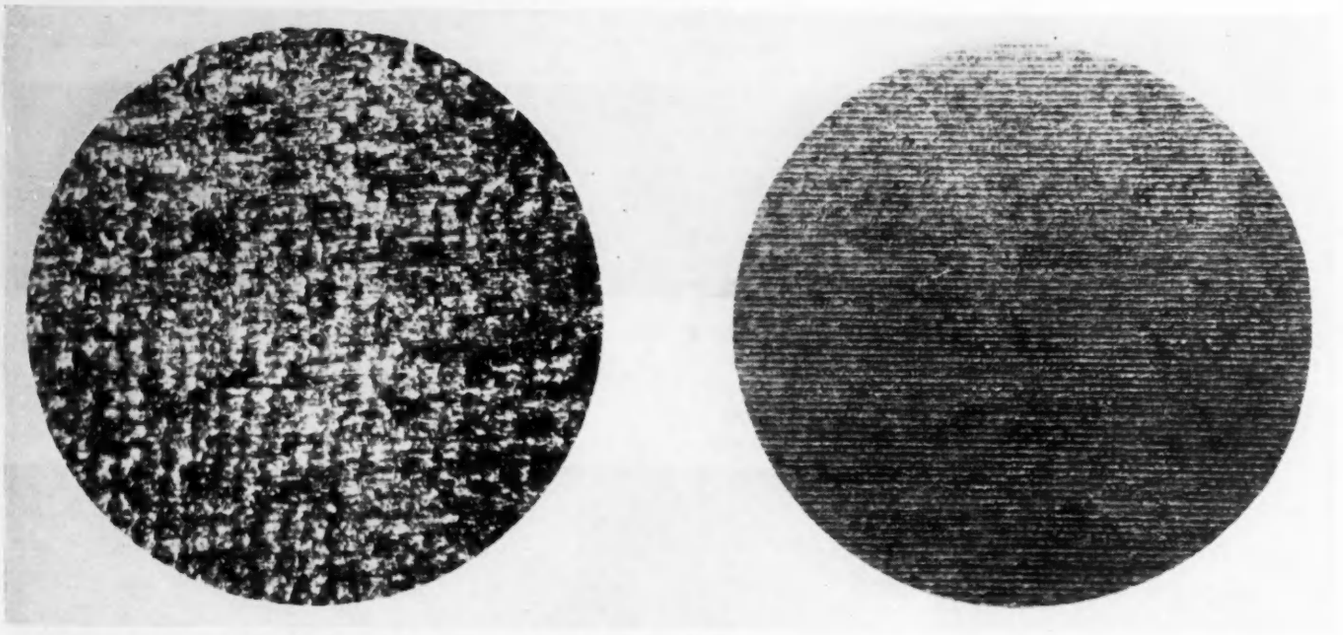
Fig. 3 – Surface Roughness after Machining Operations in Cylinder Block



A – Used 1 Qt. in 300 Miles – Average Roughness 13 Micro-In.

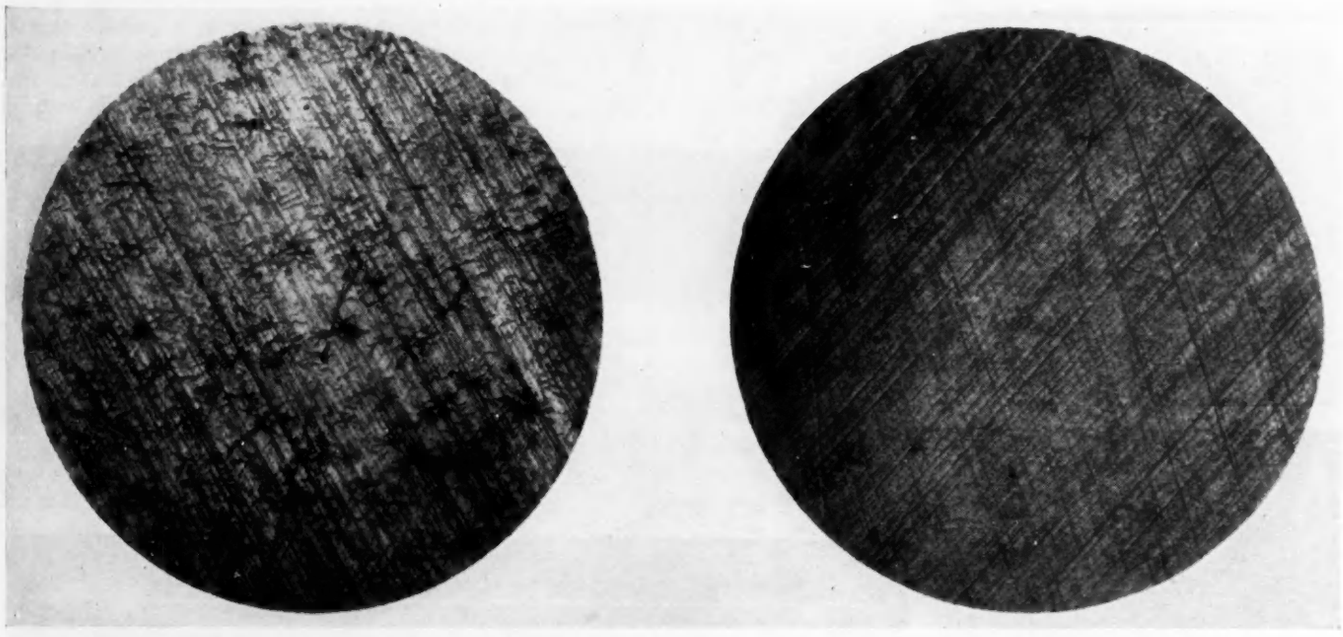
B – Used 1 Qt. in 1500 Miles – Average Roughness 9 Micro-In.

Fig. 4 – Surface Roughness Related to Oil Consumption



A – Reamed

C – Precision-Bored



B – Reamed, Mechanically Rough- and Finish-Honed

D – Precision-Bored, Hydraulically Finish-Honed

Fig. 5 – Comparative Structural Disturbance Resulting from Typical Machining Processes

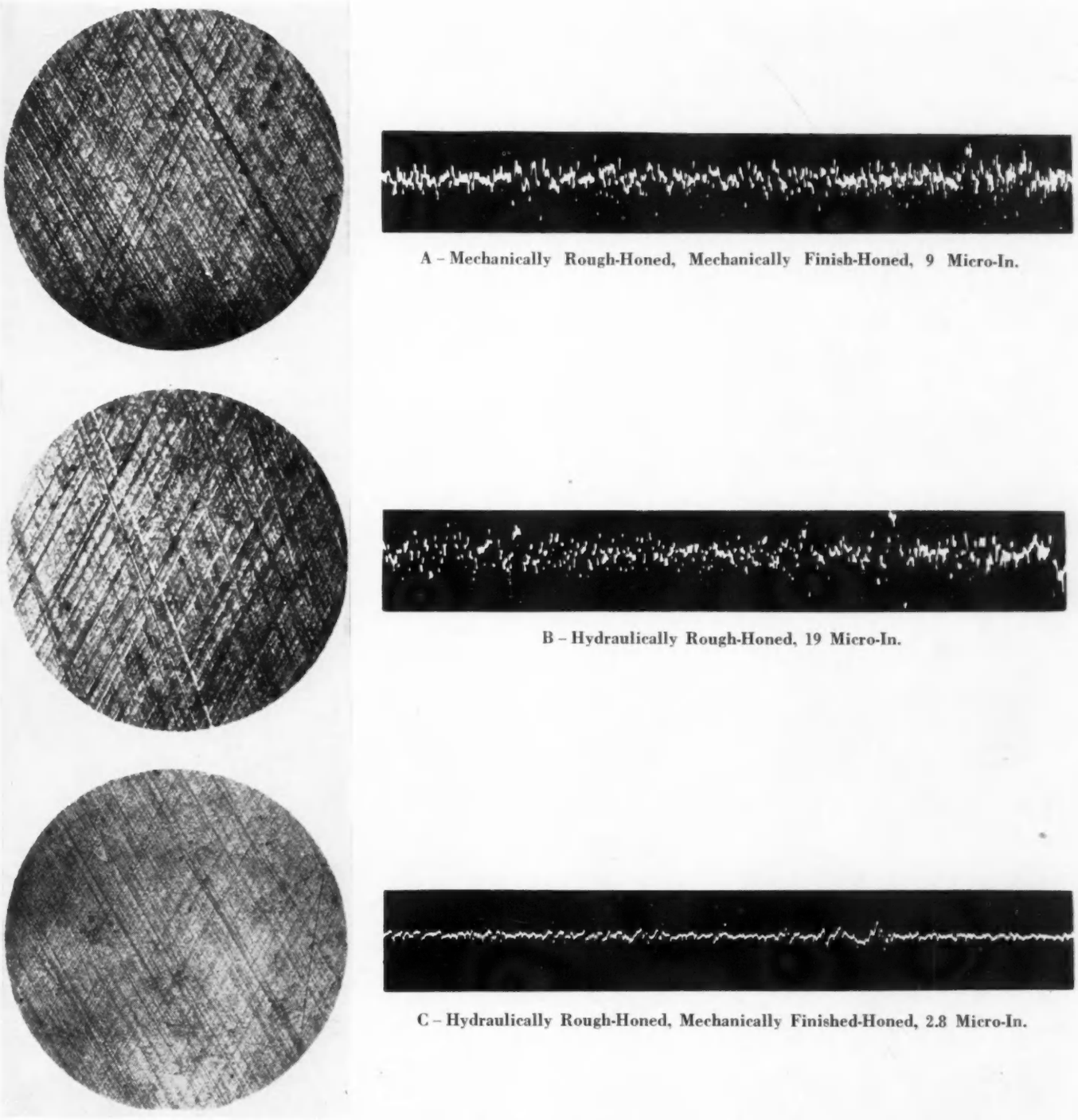
and speeds required to actuate the tools. Percussion and bearing forces definitely account for tearing and burnishing of the surface metal in various uncontrollable forms.

Evidence substantiating this viewpoint is shown in Fig. 5, View *B*, wherein it is evidenced that the tearing and burnishing action of a reaming operation is still present following rough- and finish-honing operations where mechanically actuated honing tools were used. In contrast to this structure is shown a precision-bored surface, View *C*, containing no apparent evidence of structural disturbance, and a similar surface, finished with hydraulically actuated honing tools, as in View *D*, representing the superior surface character and finish produced by this combined equipment and processing.

Tool Actuation

It is interesting to compare the results obtained by mechanically and hydraulically actuated honing tools as shown in Fig. 6. These profilograms and photomicrographs represent actual results obtained from blocks taken from regular production and include some taken prior to the development of hydraulically actuated tools.

It has been determined that, with the old mechanically actuated honing tools, pressure was applied in a reverse cycle to that required for efficient, heavy stock removal. In other words, due to mechanical limitations in the tool, a high initial pressure was followed by a decreasing pressure as the hone approached base metal. These views show that it was not



A – Mechanically Rough-Honed, Mechanically Finish-Honed, 9 Micro-In.

B – Hydraulically Rough-Honed, 19 Micro-In.

C – Hydraulically Rough-Honed, Mechanically Finished-Honed, 2.8 Micro-In.

Fig. 6 – Comparative Surface Finish Obtained by Mechanically and Hydraulically Actuated Honing Tools

practically possible to cut down below the structural deformation caused by previous operations, and demonstrate that the best final surface finish, which could be obtained with this type of actuation, represent a profilometer reading of 9 micro-in.

Hydraulically actuated honing tools are designed to permit application of variable pressure during the honing cycle, with the pressure increasing as the total contact area between the abrasive members and the work metal increases. This action permits the tools to cut away large amounts of stock, substantially below any structural deformation. Slightly coarser stones of much softer grade may be used to accomplish more practical and efficient stock removal in the rough-honing operation than could be used with mechanically actuated tools.

Profilogram *B* illustrates the typical surface character of metal rough-honed with hydraulically actuated tools, showing a profilometer reading of surface roughness of 19 micro-in. It is interesting to note that a casual inspection of these two specimens, observed by the profilograms and photomicrographs, might lead to a conclusion that the finish of 9 micro-in., shown in Profilogram *A*, is not apparently smoother than the finish of 19 micro-in. obtained with hydraulically actuated tools in a roughing operation, as shown in Profilogram *B*.

An interesting honing application on reamed bores is illustrated in Profilogram *C*, comprising hydraulically actuated tools in the roughing operation followed by mechanically actuated tools in the finish-honing operation. It is evident that this new tooling has permitted the removal of sufficient

stock to remove practically all structural disturbances while accomplishing a final surface finish of 2.8 micro-in.

Processing Variations

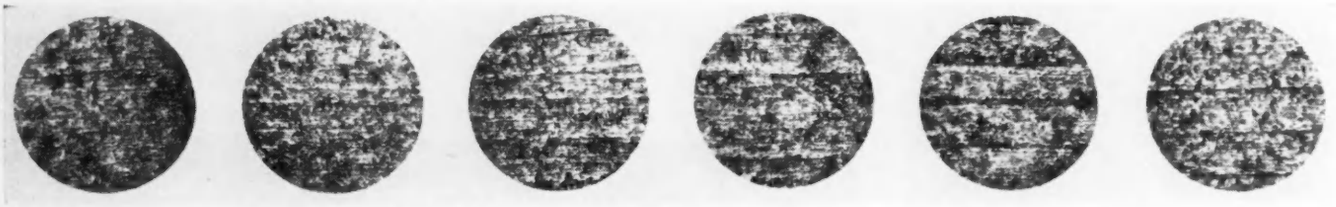
Current production processing is concerned with the amount of variation which occurs in some processing operations on multiple-cylinder in-line applications. Evidence of this variation is illustrated in Fig. 7, comprising examples of various processing taken from regular production blocks.

The average condition of tearing, which occurs in rough boring, is illustrated in Fig. 7-*A;* and Fig. 7-*B* illustrates the variations of tearing, burnishing, chatter, and similar injurious results of a reaming operation.

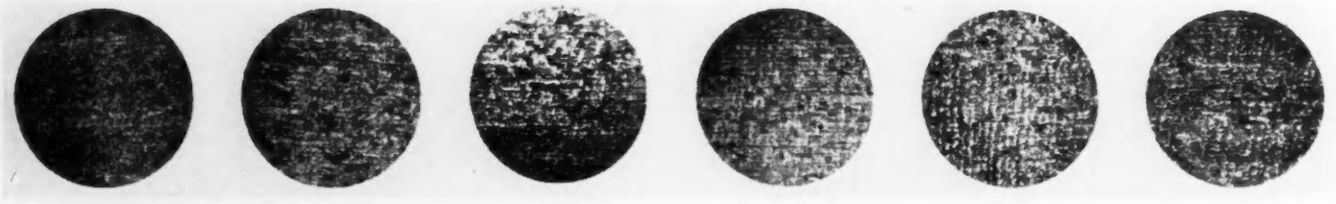
In Fig. 7-*C,* these reamed bores have been rough- and finish-honed with mechanically actuated tools, removing from 0.003 to 0.0035 in. of stock on the diameter, and again it is evident that the metal still retains the structural disturbances resulting from previous processing. The manufacturer producing this six-cylinder block recently adopted hydraulically actuated honing tools for the rough-honing operation following reaming, with the results shown in Fig. 7-*D.* The bores are then finished with mechanically actuated honing tools as shown in Fig. 7-*E.* A total of 0.004 to 0.0045 in. of stock is removed following the reaming operation, accomplishing accuracy averaging from 0.0002 to 0.0003 in. for roundness and straightness, and holding final finish uniform within 3 to 4 micro-in. in regular production.

Simplified Processing

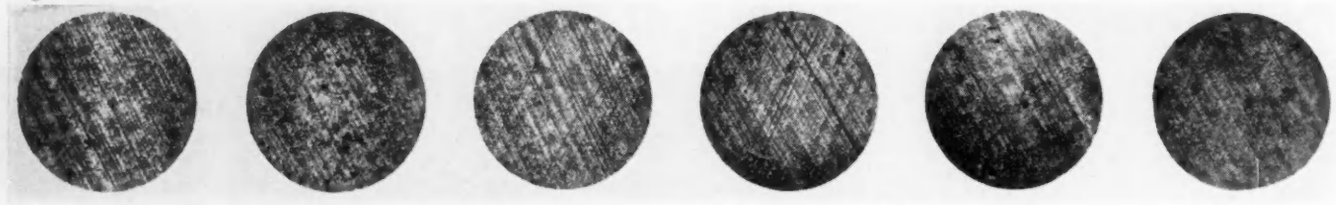
A typical, single-spindle all-hydraulic honing machine is shown in Fig. 8. An important application of this equipment has been adopted recently by a manufacturer of cast-steel-



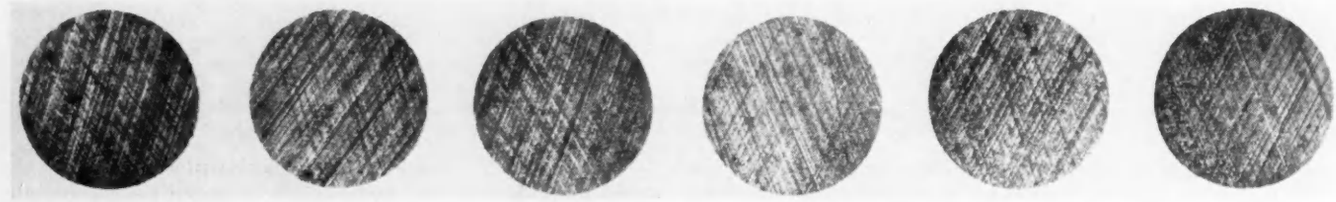
A – Rough-Bored



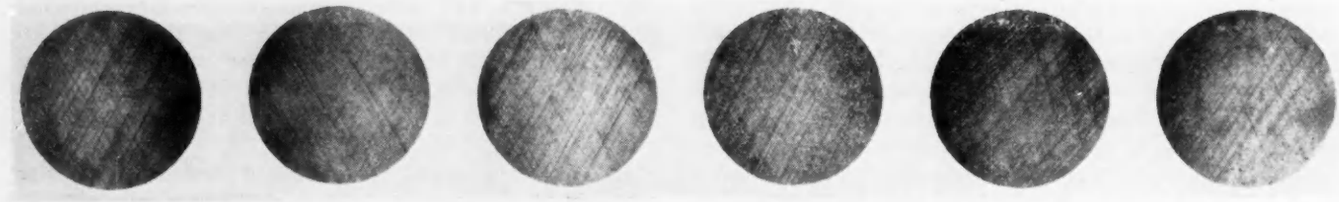
B – Reamed



C – Rough and Finish Mechanically Honed



D – Hydraulically Rough-Honed



E – Hydraulically Rough-Honed, Mechanically Finish-Honed

Fig. 7 – Photomicrographs of Surface-Roughness Characteristics Produced in Machining Six-Cylinder-In-Line Blocks

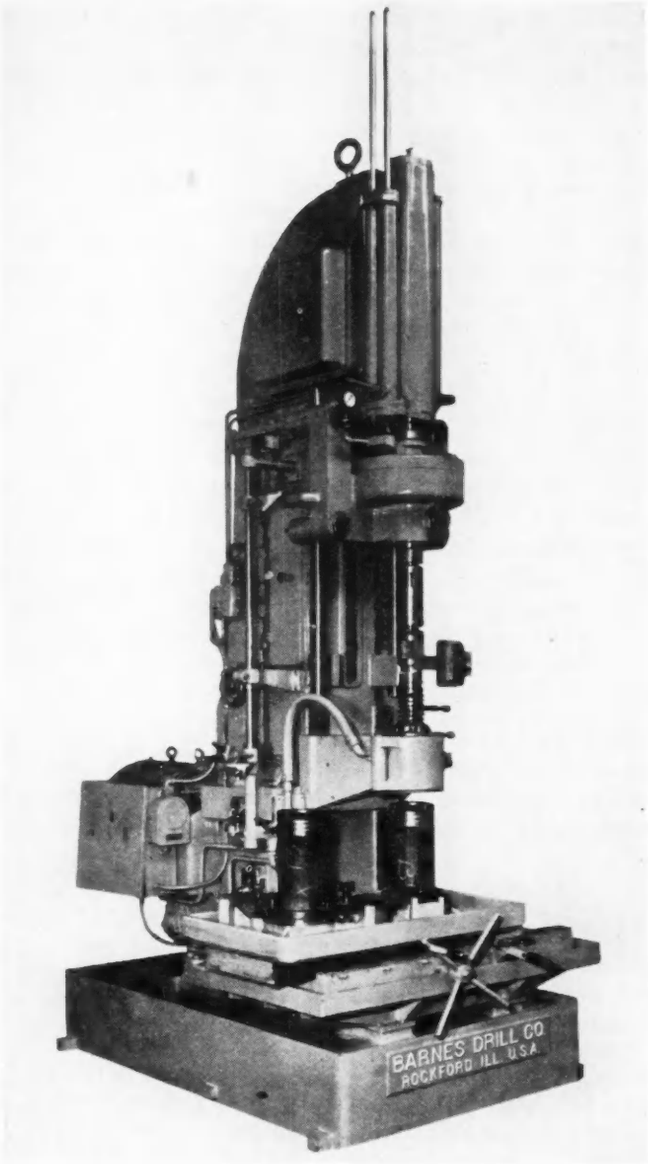


Fig. 8 – Single-Spindle All-Hydraulic Honing Machine

alloy tractor sleeves, 5¾ in. in diameter by 15 in. long, of hardness 45-50 Rockwell C scale, comprising an important accomplishment in simplified processing.

In former practice, these sleeves were processed as follows: rough-bored, semi-finish bored, hardened, ground to correct distortion resulting from hardening, rough-honed and finish-honed – a total of *five* operations.

After adopting hydraulically actuated honing tools, complete processing of these sleeves was reduced to *three* operations, eliminating semi-finish boring and the costly grinding operation, as follows: rough-bore, harden, rough-hone, finish-hone. Approximately ¼ in. of stock on the diameter is removed in the rough-boring operation. The sleeves are then heat-treated to 45-50 Rockwell C. Following the hardening, hydraulic rough-honing removes an average of 0.027 in. of stock on the diameter, at the rate of approximately 0.012 in. per min. This operation corrects all distortion produced in hardening, including from 0.010 to 0.015 in. out-of-roundness. Hydraulic finish-honing removes approximately 0.001 in. of stock on the diameter, within accuracy of 0.0005 in. for roundness, and from 0.0007 to 0.001 in. for straightness. Total production is maintained at the rate of approximately 20 sleeves per hr., and 73 sleeves per set of honing stones.

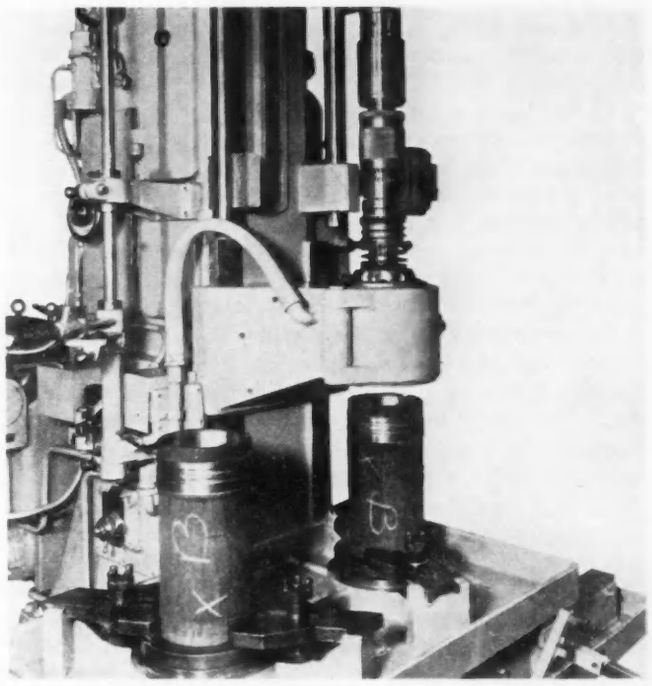


Fig. 9 – Hydraulic Honing-Tool Installation

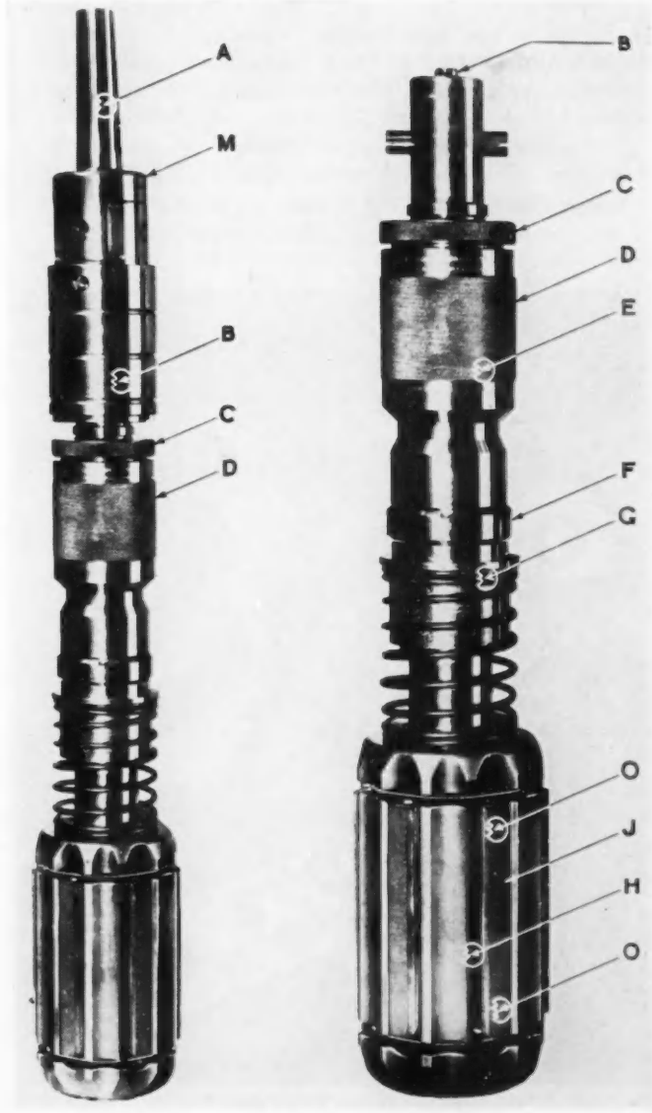


Fig. 10 – Hydraulic Honing Tool

As shown in Fig. 9, the equipment comprises a two-stage type fixture, permitting the operator to unload a sleeve while one is being honed. The honing cycle is arranged as follows:

The hone is lowered into the work in a collapsed position and the rotation and reciprocation of the spindle started. At the bottom of the first stroke, the hone automatically expands to a light pressure and is run for approximately 25 strokes to remove stock to base metal before the normal working pressure cuts in and remains until the electrical timing device, set at a predetermined time cycle, collapses the tool and releases the hydraulic circuit pressure, causing the hone to be withdrawn automatically from the work.

The tool used in this application is shown in Fig. 10. Hydraulic pressure actuation, provided in an individual hydraulic cylinder in the head of the machine, actuates push rod *B* which, in turn, actuates the tool expansion mechanism within the limits of selective adjustment provided in the head assembly.

From a recent survey it is evident that quite a few manufacturers of internal-combustion engines are still using five or six operations in processing engine cylinder bores from the rough. With the new stock removal accomplishment by the abrading method, as comprised in hydraulic honing, it is now possible not only to reduce the number of operations to as few as three, but also to obtain any desired degree of final surface finish on an economic operating basis.

The continued development and adoption of improved casting technique, and the projected adoption of electric induction hardening for cast-iron engine-block production, will assuredly demand this combined accomplishment which is obtained most advantageously by hydraulic honing.

Conclusion

In conclusion, it is at last possible to compare investigations of wear with a standardized system of measurement. It is no longer necessary to cut open a block, or part, to measure surface finish. The portable Profilometer can be used quickly and conveniently in the dynamometer laboratory as shown in Fig. 11, or at the proving ground to study wear in test engines under all conditions of operation. It is now being used, when desired, at the production line to check and maintain surface finish in hourly or daily production.

Judging from present indications of its usage as just outlined, it may be predicted safely that the Profilometer, combined with photographic or microscopic study of surface character, may expand the scope of present investigating technique and reveal more definite, correlated data for future progress in reducing engine wear.

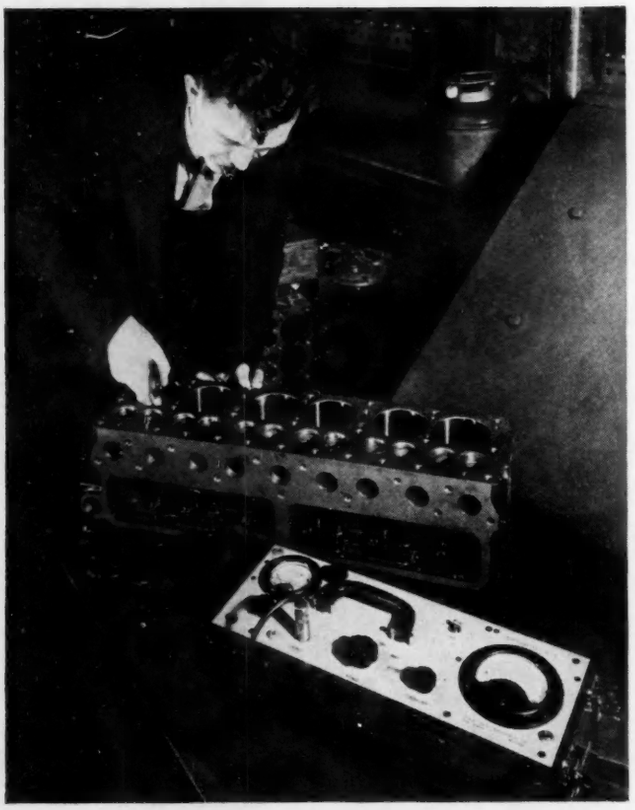


Fig. 11 – Portable Profilometer in Use in a Dynamometer Laboratory

Tomorrow's Lubricants

AN eminent authority on lubricating oils and lubrication has stated that the lubricant of tomorrow will be an "assembled oil," one made up of specialized components each of which possesses one or more properties extremely important in the make-up of a lubricant for any given purpose. Although that point has not been reached as yet, the requirements for and the manufacture of crankcase lubricants especially are trending so rapidly in that direction that discussion of what the ultimate lubricant will be is not a visionary wandering of the academic mind, but a decidedly practical look-see at what the immediate future probably holds.

A review of the properties in lubricants which occupy most the minds of refiners and automotive men alike shows that the following characteristics are of most importance: viscosity and the viscosity change with temperature change, (V.I.); oxidation stability, and the deposition of sludges, and so on, in equipment; oiliness, or film strength, the ability to maintain a lubricating film under severe conditions; pour point, the pumpability of a lubricant at low temperatures. Other properties have some importance, but on these four "hang all the law and the prophets (profits)."

Doubtless agreement hardly could be reached as to which of these properties is of the most importance to the engine and the operator. However, since the primary and all-important job of a lubricant is to prevent metal-to-metal contact between bearing and journal under all conditions, film strength or oiliness is generally considered the controlling property. The viscosity index or stability to oxidation and heat decomposition characteristics are of little use if the oil will not stay between the rubbing surfaces, reduce friction, and prevent scoring and seizure.

So far as can be foreseen now, petroleum hydrocarbons will and must be the source material for the major portion of our future motor lubricants, as at present. Our habitual loose method of verbal expression is to state that paraffinic hydrocarbons compose the best lubricating oils. That this is untrue is indicated by the fact that paraffin wax, the "type molecule" of the paraffins, is not a lubricant in any real sense. It is necessary to remove most of the wax found in paraffinic lubricant fractions, not especially because of the non-lubricating properties of the wax – which will lubricate in an indifferent manner when fluid – but because of the effect of wax on the pour point and flow characteristics of the finished product.

Excerpt from the paper: "Characteristics of Tomorrow's Lubricants; Sources and Technology of Super-Lubricants," by Arch L. Foster, National Petroleum News, presented at the Kansas City Section Meeting of the Society, Kansas City, Mo., Mar. 4, 1938.

Ride Controls and Calibration

By C. R. Paton

Chief Engineer, Packard Motor Car Co.

DISCUSSION is limited in this paper to ride problems pertinent to cars equipped with independent-type front and non-independent rear suspensions. Emphasis is placed on problems arising in connection with the newer so-called "flat-type" rides. Approach is made to the problem of the control or damping of springs and some of the factors, important from a balance standpoint, in securing these flat pitch-free ride characteristics.

It is pointed out that shock absorbers should more properly be called "ride controls," so great is the importance of proper damping characteristics in obtaining the newer flat-type drives. The tremendous number of interrelating factors involved in the attainment of ride excellence is given as the reason why effective instrumentation has never been developed. The author concludes that the maze of compromises involved will always require the experienced observer.

THE problem of ride control and calibration is such a broad subject that we must confine ourselves to some of the newer problems which have arisen in connection with present-day rides. For this reason we will, in general, limit our discussion to ride problems pertinent to cars equipped with independent-type front and non-independent rear suspensions. We particularly wish to discuss problems arising in connection with the newer so-called "flat-type" rides.

A paper presented at the same meeting[1] has dealt capably with the subject, primarily, of springs. We here approach the problem of the control or damping of springs and some of the factors, important from a balance standpoint, in securing these flat, pitch-free ride characteristics.

The problem of ride development, which only a few years ago was considered a relatively simple and relatively unimportant detail of car design, has now developed into one of the most important of the considerations in the design of our present-day cars. Springs, which not too many years ago were selected without much thought as to their frictional or damping characteristics, either have had this damping entirely removed by the adoption of the coil type or have undergone refinement of detail to insure predetermined and permanent frictional characteristics. Some of the more recent leaf-spring designs actually are calibrated to provide specific control characteristics.

A few years ago, shock absorbers of almost any type that would tie down or snub the springs successfully were all that were considered necessary. Now, shock absorbers have additional responsibilities and should more properly be called "ride controls" – so great is the importance of proper shock-absorber damping characteristics to the overall ride result obtained. Ten years ago, ride was considered to be a problem involving only springs and shock absorbers. Today, ride has assumed major importance in the entire design of both chassis and body, and ride "calibration" has become a compromise between so many opposing factors as to have actually defied complete solution.

If we were to consider only the problem of longitudinal balance, I believe many would agree that the ideal ride might be described as one in which the sprung masses, including the body and passengers, would travel substantially flat, be free from pitching, be displaced vertically only as required to average the major variations in the normal road contour, and never rise or fall at so great a velocity as to impart unpleasant accelerations through the cushions to the passengers.

When the ride engineer starts to develop a ride, he usually is faced with certain definite limitations as regards weight distribution, spring rates, suspension type, and so on. After these factors are established by the designers, it becomes his duty to develop the best result that he can obtain within the limitations imposed by the restrictions of the design. The ride engineer usually is charged with the responsibility of establishing proper damping of the several elastic systems involved, including those represented by the tires, the suspension springs, the suspension linkage, the steering linkage, the frame and body structure, and so on. This damping usually is obtained by a combination of static and dynamic friction obtainable in the suspension springs, the spring mountings, the steering linkage, the bearings in the front suspension linkage, as well as by means of the valving which determines the damping provided by the hydraulic shock absorbers.

The establishment of the proper frictional characteristics in all of these chassis parts, as well as the determination of the proper shock-absorber-valve calibration, properly becomes the responsibility of the ride engineer today. We should emphasize that the shock absorbers themselves are only one factor of many in the ride-development problem and that the frictional values obtainable in other chassis parts, such as those just mentioned, are a very important factor in the development of the ride with definite limitations being placed upon the ride engineer once the type of spring, spring mounting, or suspension design is determined.

Frictional Characteristics of 1938 Cars

Even though we confine our consideration of rides to American-built cars equipped with independent-front and

[This paper was presented at the National Passenger Car Meeting of the Society, Detroit, Mich., Mar. 29, 1938.]

[1] See "Recent Developments in the Design of Passenger-Car-Vehicle Suspension Springs," by Tore Franzen, presented at the National Passenger Car Meeting of the Society, Detroit, Mich., Mar. 29, 1938.

FRICTION CLASSIFICATION	FRONT SUSPENSION		REAR SUSPENSION	
	INDEPENDENT TYPE		NON-INDEPENDENT TYPE	
	SPRING TYPE AND SUSP. BEARINGS	FRICTION LEVEL	SPRING TYPE	FRICTION LEVEL
A	CONV. LEAF THREADED AND ANTI-FRICTION BEARINGS	HIGH	CONVENTIONAL LEAF	HIGH
B	COIL SPRING THREADED BEARINGS	MEDIUM	CONVENTIONAL LEAF	HIGH
C	COIL SPRING THREADED BEARINGS	MEDIUM	SPECIAL LEAF WITH FRICTION LINERS	MEDIUM
D	COIL SPRING THREADED BEARINGS	MEDIUM	COIL SPRING	LOW
E	COIL SPRING ANTI-FRICTION BEARINGS	LOW	CONVENTIONAL LEAF	HIGH
F	COIL SPRING ANTI-FRICTION BEARINGS	LOW	SPECIAL LEAF WITH TIP INSERTS	LOW

Fig. 1 – Frictional Characteristics of 1938 American Cars

non-independent-rear suspensions, we find that there is relatively little agreement among designers as to either the friction level desirable or the ratio of friction between front and rear suspensions.

Fig. 1 presents a tabular summary of significant design specifications affecting the friction level for representative 1938 cars, dividing them into six classifications based on the magnitude of the friction in the suspension. It may be noted that there is a major variation in the friction level ranging through high, medium, and low for both front and rear suspensions.

The general character of the optimum ride which can be developed has an inherent limitation determined by the frictional characteristics in the front and rear suspensions. It is interesting to note that the friction level is determined not alone by the type of spring employed but also by the type of spring mounting and the type of bearings used in the suspension linkage. The type of bearing permissible also is related very closely to the type of suspension employed.

Assuming, as we may, that satisfactory high-speed control can be obtained without resorting to high friction in the suspension linkage or springs, it may be shown that the highest degree of flatness and the greatest freedom from pitching tendency may be obtained when the friction level at both ends of the vehicle is held to a low value.

The progress that has been made in ride improvement during the last few years first received impetus with the development of the coil-spring independent front suspension. We submit that the major improvement derived with the coil-spring independent front suspension was not alone due to the major reduction in spring rates but that, in large measure, it also was due to the reduction in the friction level. This same observation also is supported by experience with leaf springs. Distinct benefit to ride balance is derived from reduction of the friction level as spring rates are reduced.

Where a low level of friction is provided in either the front or rear suspension, supplementary means, such as rubber bumpers of relatively large capacity, must be provided. It is also necessary, in order to develop the full advantage of the low friction values in the suspension, to provide some means of encouraging flatness and freedom from pitch tendency. However, assuming suitable bumper equipment and flatness-control features, it has been our experience that the inherent tendency for flatness is greater on all types of road surface with a low friction level and that less control is required.

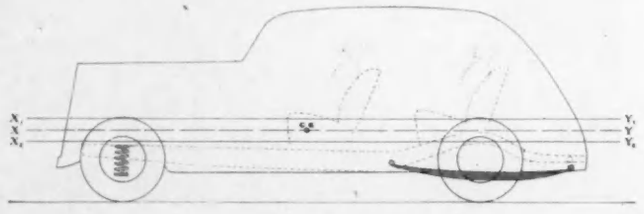


Fig. 2 – Condition for Flat Ride with Oscillation Center an Infinite Distance from Center of Gravity

Primary Factors Affecting Balance

Several of the primary factors affecting the normal ride balance should be mentioned. One of the most important is now known to be the matter of the weight distribution of the sprung masses with respect to wheelbase. This subject has been treated amply elsewhere and is now fairly well understood. However, many practical design limitations prevent obtaining the ideal weight distribution.

Another factor of primary importance to ride balance is the proper selection of spring rates which, in turn, establishes the value of spring static deflection. Proper weight distribution and static deflection determine the ride frequencies and establish a ratio of bounce to pitch frequencies which will tend to leave bounce predominantly in control.

Of importance to the matter of balance from the viewpoint of control are the values of static and kinetic friction provided by leaf springs, spring mountings, independent suspension bearings, and so on, previously mentioned, and also the character of the *kinetic* damping provided by the hydraulic shock absorbers. All of these several control functions must be harmonized to obtain the proper balance and degree of riding ease.

Possibly the factor of greatest importance to the ride engineer in obtaining a flat or "balanced" ride is the attainment

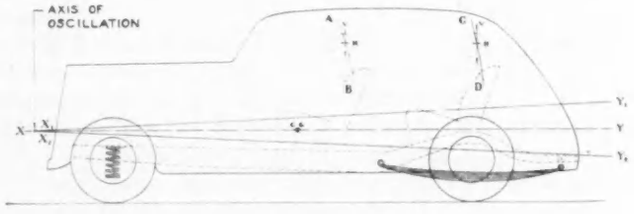


Fig. 3 – Condition with Oscillation Center a Short Distance Forward of the Vehicle

of a balance of forces to insure the most satisfactory ride oscillation center for all conditions. Those who are familiar with this problem will appreciate that, for most cars, there is an inherent tendency for the sprung masses to oscillate in the longitudinal plane or to develop pitching under many common road conditions *unless* effective means are provided to restrain this type of movement. Theoretically, it is possible to obtain a weight distribution and spring rates such that the oscillation center will fall an infinite distance from the center of gravity. In such case, the sprung mass will rise and fall parallel to the ground without any pitch component.

This condition, often called a "flat ride," is represented by Fig. 2 in which the center of gravity is shown indicated by letters C. G. Line X-Y represents an imaginary horizontal plane drawn through the center of gravity. X_1-Y_1 and X_2-Y_2 represent the positions assumed by the X-Y plane during maximum rebound or compression movement for any given condition.

In actual practice, this condition is not difficult to obtain

for certain limited operating conditions. For example, if front and rear frequencies are substantially equal and weight distribution is reasonably satisfactory, this type of ride is often obtained at fairly high road speeds. However, it is quite difficult in practice to maintain these conditions at low speeds on road surfaces which have a pronounced tendency to cause pitching. Further, with variation in the number of passengers carried, the equality of front and rear frequencies is destroyed. Our rides, of course, must be satisfactory under varying conditions of passenger load, speed, and on all kinds of road surfaces.

The actual center of oscillation is found frequently either within or somewhere near the wheelbase and those familiar with ride calibration problems are familiar with such cases as represented by Figs. 3 and 4. Fig. 3 shows the condition where the center of oscillation is a short distance forward of the vehicle. In this instance planes X-Y, X_1-Y_1 and X_2-Y_2

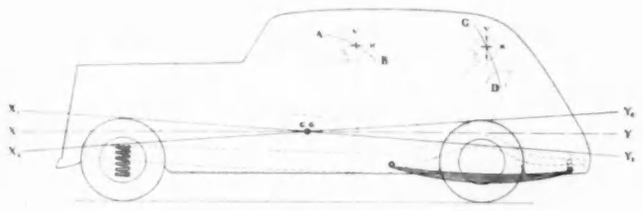


Fig. 4 – Condition with Oscillation Center within the Wheelbase

intersect at a common point which becomes the axis of oscillation. In the past, this has been a relatively common condition and one not too unpleasant to passengers since it involves principally vertical movement for both front and rear seat occupants, as represented by arcs *A-B* and *C-D*. The principal objection to this oscillation center is the feeling had by both front-seat and rear-seat passengers that the rear end of the vehicle is doing too much swinging and lacks control. It will be noted that the neck-whip tendency of fore and aft component of arcs *A-B* and *C-D* are relatively small for both front and rear passengers.

Another, more-common condition is represented by Fig. 4 in which the center of oscillation falls within the wheelbase. This is the condition that gives rise to most of our trouble with pitching, neck-whip, and ride balance. Unless otherwise restrained, there is always a decided tendency for the center of oscillation to fall near the center of gravity. This particular type of movement is quite objectionable to both front-seat and rear-seat passengers. The front-seat passenger is subjected to an objectionable neck-whip with relatively little vertical movement, whereas the rear-seat passenger is subject to both objectionable neck-whip and vertical tossing. This is often the condition encountered at relatively low speeds on many common road surfaces with ride classifications *A* and *D*, Fig. 1 and, to a lesser degree, with ride classifications *B*, *C* and *E*. It is our observation that this type of movement can be held to a minimum with ride classification *F* having a low friction value at both front and rear.

Unless we are to obtain the center of oscillation represented by Fig. 2, that is one located at infinity, a consideration of the simple geometry of the problem may be made to reveal that the next most desirable condition is to have the center of oscillation located to the rear of the vehicle.

The condition shown in Fig. 5 is very similar to that of Fig. 3 except that it is reversed. Both the front-seat and rear-seat passengers are subjected to vertical movement of relatively small magnitude and are practically free from neck-whip. This condition is desirable as compared with that represented by Fig. 3, principally because it is possible to reduce the rear-seat amplitude to a value less than that encountered by the front-seat passengers.

Ride engineers will appreciate that the problem of obtaining satisfactory ride "balance" is concerned to a major extent with the problem of locating the axis or so-called "center" of oscillation at such a point as to obtain the greatest degree of comfort for both the front and rear passengers. It is probably unnecessary to mention the desirability of keeping this center of oscillation, which so largely determines ride balance, as definitely located as possible for all ride conditions. In our opinion, all of the factors contributing to static friction, dynamic friction, and fluid damping which go to make up the ride control should be adjusted individually so that the controls at the rear of the vehicle will always be slightly dominant over those at the front. In our experience, only by insuring this condition for each distinct component of control, can the maximum flatness and freedom from pitching be obtained. If, for example, we take the case of a car having a high friction value in the front suspension, it may be seen that, unless a correspondingly high level of friction is maintained in the rear suspension, the front end will tend to dominate under many road conditions and will defeat the possibility of obtaining the balance condition represented by Fig. 5.

Many control functions, of course, combine to determine the matter of front versus rear dominance, but we believe that it may be seen readily that, when all ride conditions are considered, rear dominance over front has many inherent advantages. Fig. 4 shows the movement to which passengers are subjected when the front end dominates the rear. Under this condition the *front end* of the vehicle tends to follow the road surface, resulting in vertical acceleration at the front. Oscillation of the sprung mass tends to occur about the center of gravity, resulting in vertical pitching of rear-seat passengers and neck-whip of front-seat passengers.

Fig. 5 shows the condition for passengers in the case of rear-end dominance. In our opinion, it is possible to obtain a more agreeable all-around ride result by adjusting all control functions so that rear suspension and controls always will be dominant, than is possible in the case of either front-end dominance or where control changes from front to rear under different road conditions. Most of us are familiar with rides which, on some road surfaces, appear to be quite flat and satisfactory, particularly at the higher speeds, but which develop a pronounced pitching tendency, particularly at low speeds, on certain roads. This tendency can be minimized by calibrating the ride-control functions to insure rear dominance over front. With rear dominance vertical movement as well as neck-whip is held to a minimum and a pleasing ride "balance" is obtained.

If we may agree to the desirability of the ride "balance" represented by Fig. 5, we may then discuss some of the control factors that are available to us for the maintenance of this condition. Fortunately, lack of the theoretically correct

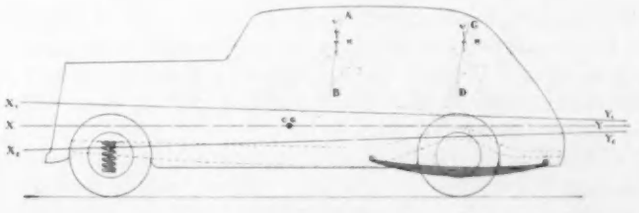


Fig. 5 – Condition with Oscillation Center to the Rear of the Vehicle

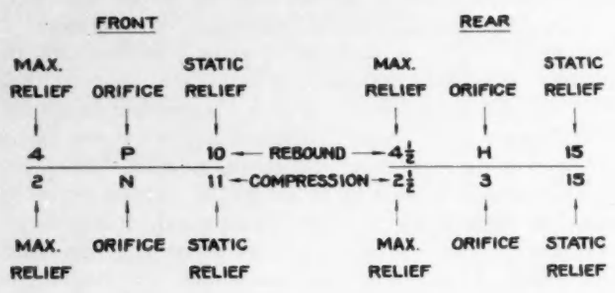


Fig. 6 – Typical Calibration Diagram – Hydraulic Shock Absorber

weight distribution and spring rates can, in some measure, be replaced by proper handling of the available control factors. To accomplish this purpose the hydraulic damping must combine with the friction in the springs and mountings to insure the desired center of oscillation.

Balance with Hydraulic Controls

For this discussion we will avoid reference to specific types of hydraulic shock absorbers and, rather, will confine ourselves to typical controls which many *do*, and all types of hydraulic instruments which we *may*, employ. To avoid complicating discussion, we omit reference to inertia-type controls, although we acknowledge their potential value in securing flatness.

The twelve commonly used shock-absorber variables are presented diagrammatically in Fig. 6. In order to insure a flat, pitch-free ride, the six front variables usually must be balanced against the six rear variables so as to insure that both ends of the car will rise and fall with the same velocities. Fig. 7 is a chart of the control variables which most commonly have to be so balanced. This chart presents a picture of the hydraulic control functions which should be adjusted rather accurately relative to one another before we may hope to obtain any substantial degree of ride flatness.

It is, of course, unnecessary to attempt any detailed explanation of how these several control functions should be balanced. Ride engineers have acquired a familiarity with valving through actual experience in handling these variables. We should only like to point out, however, that where possible, other compromises permitting, we consider it desirable to have rear maximum relief always slightly greater than the front, and rear orifice always more effective than the front. This general suggestion applies with equal importance to both the compression and rebound sides of the control.

Leaf Springs, Mountings, Bearings

Earlier in our discussion we have mentioned the importance of the static and kinetic frictional characteristics of leaf springs, spring shackles, and suspension bearings to the overall ride result, particularly as it affects the optimum ride balance and flatness.

The preceding paper[1] has treated very ably these chassis units and we, therefore, refrain from duplication of coverage so far as possible. It, however, appears necessary, by way of preparing a foundation for our subsequent discussion of the control characteristics of these units, to present here a few significant facts as follows:

1. Within the limits of the applications here discussed, coil-type suspension springs have neither static nor kinetic friction.

2. Rubber, ball and roller bearings, when used in front suspension linkages or for leaf-spring mountings may, as now usually applied and at the ride frequencies involved, be all considered to have neither static nor kinetic friction.

3. Minor values of static friction and major values of kinetic friction occur in threaded-type bearings commonly used for spring mountings and suspension linkage bearings. These values are subject to substantial change with varying conditions of lubrication, wear, and cleanliness.

4. Major values of both static and kinetic friction occur in plain-type bearings when used in suspensions. Values also are subject to change with lubrication, wear, and cleanliness, the same as with threaded bearings.

5. Conventional-type leaf springs have relatively high values of both static and kinetic friction which, together with the values included under Items Nos. 3 and 4, constitute important factors in determining the control, balance, and character of the ride.

6. Control disadvantages of conventional leaf springs include:

A. Static and kinetic friction varies in production due to normal manufacturing variations and differences in surface condition and lubrication. Also, it varies over wide limits in service.

B. The necessity for lubrication results in wide variations in friction dependent on road conditions and temperatures.

C. Inability to control closely the ratio of static to kinetic friction.

D. Frictional values are not readily subject to adjustment as is desired frequently to permit balancing with the frictional values in other suspension parts.

E. Inability to establish separately the values of vertical and transverse friction.

The control limitations of conventional-type leaf springs, as indicated by the foregoing Items Nos. 5 and 6, are rather well known, and numerous attempts have been made to improve their characteristics. These modifications of the conventional leaf spring have been covered by the preceding paper[1]. Several of the most important of these modifications which have been used, should be mentioned again, including the wax-treated fibrous-board liners used between certain leaves by Cadillac and LaSalle; the self-lubricating bronze inserts used by Chrysler; and the low-friction molded-composition inserts used by Nash. A new spring used by Packard has certain important features which should be included in this paper because of the control obtainable with it. This spring can be used to provide quite definite control characteristics – to be more specific, it can be given a definite frictional calibration.

The spring to which we refer has been described in the preceding paper[1] and makes use of a combination of tip inserts of rubber, metallic, and other non-metallic materials. Fig. 8 shows a typical insert location diagrammatically. Note

1 – MAX. RELIEF-FRONT-REBOUND vs MAX. RELIEF-REAR-REBOUND

2 – ORIFICE – FRONT-REBOUND vs ORIFICE – REAR-REBOUND

3 – STATIC RELIEF-FRONT-REBOUND vs STATIC RELIEF-REAR-REBOUND (IF USED, OFTEN OMITTED)

4 – MAX. RELIEF-FRONT-COMP. vs MAX RELIEF-REAR-COMP.

5 – ORIFICE – FRONT-COMP. vs ORIFICE – REAR-COMP.

6 – STATIC RELIEF FRONT-COMP. vs STATIC RELIEF-REAR-COMP.

Fig. 7 – Control Functions – Hydraulic Shock Absorber

the different bearing materials used to separate the leaves at the tips.

Fig. 9 presents a table showing three typical materials which have been used in combination and the particular characteristics of these materials. Note in this table we indicate three characteristics named *X*, *Y*, and *Z* for convenience in discussion. *X* represents the static friction or break-away value. *Y* and *Z* represent two different "varieties" of kinetic friction. Both *Y* and *Z* provide low-frequency damping, whereas *Y* provides only a low value of high-frequency damping. As here used, low frequency is taken as applying to normal car sprung-mass frequencies in the general range from 0 up to approximately 100 cycles per min. High frequencies are taken to mean those of the approximate value of wheel bounce in the approximate range of 400 to 600 cycles per min. Tests to define the exact limits of these frequencies have not been conducted; however, road observations have shown the interesting characteristics covered by the table for frequencies in these two general ranges.

Note that the static friction or break-away value, *X*, can be adjusted readily with the lead-antimony bearings by changing the antimony percentage. This adjustment has been tested between the limits of 0 and 15 per cent. It has been used in production between 5 per cent and 10 per cent. Small changes in antimony percentage are detected readily on road tests, a difference of 2 per cent having been noticed by the riders who did not know that the percentage had been changed.

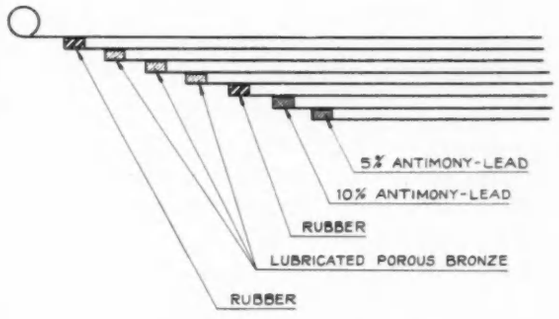


Fig. 8 – Spring Used To Provide Definite Control Characteristics Showing Different Bearing Materials Used To Separate the Leaves at the Tips

Lead-antimony inserts show nearly the same values when dry or lubricated with a light grease. They are used lubricated to insure against glazing of the surface in which latter condition they sometimes exhibit a slight break-away click. Cold flow at the specific loadings here employed is negligible, with approximately 5 per cent of antimony being just sufficient to conform readily to the rough forged spring tip depression.

As noted on the chart, the *Y* or high-frequency damping value of the lead-antimony alloy is low. This characteristic has a special value to be described later. The *Z* value or low-frequency damping value, however, is very high, being approximately double the value of lubricated bronze. The characteristics of the lubricated bronze and rubber are, of course, fairly well known so that no further explanation of the chart is needed. It may be pointed out, however, that a spring may be calibrated to secure a substantial range of variation of any one of the *X*, *Y*, or *Z* values without major change in either of the other two factors. Rubber has practically no *X*, *Y*, or *Z* value, as used, and is employed in the calibration to fill the remaining positions after the desired "calibration" of inserts and insert positions has been worked out. Stated another way, control calibration is accomplished by starting with all-rubber inserts and replacing certain of them with lead-antimony or bronze inserts, for example, as the conditions may require.

CHARACTERISTICS AND EFFECTIVENESS

TYPE OF BEARING	DESIGNATION	"X" STATIC OR BREAKAWAY FRICTION	"Y" HIGH FREQUENCY DAMPING	"Z" LOW FREQUENCY DAMPING	USES
ANTIMONY LEAD	AL	HIGH ADJUSTABLE BY ANTIMONY PERCENTAGE	LOW	HIGH EQUAL TO DOUBLE BRONZE	X Z
LUBRICATED BRONZE	B	LOW TO MEDIUM	MEDIUM	MEDIUM	XYZ
RUBBER	R	PRACTICALLY NONE	PRACTICALLY NONE	PRACTICALLY NONE	ZERO VALUE OF X Y & Z

CHARACTERISTIC "X" IS USED TO OBTAIN TIRE ABSORPTION OF MINOR ROAD IRREGULARITIES AND FOR CONTROL OF PITCHING.

CHARACTERISTIC "Y" IS USED TO CONTROL WHEEL BOUNCE.

CHARACTERISTIC "Z" IS USED TO CONTROL SLOW SWINGS AND PITCHING TENDENCY.

Fig. 9 – Characteristics and Effectiveness of Three Bearing Materials

Characteristic *X* may be used to obtain tire absorption of minor road irregularities without introducing harshness and also for the restraint of pitching tendency. This characteristic provides "rear support value" exactly equivalent on the road to reduced rear compression orifice. It may, therefore, be used interchangeably with rear compression orifice where it is desirable to modify either factor. The advantage of this privilege should be appreciated by the ride engineer.

Characteristic *Y* is used principally to establish the jiggle-jolt compromise and to supplement the hydraulic controls in suppressing wheel-bounce tendency. *Y* also is used to obtain the proper level of spring lateral damping.

Characteristic *Z* is used to control slow swings and also to restrain pitching tendency. By placing lead-antimony inserts at the tips of the short leaves, a substantial compensation in added control can be obtained as the passenger load is increased and short-leaf tip pressure is increased correspondingly.

In actual practice it has been found possible in some cases to obtain the needed high-frequency damping (*Y* value) and necessary wheel bounce control in the hydraulic shock absorbers and, in these cases, an insert specification entirely omitting the bronze is found satisfactory. Fig. 10 shows a diagram of such a specification.

Probably the most valuable and interesting characteristic of the lead-antimony inserts is the peculiar difference in damping ability at high and low velocities. Because this alloy, as used, provides high damping at low velocity and low damping at high velocity and also, because with it the static friction value can be adjusted so readily by changing the antimony percentage, it is very useful in maintaining a flat, pitch-free ride.

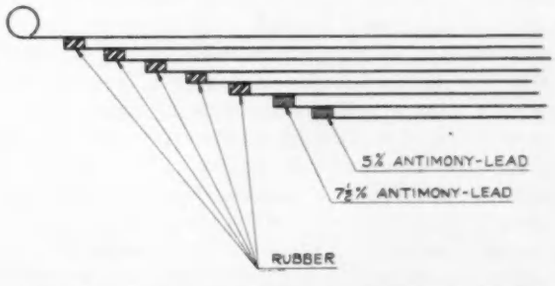


Fig. 10 – Insert Specification Omitting the Bronze Entirely

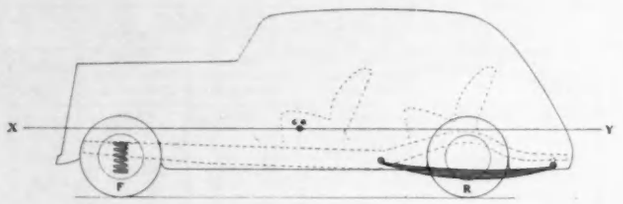


Fig. 11 – Conventional Car on a Smooth Road Surface

We will resort to diagrams to show how this ride may be accomplished. Fig. 11 represents a conventional car on a smooth road surface, line *X-Y* again representing an imaginary plane passing horizontally through the center of gravity.

This car either may be imagined as moving forward or, for mental convenience, as remaining stationary while the road surface passes through underneath it. If an obstruction

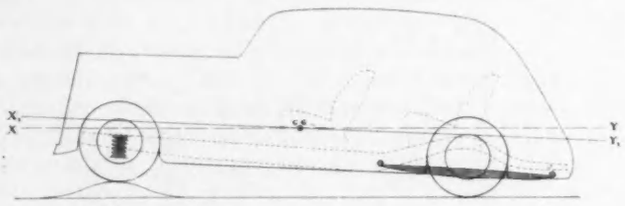


Fig. 12 – Conventional Car with Obstruction Passing under Front Wheels

passes under the front wheels (Fig. 12), the upward front reaction produces a moment about the center of gravity and tends to displace the *X-Y* plane to the new position X_1-Y_1. This moment, in turn, causes downward movement at the rear of the vehicle with resulting compression of the rear springs. As the obstruction passes under the rear wheels, the *X-Y* plane is displaced to the position X_2-Y_2 as indicated by Fig. 13. This type of pitching movement tends to persist and normally, of course, it is quite objectionable to passengers, resulting in tiresome "neck-whip" for front-seat passengers and tossing of rear-seat occupants.

If, now, by some mechanism we are able, when an obstruction passes under the front wheels, to support the rear end of the car such as by preventing compression of the rear springs, it will be seen readily, by reference to Fig. 14 that the oscillation center may be shifted from the C. G. to the rear with a major reduction in the vertical displacement of the front end of the vehicle and a corresponding reduction in pitch tendency. This "support" ability in the rear spring can be insured by providing sufficient inter-leaf friction. In the conventional spring, however, friction of sufficient magnitude to be helpful in this manner is, unfortunately, also present and objectionable when the obstruction attempts to pass under the rear wheels.

Considering this problem further, we discover that what we need in a rear spring to insure a ride which is flat and pitch-free is a substantial amount of frictional resistance when the front wheels meet an obstruction but that, an instant later when the obstruction has passed under the fronts and is just meeting the rears, we wish this rear-spring friction to become momentarily very low and permit the wheel to rise over the obstruction unrestrained by friction.

This objective has been difficult to accomplish because, for both cases of front or rear wheels passing over an obstruction, compression movement of the rear spring is involved. The rear spring, to accomplish this objective, should resist frictionally compression in the first case and be friction-free in compression in the second. The lead-antimony insert has the needed characteristic to meet this difficult requirement to a surprising degree and has been found quite valuable in securing freedom from pitch tendency.

As previously explained, this friction material, as used between normal steel spring leaves, can be adjusted by changing the antimony percentage to provide a considerable range of static friction or "support" value. Also, because of its high damping value at low velocity and low damping value at high velocity, it can, in effect, be calibrated actually to distinguish between front and rear-wheel obstructions.

From road observations, it was found that this lead-antimony alloy, as used, was relatively incapable of providing wheel-bounce damping at frequencies around 400 to 600 cycles per min. which could be obtained readily with bronze inserts. At the same time the lead-alloy inserts, on a test machine, showed roughly twice the damping value of bronze at velocities corresponding to normal ride frequencies of 50 to 100 cycles per min. These observations led to the eventual pitch-control applications here reported.

One interesting and important commercial sidelight on the flat or pitch-free ride is that, once a satisfactory degree of flatness control is attained, only one ride calibration – shock-absorber valving and spring inserts – is required for all body types. It may be seen in this connection that, so long as there is a substantial pitching tendency about any center of oscillation located longitudinally near the center of the wheelbase, different body types having different mass distribution, different spare tire locations, and so on, will, for best results, require different control specifications. Conversely, when pitching is restrained, the sprung-mass distribution or longitudinal polar moment no longer has its former influence on the ride calibration and a corresponding simplification in the number of ride specifications required for a given "line" of cars is accomplished.

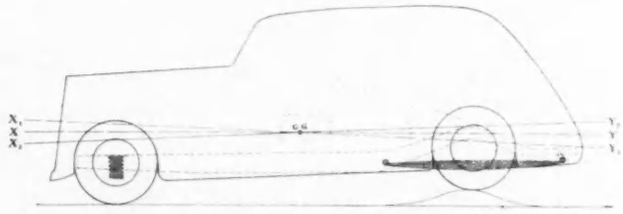


Fig. 13 – Conventional Car with Obstruction Passing under Rear Wheels

Secondary Factors Affecting Ride Character

A discussion of ride control and calibration problems without reference to ride character would be incomplete. At least of equal, and often considered of greater, importance than ride balance are the secondary factors which affect the character of the ride and actually determine the character of the car. Ride character is an extremely important matter, very closely

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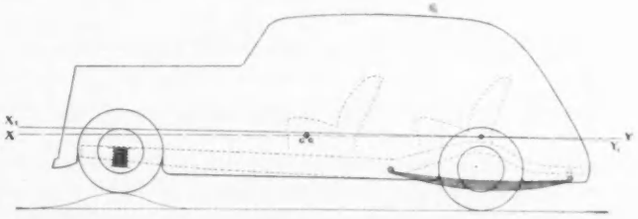


Fig. 14 – Obstruction Passing under Front Wheels of Car with Provision for Supporting the Rear End of the Car, such as by Preventing Compression of the Rear Springs

Bending Stresses in Box Beams as Influenced by Shear Deformation

By Paul Kuhn

Assistant Aeronautical Engineer,
National Advisory Committee for Aeronautics

THE history of the box-beam problem is outlined. This problem has interested naval architects for a long time, but no design methods suitable for aeronautical applications have been evolved. The advent of stressed-skin construction in aeronautics has necessitated further development of the theory.

The problem of redistribution of bending stresses caused by shear deformation on the cover sheet, which cannot be avoided even in the most favorable cases, is considered as the most fundamental problem in box-beam design. Simplifying assumptions are discussed that permit a mathematical approach to the problem. Solutions for special cases finally are combined into a method of analysis intended chiefly for box-beam wings.

Analytical formulas are given for a simple case. With the help of these formulas, various structural arrangements of a small transport-size wing are compared. Numerical values are given for each case to show the error introduced by using the ordinary bending theory, and also to show how much extra weight is required to make up for the inefficiency caused by shear deformation.

THE box-beam problem is not a new one. Box beams have been built for hundreds of years in large sizes, namely, as ships, and strength calculations on box beams, therefore, antedate the flights of the Wright brothers by many years. Iron ships and the formula: stress $= Mc/I$, were introduced into engineering at about the same time. As usual, practice developed more rapidly than theory, and the time came when engineers began to question the accuracy of the formula. In 1894, deflection measurements made on a ship in England proved that the deflections under load were larger than the theory predicted. These tests, and especially the famous tests on the destroyer Wolf by Biles, furnished the basis for arguments that, to this day, are occupying naval architects.

Obviously, there was ineffectiveness somewhere, but progress in tracking down this ineffectiveness was slow. Looking back now, it is not difficult to see the reason. The ship presents a bad combination of three factors: large cutouts, namely, the loading hatches; large, sudden variations of cross-section caused by the upper decks; and, finally, shear deformation.

Cutouts and sudden variations of cross-section are beyond the scope of this paper. Credit for being the first to point out the importance of shear stresses, as well as other items, belongs perhaps to Bruhns, Director of the Scandinavian ship-classification society, Norske Veritas. Pietzker, German naval constructor, is generally given credit for introducing the concept of effective width of a plate buckled under compressive load between 1905 and 1910; he was inspired by American tests. Unfortunately, Pietzker himself discredited his own theory to some extent and for some time by assuming that the effective width on the tension side was the same as on the compression side, which he estimated as 40 times the plate thickness. This theory of the tension side naturally found no favor, and it slowed down the acceptance of the theory for the compression side considerably. It is interesting to note, however, that many years later the shear-deformation theory justified, at least qualitatively, Pietzker's claim that the tension side is not fully effective.

In 1924, von Kármán published a method for finding the effective width of T-beams with wide flanges, intended chiefly for civil-engineering structures. In 1926, Schnadel used somewhat similar methods to study the effective width of the deck plating in ships. These studies were of theoretical interest to aeronautical engineers because they showed clearly that the reason for the ineffectiveness of wide flanges was the shear deformation on these flanges. They were, however, of little practical interest because the methods used precluded the possibility of taking into account the existence of stringers and of the formation of diagonal-tension fields.

Development of Methods of Analysis

In aeronautical applications of the box beam, the cover practically always consists of a stiffened sheet or, to use a technical term, the cover is an anisotropic (orthotropic) plate. In order to make this plate problem amenable to mathematical calculations, it is advisable to idealize the actual structure into a skeleton structure that embodies only the most important properties of the actual structure. Fig 1 *a* shows a mechanical model of the corresponding skeleton structure. Since the structure is symmetrical, only one-half of it is represented in the model. The helical springs in Fig. 1 *b* represent the stringers with their most important characteristic, namely, elastic resistance to changes in length. The coil springs with attached levers represent the sheet with its elastic resistance to angular changes, or shear strain. The angles between the levers and the horizontal correspond to the shear strains of the structure. Fig. 1 *c* shows the stress distribution in the panel. In the stringer F, which is directly under the load P, the stress decreases from the maximum at the top to the minimum at the bottom. The

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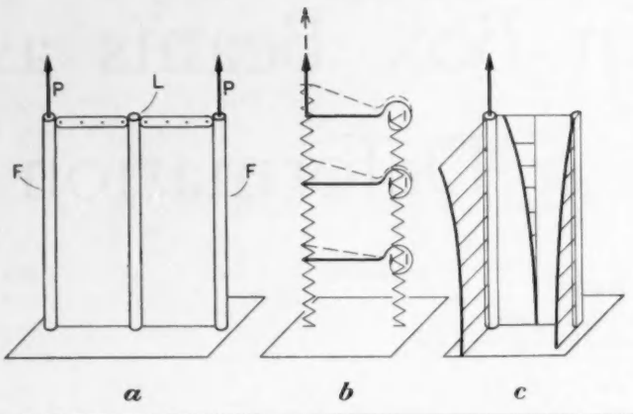


Fig. 1 – Axially Loaded Panel

stress in stringer *L* is zero at the top and reaches a maximum at the bottom. As the length of the panel and the shear stiffness of the sheet increase, the stresses in both stringers approach asymptotically the same value. The shear stress is a

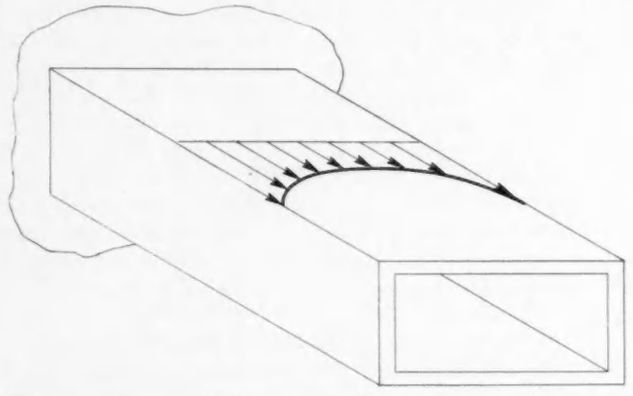


Fig. 2 – Stress Distribution in Box Beam

maximum at the top and zero at the bottom of the panel. The axially loaded panel may be considered as a special case of a box beam, and there is no difficulty in applying the same simplifying concepts to more general cases.

The first analytical solution was obtained by Younger. He considered a beam as shown in Fig. 2, and assumed the bending-moment diagram to be a cosine curve. The cover was assumed to be a plate without stiffeners, which may be considered as the limiting case of a sheet with infinitely many stiffeners. The mathematical expressions used by Younger conform with the physical assumptions depicted by the model shown in Fig. 1 *b*. The stress distribution calculated by Younger is indicated in Fig. 2. The ratio of the average stress, computed by the ordinary bending theory, to the maximum stress at the corner was introduced as an efficiency factor.

Younger advocated the use of this factor for other cases, arguing that it probably would not change very much for different loading conditions. Unfortunately, this assumption is not true as can be shown by a simple extension of Younger's result. The bending moment may be assumed to be in the general form of a multiple cosine wave, and any desired bending-moment diagram can then be represented by superposing a number of these waves, as in a Fourier series. This superposition was carried out for the two loading cases shown in Fig. 3 besides the basic cosine curve, that is, for a moment constant along the span and for a triangular moment distribution such as given by a concentrated load at the tip. For a reason that will become apparent later, the reciprocal of the efficiency factor, the ratio of maximum stress to average stress, is more convenient than the efficiency factor for this investigation. Fig. 3 *b* shows the spanwise distribution of these stress ratios for the three loading cases indicated in Fig. 3 *a*. It is obviously necessary to take the shape of the bending-moment diagram into account. The use of a simple efficiency factor is unsatisfactory, and a more detailed analysis will be necessary.

In the majority of practical cases, the cross-section of the beam varies along the span. A very brief study of this problem showed that there was little hope of obtaining mathematical solutions for multi-stringer beams of variable cross-section. Attention was therefore directed to the beam with a single stringer attached to the cover sheet. Analytical solutions could be obtained very easily for simple cases such as shown in Fig. 4. For the general case of variable cross-section and arbitrary loading, a numerical method of solution by a trial-and-error process was found to be feasible.

The final solution of the problem was now effected by combining the methods and solutions thus far developed. The analysis of the multi-stringer beam was divided into two separate steps: The calculation of the stresses in the corner flange *F* (Figs. 1 and 4), and the calculation of the stresses in the stringers.

The calculation of the flange stresses is made with the help of a device not uncommon in the analysis of trusses, namely, the device of a simple phantom member that is substituted temporarily for a complicated assembly of members. In this case, the many stringers distributed along the chord are replaced by a single phantom stringer located on the centerline of the beam. The cross-sectional area of the phantom stringer is equal to the total area of the actual stringers (including effective width of sheet) multiplied by a correction factor. This factor is usually only slightly above unity and does not affect the results very much.

The analysis of the structure containing the phantom stringer gives the stresses in the flange *F* and the shear stresses in the skin next to the flange. Unless the cover has appreciable camber, this information is all that is required, at least for preliminary design, because the design stress for the most highly stressed stringer is nearly equal to the flange stress.

Although the flange stresses must be calculated for the entire span in one operation, it is possible to calculate the stringer stresses for any desired station independently. The calculation for any desired station is made by using the equation of static

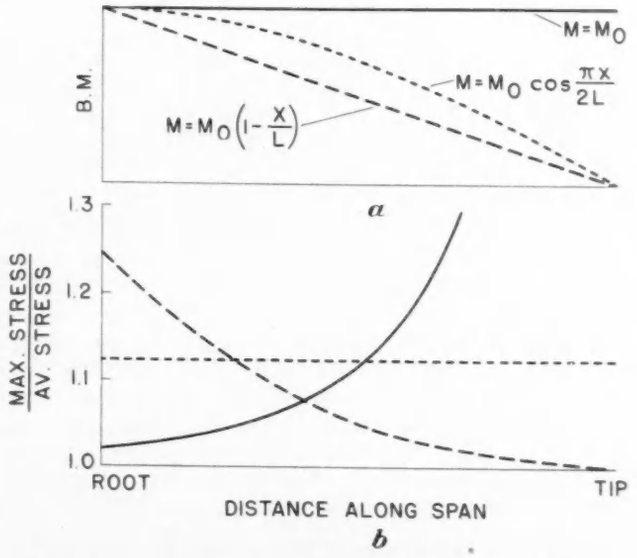


Fig. 3 – Stress Ratios for Different Types of Bending-Moment Diagrams

equilibrium with the external bending moment and adopting the law of chordwise distribution that was found in analytical solutions for multi-stringer panels, a hyperbolic cosine law. As illustration of the results obtained, Fig. 5 shows a typical spanwise stress-distribution curve for a single-stringer beam with a tip load. It will be seen that, in the outboard region, the deviations from the ordinary bending theory are small. With increasing distance from the tip, however, the flange stress progressively increases over the Mc/I stress, whereas the stringer stress drops progressively below the Mc/I stress.

Fig. 6 shows typical chordwise stress-distribution curves for a beam with a flat cover and for a beam with a cambered cover. In a beam with flat cover, the flange stress is always the highest stress. In a cambered beam, the highest stress may occur at the flange or it may occur at the stringer most remote from the neutral axis, depending on the shear stiffness of the cover sheet. Fig. 6 *b* may be used to illustrate the inadequacy of strength tests without correlated theory. A test failure in the corner flange or in the adjacent stringer would be attributed to insufficient strength or allowable stress in these members, if the test is interpreted in the light of the incorrect theory. In fact, the trouble lies in the failure of other parts

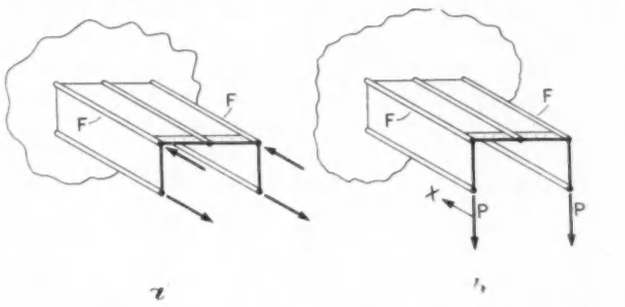


Fig. 4 – Simple Case Used for Analytical Solution

of the structure to cooperate fully on account of insufficient stiffness somewhere in the structure.

A number of cutout problems also can be treated approximately by the method developed. Fig. 7 shows, as an example, a fuselage with a large cockpit cutout. The loads are carried past the cutout by longerons, and the distribution of these concentrated loads into the aftersection of the fuselage can be treated with a fair degree of approximation by the methods developed for axially loaded panels.

As far as analysis of beams is concerned, the method is sufficiently general to take care of any spanwise variation of cross-section or loading, with the usual limitation, of course, that the analysis will not be very exact in the vicinity of a major abrupt change. Chordwise variations of section properties, such as sheet thickness, have not been considered, but this limitation should not be serious in most practical cases.

It is obvious that the method of analysis outlined contains approximations and assumptions beyond those made to obtain the skeleton structure. The validity of the method must therefore be confirmed by strain-gage tests.

Fairly extensive strain-gage tests are on hand for some axially loaded panels and a number of beams of various sizes and shapes, ranging from the extreme cases of single-stringer structures to the opposite extreme of plate or sheet without any distinct stringers. The agreement is not always good. In cases where there are noticeable systematic discrepancies, however, it is usually possible to point out an important violation of the assumptions made in the theory. It is difficult to build a practicable test specimen that fulfills all the assumptions of the theory. Other difficulties, more or less widely

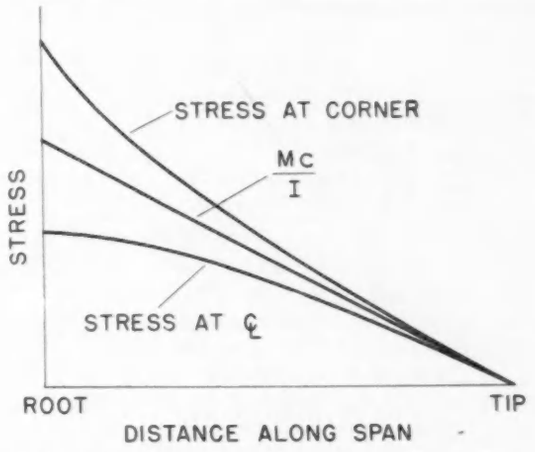


Fig. 5 – Spanwise Stress Distribution in Box Beam

known, are attached to strain-gage tests on sheet-metal structures. Finally, even carefully built and tested sheet-metal structures exhibit inexplicable irregularities of action. Taking all these factors into consideration, it is probably safe to state that the method of analysis is confirmed by these tests as well as can be expected. The details of the method, as well as the experimental evidence, will be published during 1938 in a technical report by the National Advisory Committee for Aeronautics.

Formulas Used for Comparative Study

The analysis of tapered beams with arbitrary loadings is too complicated to permit a very general study. However, a number of interesting points can be brought out by studying the simple case of a beam of constant cross-section with a concentrated load at the tip. It might be mentioned that the solution for this case also applies to a beam on which the load is uniformly distributed along the span, provided that the depth of the shear web tapers linearly to zero at the tip, because, in both cases, the total force in the cover M/h is proportional to the distance from the tip.

For the range to be covered in this study, the area of the phantom stringer may be taken as equal to the sum of the stringers attached to the sheet and of all the sheet considered effective in carrying bending stresses, that is, the correction factor may be set equal to unity. The beam with the phantom stringer then corresponds to the case shown in Fig. 4 *b*.

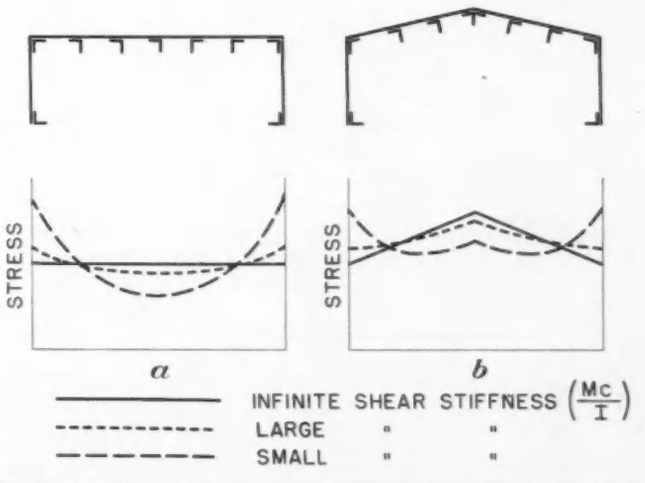


Fig. 6 – Chordwise Stress Distribution in Box Beams

The fundamental parameter appearing in the solution of single-stringer structures is defined by

$$K^2 = \frac{G_e t}{Eb}\left(\frac{1}{A_F} + \frac{1}{A_L}\right) \tag{1}$$

where

G_e is the effective shear modulus of sheet.

t, thickness of sheet.

b, width of sheet for half-beam.

A_F, area of corner-flange.

A_L, area of phantom stringer for half-beam, or total area of all material taking bending stresses exclusive of A_F.

It will be noted that all data are given for the half-beam, as indicated again in Fig. 8, which shows the basic dimensions selected for the comparative study. The parameter K always appears in the form Kx or KL, where x is the coordinate of the station and L is the length of the beam. The factor KL may be called the "shear stiffness factor," and the magnitude of this factor is a convenient measure for the errors that may be expected from the use of the ordinary bending theory, which assumes this factor to be infinite. It will be seen that these errors decrease with increase of shear stiffness G_e, thickness t, or length L, and with decrease of width b.

For the purpose of comparison, the most important item is the maximum stress in the beam, which occurs in the corner flange at the root. For $KL > 5$, it is given by the simple approximation

$$\sigma_F = \frac{M}{h A_T}\left(1 + \frac{A_L}{A_F KL}\right) \tag{2}$$

where

$$A_T = A_F + A_L$$

The second term in the bracket is the correction to be added to the ordinary bending formula to account for shear deformation.

For the purpose of estimating the effective shear stiffness of the sheet, the shear stress in the sheet must be obtained. This stress varies along the beam but, in the decisive outboard region, it varies comparatively slowly in the case considered, so that the maximum shear stress at the tip may be used as a criterion. This stress is given by

$$\tau_{max.} = \frac{P A_L}{th A_T} \tag{3}$$

the correction for shear deformation being negligible.

The shear stress $\tau_{crit.}$ at which the sheet begins to buckle into a diagonal-tension field can be calculated by standard formulas. In this study, the sheet was assumed to be supported simply at the stringers, which are spaced 4 in. apart.

Until the sheet buckles, the effective shear modulus equals the shear modulus of the material. As the buckles deepen, the effective shear modulus decreases. For the present study, it was assumed that the state of a pure diagonal-tension field is reached when the buckling shear stress is exceeded six times, and that the variation is linear between these two limits. This assumption gives the formula

$$\frac{G_e}{E} = 0.40\left(1.075 - 0.075\,\frac{\tau_{max.}}{\tau_{crit.}}\right) \tag{4}$$

This formula is based on some preliminary N.A.C.A. tests obtained several years ago and is preferred on account of its simplicity for the present purpose to information published since then by Wagner. At any rate, the bending stresses are not influenced critically by variations of the shear stiffness.

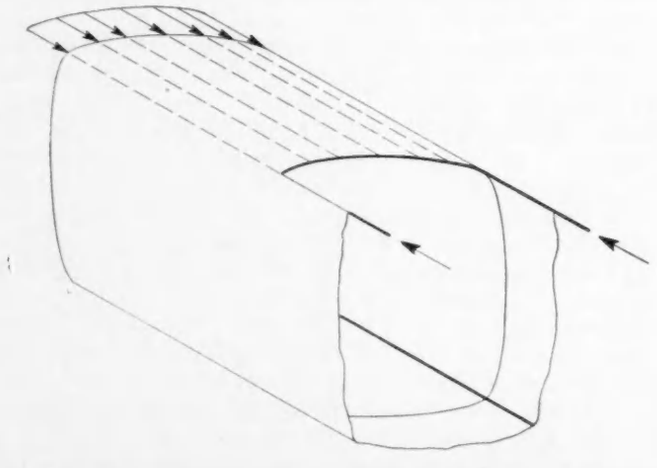


Fig. 7 – Fuselage with Large Cockpit Cutout

Comparative Study of Typical Arrangements

The dimensions of the root section of a typical small-size transport wing were chosen as the basis for the comparative study, and are shown in Fig. 8*a*. The corresponding beam

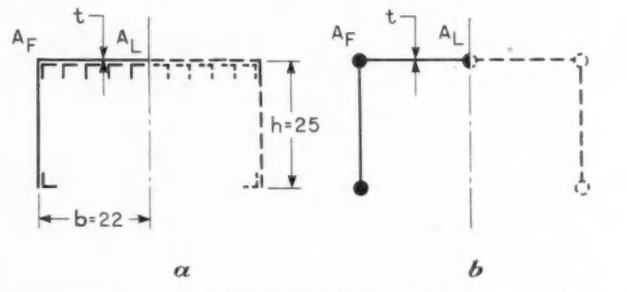


Fig. 8 – Actual and Phantom Section of Beam for Comparative Study

with the phantom stringer is shown in Fig. 8*b*. The lower cover is omitted in all figures in order to draw attention to the fact that only variations of one cover are investigated. The length of the beam is $L = 300$ in.

A number of typical arrangements of the cover material will be investigated, and two items will be calculated for each beam. The first item is the stress ratio, *SR*, which is the ratio of the maximum stress given by Formula (2) to the stress given by the ordinary bending formula, making proper allowances for effective width in both cases where indicated. The stress ratio is given by the bracketed expression in Formula (2) and shows directly the error introduced by using the ordinary bending formula. The stress calculated for the ideal case of no shear deformation, that is, by the ordinary bending formula, will be chosen as allowable stress. The stress ratio, *SR*, will therefore indicate the over-stress in each beam, or the negative margin of safety if subtracted from unity.

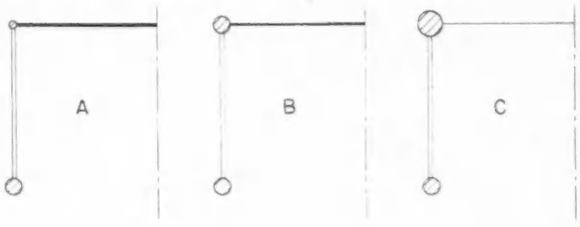
The beams will then be redesigned to reduce the maximum stress to the allowable stress. The entire cross-sectional area of the redesigned cover, including corner flange, stiffeners, and skin, divided by the original area of 2 sq. in. gives the weight ratio, *WR*, of the beam and is a measure of the structural efficiency of the beam. The redesign may be made in a number of ways. The study was confined to two possible ways.

Consider now the fundamental case of infinite shear stiffness. The stress in the cover is given by the ordinary bending formula.

$$\sigma = \frac{M}{h A_T} = \frac{1{,}500{,}000}{25 \times 2} = 30{,}000 \text{ lb. per sq. in.}$$

This stress of 30,000 lb. per sq. in. will be chosen as allowable stress for all cases.

Next, consider the series of three beams shown in Fig. 9.



A_F	0.5	1.0	1.5
A_L	1.5	1.0	0.5
A_F/A_L	1:3	1:1	3:1
$t=A_L/b$	0.0682	0.0454	0.0227
SR	1.17	1.08	1.03

Fig. 9 – Three Beams Treated as Ideal Cases for Comparison

As indicated by the schematic method of drawing, they will be treated as somewhat ideal cases, namely, as cases in which the sheet is fully effective in carrying bending stresses. In practice, this assumption would be true on the compression side only at very low loads; on the tension side, of course, it would apply at all loads provided that no very deep diagonal-tension folds are formed.

The three beams *A, B,* and *C* differ in the relation between corner flange area A_F and skin area A_L. The lowest ratio of A_F to A_L corresponds to that of the actual wing from which the basic dimensions were taken. The highest ratio corresponds to a two-spar wing with metal covering, rather than a true box beam; most of the material is concentrated at the shear web, and the skin is reduced to about the practicable minimum. It will be seen that, in the case of the two-spar metal-covered wing, Beam *C,* the stress ratio is very little above unity; the error introduced by using the ordinary bending formula is, therefore, very small. This result is quite natural. Such a small amount of the total material is exposed to the detrimental effect of shear deformation on the sheet that the percentage effect on the flange is very small. In the case of the true box beam, however, represented by Beam *A,* the effect is very appreciable, amounting to 17 per cent in spite of the fact that this case is an ideal one, which could be realized only if it were possible to design the cover for tension only. The occurrence of inverted loads precludes this possibility, so that the cases *A, B,* and *C* still represent practically unattainable ideals.

For the following cases, the study will be confined to box-beam types, that is, beams with the ratio of A_F to A_L equal to one-third.

In the series of beams *D, E,* and *F,* which are shown in Table 1, the cover has been designed so that it can take compressive stresses. The beam with the corrugated cover has the lowest stress ratio of the three for Condition 1, because all the material in the sheet resists normal as well as shearing stresses. The beams with separate stringers have progressively higher stress ratios as more and more material is diverted from the task of resisting shear deformation. It is still assumed, however, that the sheet is fully effective in carrying normal stresses and that it does not form diagonal-tension folds. The values given for Condition 1 therefore apply only for low tensile stresses or very low compressive stresses, excepting the beam with the corrugated cover.

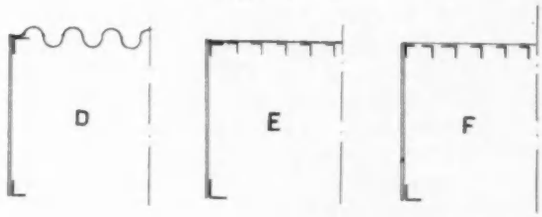
The values given for Condition 2 apply to the condition of compressive design load. It is assumed that the skin takes part in carrying bending stresses only to a total amount of 165 *t,* which is added to the stringer area to obtain A_L. At the same time, the reduced effective shear modulus is computed by the formulas given. The stress ratios are based on the assumption that the ordinary bending formula is applied to the effective cross-section, not to the actual cross-section. The stress ratios are lower than they were for Condition 1, but the actual maximum stresses are much higher.

The beams will now be redesigned to reduce the maximum stress to the allowable stress of 30,000 lb. per sq. in. Redesign 1 will be made by multiplying all component parts of the cover, skin, stringers, and corner flange, by the same factor, until the maximum stress of 30,000 lb. per sq. in. is reached. This factor is the weight ratio, *WR,* because all beams had originally the same cross-sectional area (2 sq. in.) in the cover. It will be seen that the corrugated cover is more efficient than the stiffened covers, but the two stiffened covers happen to come out equal. That this result is only accidental will be obvious from the next redesign.

In Redesign 2, it was assumed that the area of the corner flange A_F is kept constant; and also, in the beams *E* and *F* with stiffened cover, that the thickness of the cover sheet is kept constant. It will be seen for beams *E* and *F* that the stress ratios are very high, indicating very inefficient beams, and also that beam *F* with the heavy stringers and light skin is very much heavier than beam *E* with the light stringers and heavier skin.

A question often asked concerns the relative merits of two

Table 1 – Series of Beams Compared for Different Conditions and Redesigns



	D	E	F
	Condition 1 (tension)		
A_F	0.50	0.50	0.50
Skin	1.50	.75	.50
Stringer	-	.75	1.00
A_L	1.50	1.50	1.50
t	.0560	.0341	.0227
G/E	.400	.400	.400
σ_{max}	36,400	37,400	39,100
SR	1.21	1.25	1.30
	Condition 2 (compression)		
A_F	0.50	.50	.50
Skin	1.50	.75	.50
Stringer	-	.75	1.00
A_L	1.50	.94	1.085
t	.0560	.0341	.0227
G/E	.400	.397	.313
σ_{max}	36,400	47,750	46,700
SR	1.21	1.14	1.23
	Redesign 1		
A_F	.605	.750	.750
Skin	1.819	1.126	.750
Stringer	-	1.126	1.500
A_L	1.819	1.558	1.692
t	.0680	.0512	.0341
G/E	.400	.400	.395
σ_{max}	30,000	30,000	30,000
SR	1.30	1.16	1.22
WR	1.21	1.50	1.50
	Redesign 2		
A_F	.500	.500	.500
Skin	2.040	.750	.500
Stringer	-	2.100	2.800
A_L	2.040	2.292	2.885
t	.0761	.0341	.0227
G/E	.400	.397	.313
σ_{max}	30,000	30,000	30,000
SR	1.25	1.39	1.70
WR	1.25	1.67	1.91

shear-web beams and multiple shear-web beams. On the basis of the assumptions made in this comparison, any additional shear web would increase the efficiency of the beam by decreasing the width *b*. Theoretically, this change would involve no increase in weight because it would merely mean splitting the existing shear web into independent parts. In the practical case, the necessity of designing for torsion as well as bending changes the picture; but this fact and other considerations of a more practical nature remove the comparison from the scope of this paper.

Lessons From Comparative Study

A number of lessons can be drawn from the comparative values given. The first and most important one is that the error committed by using the ordinary bending formula is very much larger than commonly supposed even for such nearly ideal cases as corrugated covering. It should be emphasized again that there is a loss of efficiency on the tension side as well as on the compression side, the main difference between the two sides lying in the reduced effective width of sheet on the compression side. There is also a slight difference in effective shear modulus, but this difference is quite difficult to evaluate numerically and does not affect the stresses critically, as pointed out before. It also should be noted that the beam used in the comparative study is fairly long in relation to its width and that the stress ratio increases rapidly if the ratio of length to width decreases.

The second lesson is that high efficiency is obtained by utilizing all the material to resist shearing as well as bending stresses. In the covers consisting of a stiffened sheet, there are two sources of inefficiency: part of the sheet does not carry bending stresses, and the stringers carry no shearing stresses.

The third lesson is drawn from the comparison of beams *E* and *F*. Two sources of inefficiency in stiffened sheet covers were just mentioned. Both effects are aggravated by making the stringers heavier at the expense of the sheet and, at the same time, a third source of inefficiency is opened up: the thinner sheet has a lower effective shear modulus because it buckles sooner.

It is common practice to establish the allowable stresses for sheet-stiffener combinations by testing sample panels in testing machines. Studies made on such panels usually have led to the conclusion that the best strength-weight ratio is obtained by putting as much material as possible into the stiffeners, allotting to the skin only as much as necessary from practical considerations. The comparison of beams *E* and *F* shows that the strength-weight ratios obtained for test panels are not true criteria for their efficiency in the complete structure. The combination of heavy stringers and light skin, in many cases, is handicapped by a more uneven distribution of stress, resulting in a higher stress on the critical stringers. This handicap may be sufficient to overbalance the advantage of a higher allowable stress. Unfortunately, as a glance at the figures for beams *E* and *F* shows, about the only general conclusion that can be drawn is that no general conclusion can be drawn; each case must be considered on its own individual merits. Reductions in skin thickness, however, should always be considered with suspicion and should be investigated carefully.

The fallacy of judging the efficiency of skin-stringer combinations by their testing-machine capabilities is only one symptom of a more general failing. This failing is the tendency to consider the semi-monocoque structure as a determinate structure, to forget that it is an indeterminate structure in which the stress distribution depends on the relative stiffnesses of the component parts, and to forget that stiffness and strength are not always directly related. The designer experienced in stressed-skin work has learned by test experience the importance of distributing stresses; in other words, he has developed a subconscious qualitative appraisal of the situation. Unfortunately, the stress analyst does not always follow up quantitatively, partly from the habit of reducing the structure to determinacy by making assumptions, partly because he has not the time necessary for laborious calculations of indeterminate structures. It must be admitted that the semi-monocoque structure is so complex that sweeping assumptions must be made to arrive at results. Nevertheless, the habit of looking at such problems as statically indeterminate problems should be cultivated.

As an example, consider the question of reducing the stringer stresses at the root of a beam. Looking at the problem as a determinate problem, the answer would be: make the stringers heavier. The comparative study showed that this solution might make conditions worse instead of better. It might be more economical to increase the thickness of the skin to insure more uniform action. It might even be more economical to add to the skin thickness not in the apparently critical region at the root but at the opposite end of the beam, at the tip, because the deformations here are so much larger that they are easier to control. Some brief calculations indicate that this case is not likely to arise in wing beams of the type considered, but it might arise in stub wings. It is a possibility that should be kept in mind on sharply tapered wings, too, because the conditions in such wings may vary materially from those in the constant-section wings upon which this study was based.

The study has dealt only with beams having flat covers. In beams with cambered covers, any inefficiency caused by shear deformation will be aggravated because the stresses tend to wander off to the stringers near the corner flange, which have a smaller moment arm about the neutral axis than the stringers near the centerline of the beam.

Ride Controls and Calibration

(*Concluded from page* 318)

associated with the problem of ride calibration. It should be distinguished from the problem of ride balance.

Character has to do with such factors as the harshness level or, conversely, the impression of mellowness which the passenger feels through the cushions, floor, or steering wheel. It has to do with the compromise between so-called jiggle and jolt. Also important in this connection is the matter of proper control of steering-wheel fight and shock in the steering – the freedom from nervousness of steering.

Other factors of importance include the so-called hard-tire effect, the degree of floating tendency permitted, the matter of wind steering and road handling. A very important factor is the matter of reaction reaching the passengers resulting from excessive wheel bounce, the direct jolt from static friction in the suspension springs – even extending to the matter of the noise level on various road surfaces and to the proper tire construction and air pressure. All of these factors have a very important bearing upon what we may call ride character.

In closing, we invite attention to the important part which the experienced ride engineer must continue to play in ride development and calibration problems. After considering the tremendous number of interrelating factors involved in the attainment of ride excellence, it becomes quite clear why really effective instrumentation has never been developed. Certain factors can be measured, but the maze of compromises involved will always require the experienced observer. On his judgment the excellence of the product, in many important respects, must depend.

Testing of Hypoid Lubricants

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IN the work reported in this paper the performance of several recognized commercial types of hypoid lubricants has been studied and a comparison made of a number of test methods including: film-strength machine tests; various types of continuous-load tests; laboratory shock tests; and road shock tests. This work has been in progress for more than fifteen months, has involved more than 50 lubricants, and more than 150 individual gear tests, each test requiring the use of a new set of gears.

The results of these tests indicate deficiencies in some of the lubricants under certain of the extreme conditions employed. Of the commercial lubricants studied, those that passed the laboratory shock test lubricate hypoids quite satisfactorily under practically all other test conditions where normal temperatures were used. The performance of the lubricants in the gear tests seems to bear but a slight relation to film-strength data as obtained on laboratory test machines.

THE recent widespread adoption of hypoid gears by the automotive industry has focused increased attention on the development of test methods for the true evaluation of hypoid lubricants. Some of the factors that have caused considerable confusion have been the number of laboratory test machines designed to measure that elusive property of a lubricant known as "film strength," and the attempts to use such "film-strength" results as an indication of service performance. The widely divergent values obtained when a given lubricant is tested on these machines, particularly since the lubricating conditions are in most cases somewhat different from those existing in hypoid gears, make one quite skeptical as to their worth in determining the true merit of a hypoid lubricant. Although it cannot be questioned that all of the development work carried out on these machines has been of great value in the study of hypoid lubrication and that future studies undoubtedly will perfect a good laboratory machine, it is becoming increasingly evident that the data obtained with them, using present procedures, can be utilized only as secondary test data for the indication of probable merit, or in setting up general lubricant specifications.

An accurate picture of lubricant performance can be obtained better from gear tests performed in the laboratory or on the proving grounds, together with a study of performance in field service over long periods of time.

[This paper was presented at the Kansas City Section Meeting of the Society, Kansas City, Mo., Mar. 4, 1938.]

Numerous laboratory gear tests have been proposed and are in use. They differ in many respects from one another but, in general, a uniform load is applied to the gears for a definite time interval, while the temperature is maintained approximately constant. Recently the application of shock loads in gear tests has been used by General Motors, Packard, and others on their proving grounds; and, by still others, some attempts have been made at shock loading with laboratory equipment.

In the work covered by this paper, we have studied the performance of several recognized commercial types of hypoid lubricants and made a comparison of a number of test methods including: film-strength machine tests, various types of continuous-load tests, laboratory shock tests, and road shock tests. This work has been in progress for more than fifteen months, has involved more than 50 lubricants, and more than 150 individual gear tests, each test requiring the use of a new set of gears.

The primary requirement of any gear lubricant is that it shall have the ability to prevent serious wear or distortion of the contact surfaces under all reasonable kinds of service during the normal life of the lubricant.

In this discussion we shall consider wear to mean either the removal of metal from bearing or gear surfaces, or the displacement of metal from one part of such surface to another. This second type is sometimes called "scuffing" or "scoring," and is at least an indication of incipient loss of metal from the contact surfaces.

Load-Carrying Capacity

Since prevention of serious wear, as we broadly define it, is the primary function of a lubricant, a test to determine lubricant efficiencies in this respect is of major importance. Test methods and specifications for stability and other of the secondary characteristics have been set up by General Motors and others.

1. Laboratory "Film-Strength" Tests – A number of laboratory test machines have been developed for measurement of lubricant "film strength." Among the better known machines of this type are the S.A.E., Timken, Almen, and Falex machines. When used to test lubricants, these machines indicate that excessive or rapid wear of the bearing surfaces does not occur until a certain load is exceeded. This limiting load is called the "film strength," "load-carrying capacity," or "O. K. load" of the lubricant. There has been serious question of, and some evidence against, the idea that high film strength determined on any test machine and low wear under service conditions go hand in hand. In some quarters, it is even believed that high film strength usually indicates high rate of wear in service. One argument advanced to support this belief is that a suspension of fine abrasive in oil will give a very high film strength value on most of these test machines and naturally an excessive rate of wear.

Laboratory machines, such as the Timken, S.A.E., Falex, and Almen, used to measure the "film strength" of a lubricant,

CHEMICAL COMPOSITION AND PROPERTIES OF LUBRICANTS

LUBRICANT TYPE	Pb. %	S. %	Cl. %	Vis. S.U.S. 210° F.	Acid No.	Sap. No.	Ppin. No.	Non-Comb. Sed. %	Foam Vol. Incr. %
1. LEAD SOAP ACTIVE SULFUR	2.62	2.67	0.00	91	3.61	33.9	.05	.03	27
2. LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	0.78	2.80	1.08	98	1.88	68.1	.25	.05	35
3. INACTIVE SULFUR SAPONIFIABLE OIL	0.00	1.50	0.10	98	1.10	31.7	TRACE	.00	28
4. SULFUR CHLORIDE SAPONIFIABLE OIL	0.00	0.78	0.75	105	0.81	21.6	TRACE	.00	11
5. CHLORINE COMPOUND	0.00	0.50	1.10	97	0.22	12.6	TRACE	.01	31
6. CHLORINE COMPOUND SULFUR COMPOUND	0.00	1.10	2.96	88	0.03	11.9	TRACE	.01	25
7. MINERAL OIL	0.00	0.13	0.00	88	0.03	0.8	TRACE	.01	30

Fig. 1

RESULTS OF FILM STRENGTH MACHINE TESTS

LUBRICANT	ALMEN LBS.	FALEX LBS.	TIMKEN LBS.	S. A. E. MACHINE 500 R.P.M. LBS.	750 R.P.M. LBS.	1000 R.P.M. LBS.	TIMKEN WEAR 20 Lb. Load MGS.
1. LEAD SOAP ACTIVE SULFUR	30+	4500+	25	580+	580+	580+	1.0
2. *LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	16+	2500+	68+	497+	133+	98+	2.8+
3. INACTIVE SULFUR SAPONIFIABLE OIL	30+	2375	86	147	96	81	1.6
4. SULFUR CHLORIDE SAPONIFIABLE OIL	30+	4500+	33	491	92	90	2.2
5. CHLORINE COMPOUND	20	1750	77	580+	116	305	3.3
6. CHLORINE COMPOUND SULFUR COMPOUND	30+	4500+	59	580+	480	330	2.9
7. MINERAL OIL	6	500	7	102	0	0	99.0

*ERRATIC ON ALL FILM STRENGTH MACHINES.

Fig. 2

may also be run for long periods of time at moderate loads in order to determine the amount of wear as indicated by loss in weight or change in dimension of the test pieces.

One of the difficulties has been to obtain a machine that could be operated for long periods of time at a reasonably high load, so that all lubricants actually found satisfactory in hypoid service could be compared. Wear tests on several of the machines have been carried out at loads only slightly higher than the load-carrying capacity of a plain mineral oil. Therefore, the direct translation of laboratory-machine wear data into an indication of wear in service is open to serious question. However, general indications of wear in particular types of service may undoubtedly be obtained from the results of such laboratory tests.

2. Laboratory Gear Tests – In order to produce conditions as close as possible to those obtained in service, the Gleason "four-square" test and various modifications of this procedure have been used widely. In general, a uniform loading of the gears over considerable intervals of time has been employed. Both high- and second-gear tests have been made, using lubricant temperatures up to 250 deg. fahr., which are typical of maximum service temperatures.

Wear may be determined by change in appearance of gear surfaces, by clearance changes, by iron content of the used lubricant, or by increased gear noise. Temperature rise also may be used as a general indication of wear, or of imperfect lubrication which probably would result in excessive wear.

Usually these laboratory gear tests have had as an object the finding of suitable lubricants for particular gears in a definite type of service; therefore, it is questionable if any single set-up or procedure of this type ever can be applied satisfactorily to all the variables that exist in field service.

3. Shock-Load Tests – The application of shock loads in laboratory gear tests has been used to a limited extent. Most of the shock-load work has consisted of proving-ground tests such as those employed by General Motors and Packard.

The rate, method, and magnitude of the shock loading in any such procedure are open to question, but it is obvious that these factors must be set within reasonable limits. The intensity of shock loading must be below the point where permanent deformation of the mechanical equipment occurs, and both the General Motors and Packard shock tests meet this requirement.

Proving-ground shock tests are subject to some question as to reproducibility of the rate of application of shock load. This rate depends on tire traction, engine friction, windage, body sway, car loading, and manner of clutch engagement.

Experimental

Lubricants Used – Lead, sulphur, or chlorine, or some combination of these elements, seem to be essential in the composition of the modern hypoid lubricant.

These elements are used generally in the form of compounds, but sulphur may be used in the elemental state dissolved in the mineral oil.

Among the principal types of commercial hypoid lubricants in use today are:

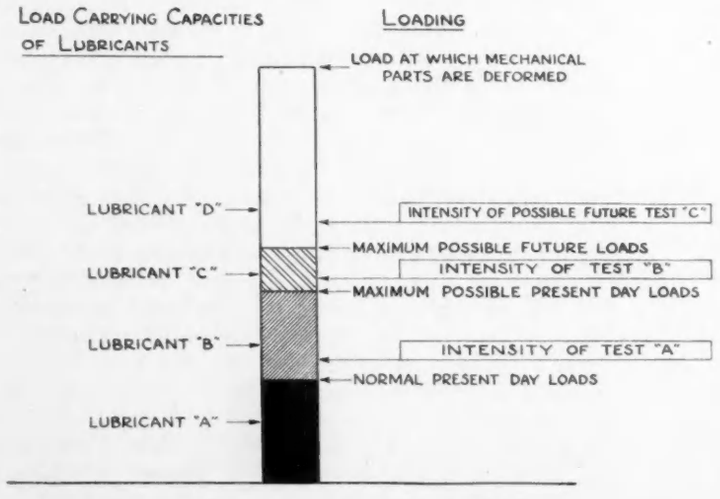


Fig. 3 – Intensity of Shock Test Conditions

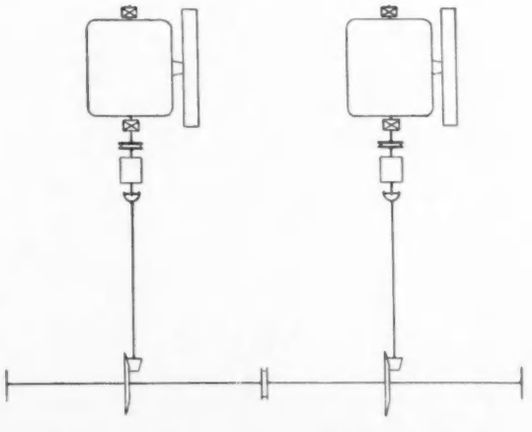


Fig. 4 – Diagram of Dual-Axle Dynamometer Test Set-Up

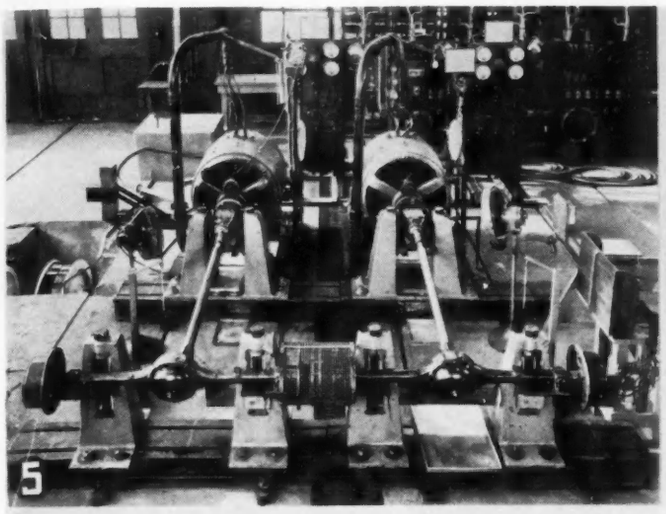


Fig. 5 – Set-Up for Double-Axle Tests

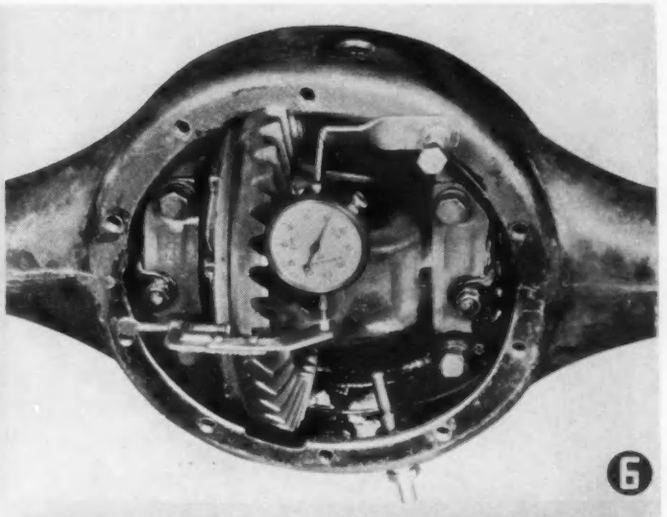


Fig. 6 – Method of Measuring Ring- and Pinion-Gear Clearances

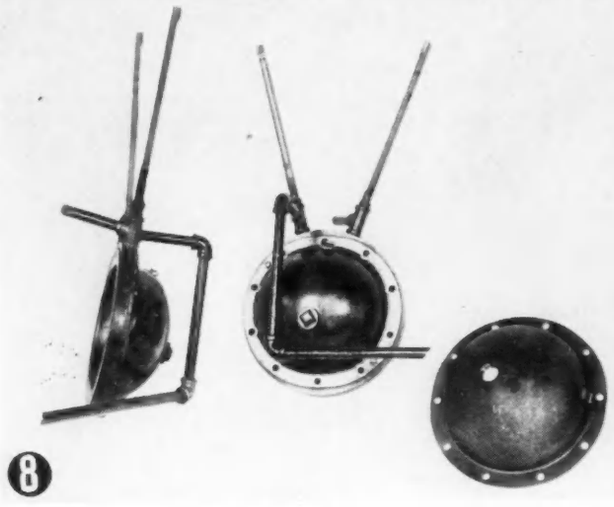


Fig. 8 – Construction of Water Jacket

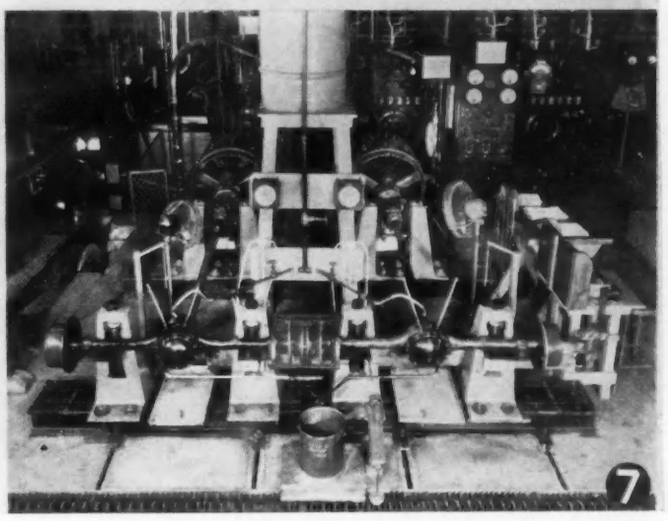


Fig. 7 – Set-Up for Constant-Cooling Test Runs

1. Lead Soap-Active Sulphur – Usually made by compounding a lead soap and a sulphur-treated oil in a suitable mineral oil. Some free or very active sulphur remains in this type of lubricant.

2. Lead Soap-Inactive Sulphur – Chlorine Compound – A lead soap, a sulphurized saponifiable oil in which the sulphur is combined completely, and a chlorine compound are usually blended with the mineral base oil.

3. Inactive Sulphur – Saponifiable Oil – A sulphurized saponifiable oil is blended into the base oil.

4. Sulphur-Chloride Saponifiable Oil – A saponifiable oil, such as lard oil, is treated with sulphur monochloride to yield a product containing both sulphur and chlorine, which is then diluted with lubricating oil.

5. Chlorine Compound – Chlorine compounds are blended into lubricating oils without the use of other addition agents. These blends are used more often for less severe services than hypoid lubrication.

6. Chlorine Compound – Sulphur Compound – A chlorine compound and a separate sulphur compound are dissolved in a suitable base oil.

7. In addition to the preceding, a straight mineral oil was included in this work, simply as a reference lubricant.

From each of the preceding classes a commercially available lubricant, which field reports indicated to be one of the best in its class, was selected. In practically all cases, lubricants in original packages were purchased from recognized distributors. Laboratory data indicated that the samples were representative, checking with results obtained on previous purchases of the same material formerly examined.

It is not intended or implied that the results of these tests made on a single member of each class of lubricants indicate satisfactory performance or failure to perform satisfactorily in service in the case of lubricants belonging to any class.

Failure of a hypoid lubricant to pass any of our tests should not be construed as evidence that the lubricant would not perform satisfactorily in *normal* hypoid service.

Laboratory Data on Lubricants

The compositions and properties of the lubricants used in this work, as determined in the laboratory, are given in Fig. 1.

The performance of these lubricants on film-strength test machines is shown in Fig. 2.

All film-strength values recorded are the average of at least five individual determinations. The Timken test procedure included the use of an automatic loading device employed in a recent cooperative study carried on with the laboratories of the Timken Roller Bearing Co. and a number of lubricant manufacturers.

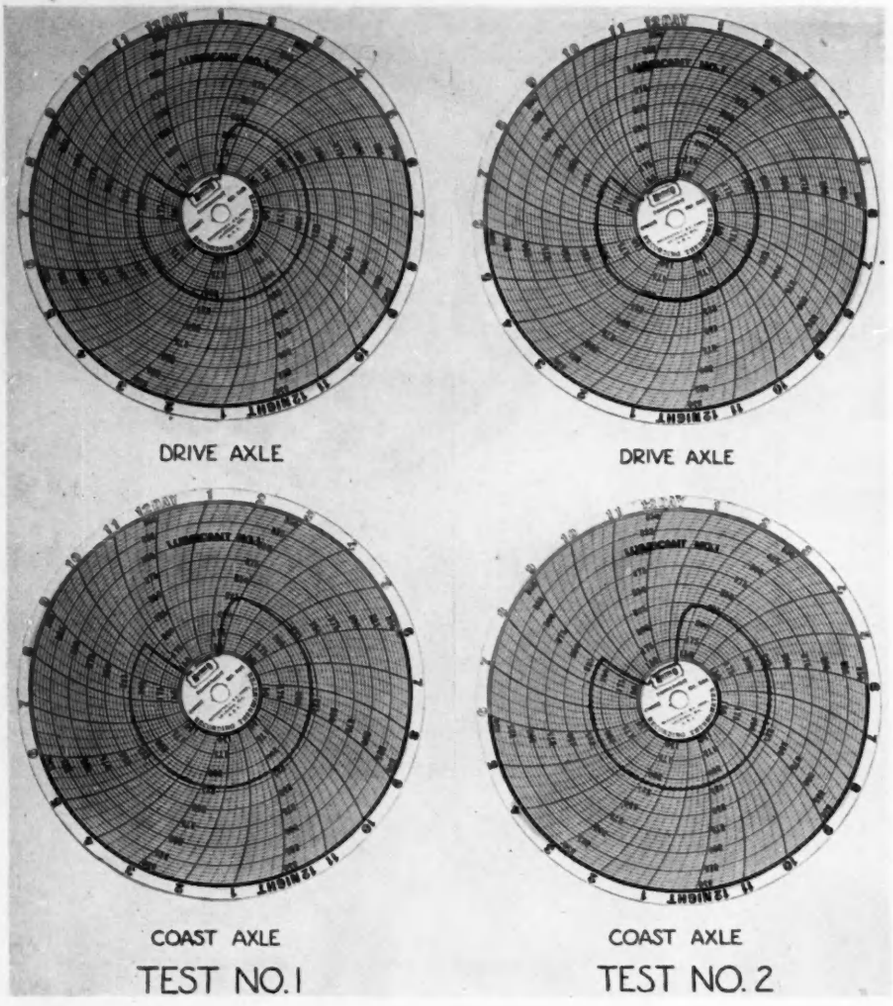


Fig. 9 (*at the left*) – **Typical Temperature Charts Representing Duplicate Runs on the Same Lubricant**

Fig. 10 (*on the opposite page*) – **Representative Temperature Charts of Three Lubricants in the Same Test**

Selection of Axles

In selecting axles for this work it was desirable that they be of a type widely distributed in the field and on which considerable service performance data would be available, and also on which definite procedures for testing of lubricants had been established. For these reasons, the axles of the 1937 Chevrolet and of the 1937 Packard "120" were selected. Other reasons for selecting the Packard axle were the pioneering studies and experience of Packard in hypoid work. Test cars and all replacement parts used in these tests were purchased from local dealers. Factory specifications covering equipment and adjustments were adhered to throughout.

Test Methods and Variables

The principal variables encountered in the use of hypoid gears in service which affect both the rate and amount of wear, are load, temperature, and speed.

A – Loads

1. Uniform Loads – Loads may be uniformly high over long periods of operation, but they should not exceed those obtainable using the maximum power developed by the car engine. These uniform high loads may be applied in high, second, or low gear. Thus it is possible to apply full engine torque in high gear at moderate and high speeds; and also full engine torque, using second or first gear, at low gear speeds, which would mean much higher unit loads on the gears.

To cover these conditions, tests were made at a load equivalent to full engine torque in high gear, using reasonably high temperatures and speeds. Also a series of runs at a load equivalent to full engine torque in second gear at moderately low speed was performed.

Preliminary studies of each type of test were conducted to determine conditions which would differentiate between the various types of lubricants. As a result of such tests the severity of each type of continuous-load test was adjusted so that scoring of the gears was just barely prevented by the use of the lead-soap-active-sulphur lubricant.

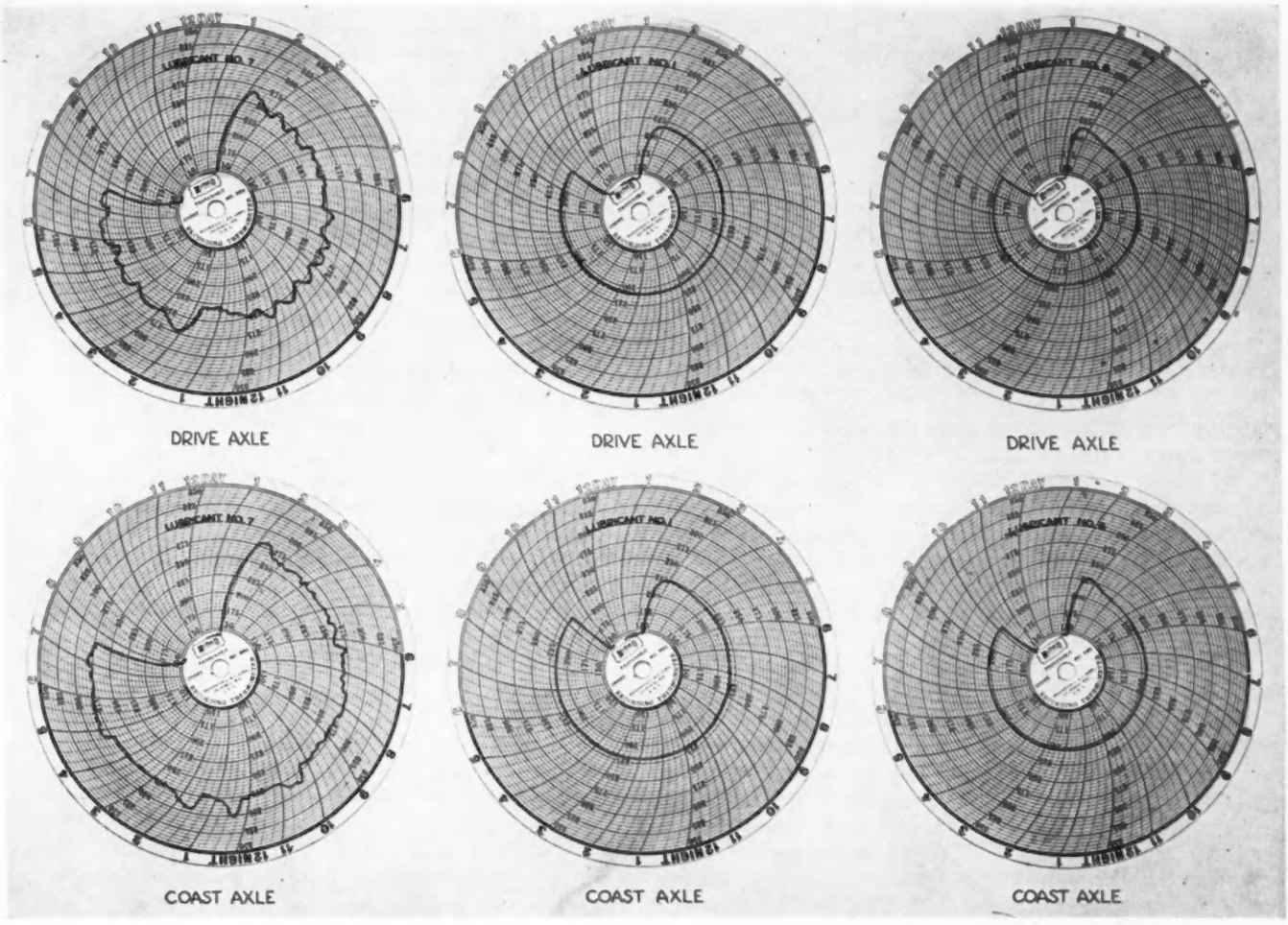
2. Instantaneous or Shock Loads – Shock loads of considerable intensity may be applied occasionally in service. The magnitude and method of application of such loads are subject to question, but it is obvious that conditions used in any test procedure should be:

(a) Of the same order of magnitude as those possible of occurrence in service.

(b) Below those required to deform permanently the mechanical parts of the axle.

Mechanical parts are designed with a considerable safety factor. Therefore, it should be logical to require lubricants, at least at the start of use, to possess a reasonable excess of load-carrying ability above that required in normal service. For the purposes of developing new and more powerful lubricants, anticipating the possibility of even greater gear loading than is used today in hypoid service, it is important that these loads be set high enough to differentiate between the better lubricants.

In Fig. 3 is shown a diagram of intensity of test conditions. This diagram indicates why the intensity should be set reasonably high in order that differentiation may be made between lubricants satisfactory only under present normal service



conditions and those which are satisfactory under exceptional present-day conditions, or under possible future higher loadings.

It is obvious that, even for present-day service, the use of Lubricant *B* would be questionable; whereas, with all other factors equal, Lubricants *C* and *D* would be preferred. The correct intensity of test at present should be above the level of maximum possible present-day loads, as indicated by Test *B* on the diagram.

Therefore, our laboratory shock test conditions were made at least as severe as those used in proving-ground shock tests.

B – Temperatures

1. Maximum Road Temperatures – Rear-axle lubricant temperatures in the 1937 Chevrolet rarely exceed 225 deg. fahr. We have conducted road tests with a recording pyrometer in a good hypoid lubricant, indicating that temperatures in the body of the lubricant seldom exceed 100 to 125 deg. fahr. above air temperatures. This result agrees with the findings of other observers.

Since the temperature generated within a lubricant varies with several factors, such as its viscosity, its chemical composition, and its ability to prevent wear, it is obvious that, under otherwise identical service conditions, the temperatures of different hypoid lubricants will vary. For this reason it was decided that, instead of running tests at constant temperature, it would be much closer to service conditions to use a constant amount of cooling. To determine the amount of cooling to use, preliminary tests were made to find that amount of cooling which would maintain the temperature of the lead-soap-active-sulphur lubricant close to 200 deg. fahr. This same amount of cooling, obtained by circulating 75 deg. fahr. cooling water at a definite rate through a water jacket on the axle housing, was then used in testing the remaining lubricants.

2. Maximum Possible Temperatures – To determine the effect of higher temperatures on the performance of the various lubricant types, a series of tests was made where no cooling other than natural convection was used. It is realized that, in these tests, some of the temperatures (varying from 243 to 342 deg. fahr.) exceed any possible service condition.

C – Speeds

Tests were made at 25, 50, and 70 m.p.h. in this work – speeds which lie well within the range of those obtained in ordinary passenger-car service.

Laboratory Test Methods and Results

Continuous-Load, Double-Axle Tests – Four series of tests (Series I, II, III, and IV) were performed using the double-axle set-up shown diagrammatically in Fig. 4. Two 1937 Chevrolet axles, having a ratio of 4.22:1, were directly connected to form a closed circuit between two electric dynamometers. The adjacent wheel flanges on the two axles were connected by means of a flexible coupling so the drive shaft rotation was transmitted directly from one unit to the other. As in the Gleason test, the differential action in the axles was made inoperative by welding the differential side gears. Each driveshaft was connected through a standard universal joint and transmission gear case to a 150-hp. electric dynamometer, as shown in Fig. 5. The transmission cases, from which the gears were removed, served only as supports for the universal housings. Driveshafts were inclined at the angle used in the regular chassis installation.

Before each test, the axle housings were cleaned carefully,

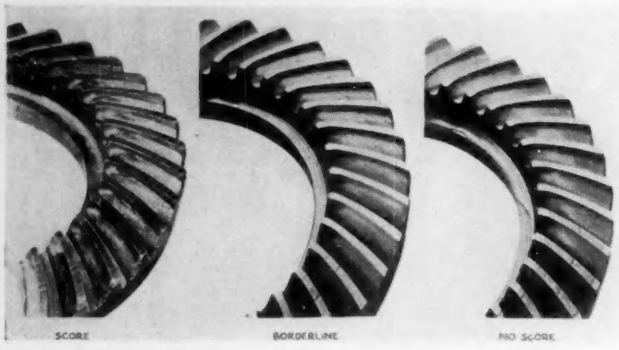


Fig. 11 – Representative Gear Teeth Rated "Score," "No Score," and "Borderline"

and ring- and pinion-gear clearances were held to the manufacturers' specifications of 0.004 to 0.008 in., as shown in Fig. 6.

Four copper-constantan thermocouples were installed in each axle assembly, one in the lubricant just under the pinion gear, one in each of the front and rear pinion-shaft bearing housings, and one just outside the axle to determine air temperature. A recording thermometer also was used to obtain a continuous temperature record of lubricant temperature during each run. This continuous temperature record was of great value in indicating similar mechanical conditions of gears, and also of lubricant performance, in check tests.

Drain holes drilled in the driveshaft housings, to eliminate any possibility of the universal lubricant getting back into the axle, and the welded differential side gears, were the only deviations from standard service equipment.

New ring and pinion gears were used for each complete test, after careful cleaning and inspection for surface finish. One feature of this work has been the ability to duplicate test results, indicating a remarkable uniformity and excellence in gear construction and finish. Bearings were inspected and, when necessary, replaced to insure against possibility of misalignment of pinion and ring gears.

In each test on the double-axle set-up, both axle housings were filled with the same lubricant. Load was applied to the drive side of the gear teeth in one axle for the first 20 hr. of test while, in the other axle, the coast side of the gear teeth was subjected to the same load. At the end of the 20-hr. period, the lubricant was drained from both axles and clearances were taken. Fresh lubricant of the same kind was then added, and the test was repeated by applying the load in the reverse direction so that the axle which had previously been driving was changed to coast operation. In this way, check determinations were obtained for both the drive and coast sides of the gears on two separate tests with each lubricant.

Cooling Method

When constant cooling was applied, a water jacket, made by welding together two back coverplates separated by a spacer ring, was used. Water at 75 deg. fahr., from a constant-level tank with a thermostat control, was circulated at a constant rate through this water jacket. The set-up for the runs made with constant cooling is shown in Fig. 7, and the construction of the water jacket in Fig. 8.

TEST SERIES I
NORMAL TEMPERATURES, HIGH CONTINUOUS LOAD, MODERATE SPEED

Procedure – At the start of each test, a 10-min. break-in at 25 m.p.h. and road torque was used. Speed and load were then built up at a uniform rate until the test conditions shown in Fig. 12 were obtained. These conditions were controlled closely during each 20-hr. run.

At the end of the first 20 hr. the lubricant was removed from the axles, clearances measured, and fresh lubricant added to each axle. The test was then repeated for 20 hr. driving under reversed load conditions.

Continuous lubricant-temperature charts were compared to check mechanical conditions on the two tests and two sets of gears. Typical temperature charts are shown in Fig. 9, representing duplicate runs on the same lubricant, and also charts obtained with three different lubricants in the same test are shown in Fig. 10.

Gears were inspected carefully for scoring and wear after removal from the axles at the conclusion of the two 20-hr. runs. Examination of gears, in order to classify them as to "score," "no score," and "borderline," was performed independently by three well-qualified observers from outside the test personnel. The ratings of all observers agree very closely.

Fig. 11 shows representative gear teeth, rated "score," "no score," and "borderline." The percentage of iron in the lubricant samples obtained after each 20-hr. run was determined.

Test Conditions – For exact test conditions see Fig. 12.

Summary.

1. The test results and rating of the lubricants by this test are shown in Fig. 13.

2. The conditions used in this test are remotely possible in service, as to load and speed, but hardly as to duration.

3. The utter inadequacy of the straight mineral oil under these conditions is shown clearly in the test results and in Fig. 14.

4. In all of these tests it appeared that the behavior of the lubricant in the first two hours of the test gave a good indication of its subsequent performance.

TEST SERIES II
HIGH TEMPERATURES, MODERATE LOAD, MODERATE SPEED

Procedure – In this series, the standard coverplate was used on the axles and the only cooling obtained was the natural convection as shown in Fig. 5; otherwise the set-up and procedure were identical with those described for Test Series I. Since the temperatures obtained in these tests were excessive, thus reducing lubricant efficiency, it was necessary to use a reduced load equivalent to road torque at 50 m.p.h.

Test Conditions – For exact test conditions see Fig. 15.

Summary.

1. The test results and rating of the lubricants are shown in Fig. 16.

2. It is hardly conceivable that temperatures as high as those obtained in this test would occur in any normal service. The

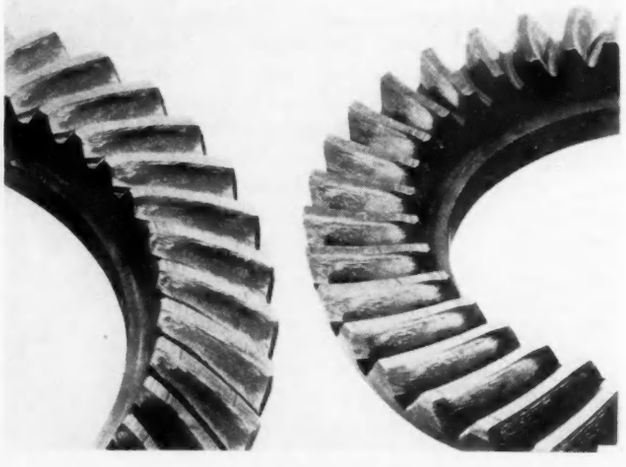


Fig. 14 – Scored Gears Caused by Mineral Oil in Test Series I

data, however, give an interesting comparison of lubricants under these excessive conditions.

3. Again the straight mineral oil failed completely to lubricate either the drive or coast side of the gears.

TEST SERIES III
VERY HIGH LOAD, LOW SPEED, NORMAL TEMPERATURES

Procedure – Constant cooling was used in this series, and the load corresponded to full engine torque in second gear at 25 m.p.h. These tests were only continued for 6 hr. direct and 6 hr. reversed.

All other conditions and procedure were the same as in Test Series I.

Test Conditions – For exact test conditions see Fig. 17.

Summary.

1. The test results and ratings of the lubricants are shown in Fig. 18.

2. Again the straight mineral oil failed completely.

SERIES I

TEST CONDITIONS

HIGH LOAD — MODERATE SPEED — NORMAL TEMPERATURE

LOAD:	150 LB. FT., FULL ENGINE TORQUE AT 50 M.P.H.
SPEED:	50 MILES PER HOUR, 2640 R.P.M. OF DRIVESHAFT
COOLING:	20 LBS. OF WATER PER MINUTE (75° F. INLET) CIRCULATED THRU COVER PLATE JACKETS
DURATION:	40 HOURS, 20 HOURS IN DIRECT DRIVE AND 20 HOURS IN REVERSE DRIVE
TEST PARTS:	NEW GEARS AND BEARINGS FOR EACH TEST, PINION - RING GEAR CLEARANCE .004 TO .008 INCHES
LUBRICANT:	SAME TYPE OF LUBRICANT FOR COMPLETE 40 HOUR TEST, NEW FILLING EACH 20 HOURS.

Fig. 12

SERIES I

TEST RESULTS

HIGH LOAD — MODERATE SPEED — NORMAL TEMPERATURE

LUBRICANT	MAX. TEMP. ° F. Drive	Coast	AVE. TEMP. ° F. Drive	Coast	DRIVE SIDE WEAR % FE.	RATING	COAST SIDE WEAR % FE.	RATING	GENERAL RATING
1. LEAD SOAP ACTIVE SULFUR	214	221	200	210	.025	NO SCORE	.037	BORDERLINE	BORDERLINE
2. LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	200	213	199	207	.023	NO SCORE	.046	SCORE	SCORE
3. INACTIVE SULFUR SAPONIFIABLE OIL	235	245	225	229	.34	NO SCORE	.220	BORDERLINE	BORDERLINE Excessive Wear Excessive Temp.
4. SULFUR CHLORIDE SAPONIFIABLE OIL	204	213	180	190	.038	NO SCORE	.046	NO SCORE	NO SCORE
5. CHLORINE COMPOUND	200	210	181	192	.024	NO SCORE	.027	SCORE	SCORE
6. CHLORINE COMPOUND SULFUR COMPOUND	212	226	179	199	.015	NO SCORE	.016	NO SCORE	NO SCORE
7. MINERAL OIL	256	274	238	267	.077	SCORE	.074	SCORE	SCORE

Fig. 13

SERIES II

TEST CONDITIONS

HIGH TEMPERATURE — MODERATE LOAD — MODERATE SPEED

LOAD:	43 Lb. FT., ROAD TORQUE AT 50 M.P.H.
SPEED:	50 MILES PER HOUR, 2640 R.P.M. OF DRIVESHAFT
COOLING:	NATURAL CONVECTION TO ROOM AT 70° F.
DURATION:	40 HOURS, 20 HOURS IN DIRECT DRIVE AND 20 HOURS IN REVERSE DRIVE
TEST PARTS:	NEW GEARS AND BEARINGS FOR EACH TEST, PINION - RING GEAR CLEARANCE .004 TO .008 INCHES
LUBRICANT:	SAME TYPE OF LUBRICANT FOR COMPLETE 40 HOUR TEST, NEW FILLING EACH 20 HOURS.

Fig. 15

SERIES II

TEST RESULTS

HIGH TEMPERATURE — MODERATE LOAD — MODERATE SPEED

LUBRICANT	MAX. TEMP. ° F. Drive	Coast	AVE. TEMP. ° F. Drive	Coast	DRIVE SIDE WEAR % FE.	RATING	COAST SIDE WEAR % FE.	RATING	GENERAL RATING
1. LEAD SOAP ACTIVE SULFUR	275	293	243	261	.036	NO SCORE	.039	BORDERLINE	BORDERLINE
2. LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	273	289	241	256	.015	NO SCORE	.013	BORDERLINE	BORDERLINE
3. INACTIVE SULFUR SAPONIFIABLE OIL	316	322	252	262	.48	NO SCORE	.74	SCORE	SCORE Excessive Temp.
4. SULFUR CHLORIDE SAPONIFIABLE OIL	246	260	214	214	.035	NO SCORE	.022	NO SCORE	NO SCORE
5. CHLORINE COMPOUND	243	247	213	221	.035	NO SCORE	.034	BORDERLINE	BORDERLINE
6. CHLORINE COMPOUND SULFUR COMPOUND	258	269	210	217	.019	NO SCORE	.022	NO SCORE	NO SCORE
7. MINERAL OIL	294	342	265	314	.093	SCORE	.136	SCORE	SCORE

Fig. 16

SERIES III

TEST CONDITIONS

VERY HIGH LOAD — LOW SPEED — NORMAL TEMPERATURE

LOAD:	285 LB. FT., FULL ENGINE TORQUE IN SECOND GEAR
SPEED:	25 MILES PER HOUR, 1320 R.P.M. OF DRIVESHAFT
COOLING:	20 LBS. OF WATER PER MINUTE (75° F. INLET) CIRCULATED THRU COVER PLATE JACKETS
DURATION:	12 HOURS, 6 HOURS IN DIRECT DRIVE AND 6 HOURS IN REVERSE DRIVE
TEST PARTS:	NEW GEARS AND BEARINGS FOR EACH TEST, PINION - RING GEAR CLEARANCE .004 TO .008 INCHES
LUBRICANT:	SAME TYPE OF LUBRICANT FOR COMPLETE 12 HOUR TEST, NEW FILLING EACH 6 HOURS

Fig. 17

SERIES III

TEST RESULTS

VERY HIGH LOAD — LOW SPEED — NORMAL TEMPERATURE

LUBRICANT	MAX. TEMP. ° F. Drive	Coast	AVE. TEMP. ° F. Drive	Coast	DRIVE SIDE WEAR % FE.	RATING	COAST SIDE WEAR % FE.	RATING	GENERAL RATING
1. LEAD SOAP ACTIVE SULFUR	217	233	217	232	.053	NO SCORE	.055	BORDERLINE	BORDERLINE
2. LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	213	223	207	220	.054	NO SCORE	.051	SCORE	SCORE
3. INACTIVE SULFUR SAPONIFIABLE OIL	254	271	252	268	.022	NO SCORE	.035	ERRATIC	ERRATIC Excessive Temp.
4. SULFUR CHLORIDE SAPONIFIABLE OIL	226	235	209	219	.030	NO SCORE	.031	NO SCORE	NO SCORE
5. CHLORINE COMPOUND	212	222	206	218	.024	NO SCORE	.022	BORDERLINE	BORDERLINE
6. CHLORINE COMPOUND SULFUR COMPOUND	223	229	201	215	.024	NO SCORE	.034	BORDERLINE	BORDERLINE
7. MINERAL OIL	278	292	270	292	.20	SCORE	.20	SCORE	SCORE

Fig. 18

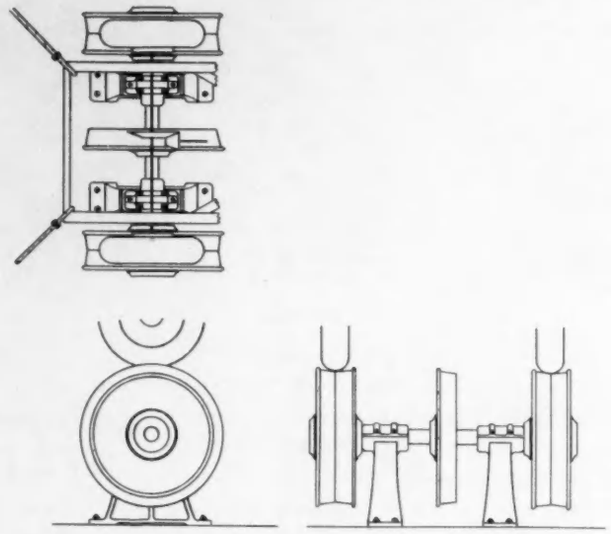


Fig. 19 – Hypoid Rear-Axle Shock-Test Equipment

TEST SERIES IV
HIGH SPEED, HIGH LOAD, NORMAL TEMPERATURES

Procedure – Two lubricants only were tested under these conditions to determine if higher speeds would modify the performance obtained at the lower speeds. Set-up and conditions were similar to those in Series I except that a speed of 70 m.p.h. and full engine torque (140 ft-lb.) were used. The tests were continued for 5 hr. direct and 5 hr. reversed for each lubricant.

Summary – Results indicated substantial agreement with the performance of the same lubricants at the lower speeds and, therefore, the other lubricants were not studied.

Shock-Load Tests

Laboratory Shock-Test Procedure – Although, so far as we are aware, shock-loading tests previously had been made only on the road or proving ground, it was decided, both as a safety measure and to be more certain of reproducibility of test conditions, to develop a laboratory test procedure employing equipment and methods similar to those employed in proving-ground tests.

This laboratory test was patterned closely after the General Motors procedure. A new 1937 Chevrolet sedan was set up in the laboratory with the front wheels fastened to the laboratory floor and with each rear wheel resting on the circumference of two balanced 32-in. cast-iron freight-car wheels mounted on a 4-in. shaft as shown in Figs. 19, 20, and 21. This car-wheel assembly on the common shaft, shown in Fig. 22, was calculated to supply 525,000 ft-lb. of kinetic energy when rotating at a circumferential speed of 70 m.p.h. This energy corresponds to that of a Chevrolet car traveling on the road at the same speed. The drive, to and from the car, is by tire contact on the surface of the car-wheels through the transmission, driveshaft and clutch to the engine. The engine is used to bring the car-wheel assembly up to speed and then the energy of the car-wheel unit is used to drive the engine in a manner similar to coasting on the road.

A strap type of mechanical brake was installed on one of the car-wheels to produce road torque during the break-in part of each test. Speedometer cable, ignition, throttle and clutch controls were extended to a station outside the car for safety, and a mechanical tripping device employed for engaging the clutch for shocking the gears. This rapid engagement of the clutch was felt to be more reproducible than the sudden release of the operator's foot from the clutch pedal.

Thermocouples were installed in the cooling-water jacket, in the crankcase oil, and in the rear-axle lubricant. To facilitate inspection of the gears between application of shocks, a section of 4-in. pipe coupling fitted with a removable plug was welded into the rear axle coverplate.

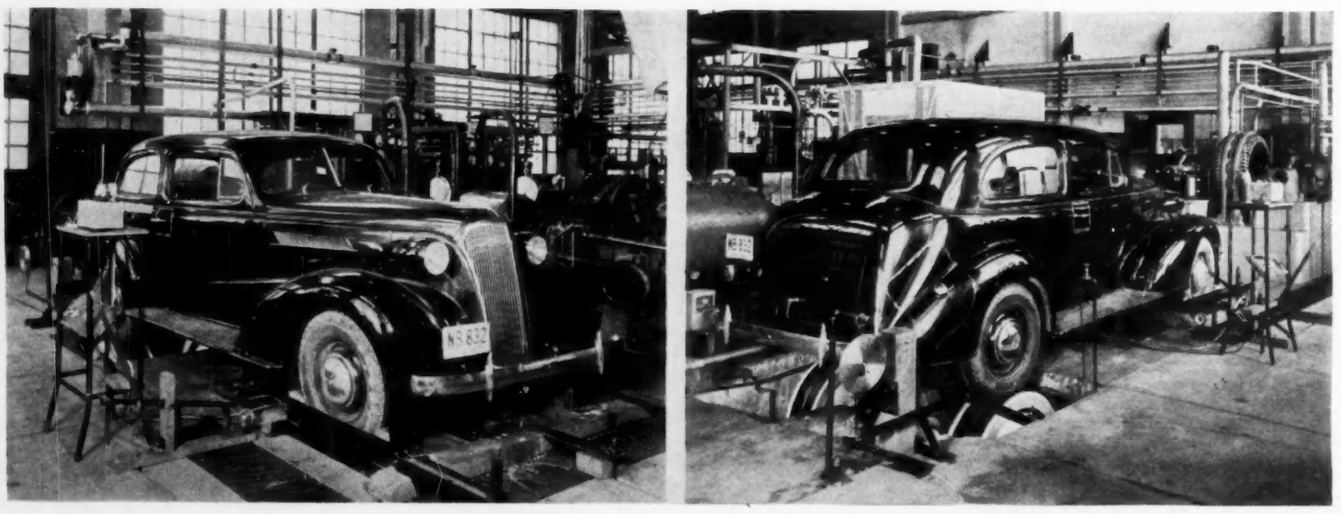
Tire pressures were maintained constant at 28 lb. per sq. in. The temperature of the hypoid lubricant was not controlled but, during the shocks, it was between 185 and 195 deg. fahr. Crankcase oil temperatures were maintained at 180 to 190 deg. fahr., in order that engine friction would be nearly constant. Clutch plates and linings were changed at about every tenth test and were noted to be in good condition when removed. There was no noticeable change in the condition of the tire treads during the series of tests.

Test Conditions – are shown in Fig. 23 which describes the entire Laboratory Shock Test Procedure.

Discussion of Results.

1. This test, which closely follows General Motors Proving Ground procedure, applies shock loads which do not permanently deform axle parts.

2. The shock loading used in this test appears to be slightly more severe, and probably more reproducible, than that applied in proving-ground tests.



Figs. 20 and 21 – Laboratory Shock-Test Set-Up

3. The possibility of shock loads of this magnitude and extent occurring in service is remote but possible.

4. Failure usually occurs either before any shock loads are applied or in the first few shocks.

5. This test clearly differentiates between a lubricant which will pass and one that fails.

6. Of the commercial lubricants examined, those which prevented scoring on the shock test were quite satisfactory in all tests made at reasonable temperatures, using uniform loading. Several experimental lubricants, however, passed the shock test and failed under some of the continuous-load tests.

7. The ratings of the lubricants in this test are shown in Fig. 24.

Road Shock Tests

To ascertain if the laboratory shock procedure would give results comparable with those obtained in road shock tests, the standard Packard Proving Ground shock test was performed on the various lubricants.

A new Packard "120" Coupe was given the standard break-in of 500 miles. New third members were installed for each test and checked for clearance before use. The only deviation from Packard test procedure was the use of an 11-mile drive to the test road in place of 7 miles given in their method.

In the Packard procedure a series of shock loads is applied in second and first gear, and scoring is determined by noise and by a series of inspections during and at the end of the test.

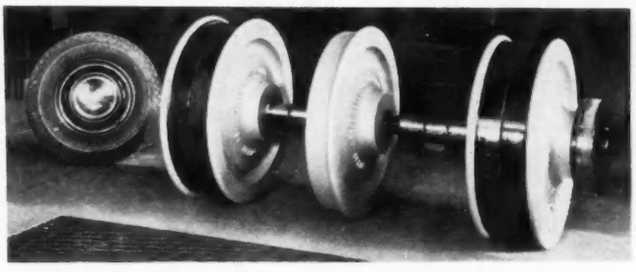


Fig. 22 – Car-Wheel Assembly

Discussion of Results.

1. The ratings on the various lubricants obtained by the Packard shock tests were similar to those obtained in the laboratory procedure on the Chevrolet. These ratings are shown in Fig. 25.

2. It is obvious from the preceding that road shock tests and the laboratory procedure are of about the same order of intensity.

LABORATORY SHOCK TEST

TEST PROCEDURE

1. Install new ring and pinion gear to factory specifications. Check runout and clearance at eight positions of ring gear. Fill axle housing with lubricant to be tested.
2. Run engine with transmission in neutral for sufficient time to bring water and oil temperatures to 160° F. as indicated by the thermocouples.
3. Start wheels rolling in low gear, shift thru second and into high. Bring up to 60 MPH, and run for 4 miles, having adjusted brake load to road torque conditions of 14 lbs.
4. Accelerate to 70 MPH, and run at this speed for 4 miles, with a brake load of 18 lbs.
5. Disengage clutch, and test for noise. Inspect gears.
6. Accelerate to 65 MPH and run for 35 minutes at road torque.
7. Take noise meter readings, and inspect.
8. With no brake load, accelerate in high gear to 80 MPH. Declutch close throttle, and cut ignition. As speed passes thru 70 MPH, release clutch suddenly, and permit speed to fall to 40 MPH. Close ignition switch, and shift to neutral. Inspect gears. Repeat this procedure for a total of seven times.
9. Check for noise, and repeat an additional 8 times, for a total of fifteen times. After last shock, make a check for noise.
10. In second gear, accelerate to 52 MPH. Disengage clutch, cut ignition, and close throttle. As speed passes thru 45 MPH, engage clutch suddenly. Permit speed to drop to 20 MPH, and close ignition switch. Inspect gears for score marks. Repeat this procedure for a total of 5 times, and check noise.
11. Repeat this procedure for a total of ten times, and make a noise check.
12. Stop engine and drain lubricant for analysis. Take runout and clearance readings of ring gear.

Fig. 23

Observations on Stability

Although stability was not a primary purpose of this study, during the gear test program certain observations were made having a bearing on stability and these comments and observations are tabulated in the last column of Fig. 27. In service, such effects as noted in our tests of rather short duration might become of vital importance. With all other lubricant properties equal, it would even seem that one would be inclined toward using a lubricant with as high stability as possible. It should be noted that the period of operation of the continuous load test was too short to give a proper indication of the performance of the lubricants over extended periods of service.

Summary

1. A series of hypoid gear tests has been made on lubricants wherein each of a series of different operating conditions has been selected and, in turn, been made severe, while the other

LABORATORY SHOCK TEST RESULTS

CHEVROLET

	LUBRICANT TYPE	AREA SCORED %	RATING
1.	LEAD SOAP ACTIVE SULFUR	0	NO SCORE
2.	LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	46	SCORE
3.	INACTIVE SULFUR SAPONIFIABLE OIL	16	SCORE
4.	SULFUR CHLORIDE SAPONIFIABLE OIL	0	NO SCORE
5.	CHLORINE COMPOUND	61	SCORE
6.	CHLORINE COMPOUND SULFUR COMPOUND	0	NO SCORE
7.	MINERAL OIL	88	SCORE

Fig. 24

ROAD SHOCK TEST RESULTS

PACKARD

	LUBRICANT TYPE	AREA SCORED %	RATING
1.	LEAD SOAP ACTIVE SULFUR	0	NO SCORE
2.	LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	84	SCORE
3.	INACTIVE SULFUR SAPONIFIABLE OIL	67	SCORE
4.	SULFUR CHLORIDE SAPONIFIABLE OIL	--	ERRATIC
5.	CHLORINE COMPOUND	53	SCORE
6.	CHLORINE COMPOUND SULFUR COMPOUND	0	NO SCORE
7.	MINERAL OIL	63	SCORE

Fig. 25

operating conditions have been maintained at moderate values.

These tests have been:

Test Series	Selected Condition
I	High Load
II	High Temperature
III	Very High Load, Low Speed
IV	High Speed
Laboratory shock test	Rate and magnitude of shock loads.
Packard shock test	Rate and magnitude of shock loads.

2. A laboratory shock-test procedure has been described for the testing of hypoid lubricants.

3. The results of these tests indicate deficiencies in some of the lubricants under certain of the extreme conditions employed. The ratings of the lubricants in the various tests, as well as Timken and S.A.E. film strengths, are shown in Figs. 26 and 27.

SUMMARY OF GEAR TEST RATINGS

LUBRICANT TYPE	CONTINUOUS LOAD SERIES I	CONTINUOUS LOAD SERIES II	CONTINUOUS LOAD SERIES III	SHOCK LOAD LABORATORY	SHOCK LOAD ROAD	GEAR TEST RATING
1. LEAD SOAP ACTIVE SULFUR	BORDERLINE	BORDERLINE	BORDERLINE	NO SCORE	NO SCORE	PASS
2. LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	SCORE	BORDERLINE	SCORE	SCORE	SCORE	FAIL
3. INACTIVE SULFUR SAPONIFIABLE OIL	BORDERLINE Excessive Wear Excessive Temp.	SCORE Excessive Temp.	ERRATIC Excessive Temp.	SCORE	SCORE	FAIL
4. SULFUR CHLORIDE SAPONIFIABLE OIL	NO SCORE	NO SCORE	NO SCORE	NO SCORE	ERRATIC	--
5. CHLORINE COMPOUND	SCORE	BORDERLINE	BORDERLINE	SCORE	SCORE	FAIL
6. CHLORINE COMPOUND SULFUR COMPOUND	NO SCORE	NO SCORE	BORDERLINE	NO SCORE	NO SCORE	PASS
7. MINERAL OIL	SCORE	SCORE	SCORE	SCORE	SCORE	FAIL

Fig. 26

4. Of the commercial lubricants studied, those that passed the laboratory shock test lubricate hypoids quite satisfactorily under practically all other test conditions where normal temperatures were used.

5. The performance of the lubricants in the gear tests made in this work, seems to bear but a slight relation to film-strength data as obtained on laboratory test machines. Some lubricants with moderate or low film strengths were satisfactory in all of the tests, and other low-film-strength lubricants failed in all tests.

GENERAL SUMMARY

LUBRICANT TYPE	FILM STRENGTH TIMKEN LBS.	FILM STRENGTH S. A. E. 1000 R.P.M. LBS.	GEAR TEST RATING	OBSERVATIONS CONTINUOUS LOAD TESTS
1. LEAD SOAP ACTIVE SULFUR	25	580+	PASS	Grey Deposit on Gear Teeth Dark Color on Contact Surfaces
2. LEAD SOAP INACTIVE SULFUR CHLORINE COMPOUND	68 ±	98 ±	FAIL	Light Colored Deposit in Center of Pinion Gear
3. INACTIVE SULFUR SAPONIFIABLE OIL	86	81	FAIL	No Deposit, No Stain on Gears
4. SULFUR CHLORIDE SAPONIFIABLE OIL	33	90	--	Yellow Stain on Housing and Gears
5. CHLORINE COMPOUND	77	305	FAIL	Occasional Stain on Gear Teeth
6. CHLORINE COMPOUND SULFUR COMPOUND	59	330	PASS	No Deposit, No Stain on Gears
7. MINERAL OIL	7	0	FAIL	No Deposit, No Stain on Gears

Fig. 27

For sponsoring this investigation, acknowledgment is made to The Lubri-Zol Corp.

Industrial Mobilization for War

MODERN armies subsist on industry and only because of industry. The increased mechanization and motorization and increase in effectiveness and capacity of weapons have made industrial mobilization the key to the successful prosecution of war.

What is more important, it is the key to not having a war. The nation that has an effective mobilization which can make munitions and war supplies in large quantities at the earliest possible date before and after the declaration of war is the nation that will probably never have the occasion to use its facilities and armies.

If you believe in peace as we all do, the best guarantee of peace is an effective industrial organization to back up the army.

Our government, under the direction of Col. Louis Johnson, the Assistant Secretary of War, has developed a vast organized plan of industrial mobilization.

Under his direction, some 10,000 firms have been allocated schedules of production. These schedules will require, in time of war, from 20 to 50 per cent of their capacity so as to leave ample capacity for civil markets.

The study of these plants under the direction of the organization of the Assistant Secretary of War is carried out through a number of Divisions and Boards, such as the Allocation Division, the Planning Division, the War Resources Board, the Industrial War College, and so on.

The types of contracts offered are these: a negotiated-price contract; an adjusted-compensation contract, and an evaluated-fee contract. These contracts are designed to insure a fair profit to the contractor and a protection to his civil business while, at the same time, securing for the government the highest quality of goods at the earliest possible moment.

The organization of the Assistant Secretary of War and the various Bureaus of the government is arranged so that plants are allocated to the Army and Navy, or divided between them. Thus the Bureaus of the government have plants that are adapted to their particular needs allocated to them, and then a tentative arrangement, such as an allocation of facilities and schedules of production, is entered into between the contractor and the Government.

Colonel Johnson has been advocating the placing of experimental orders in plants so that he can have what is known in England as "shadow plants," that is, small production units for experimental orders.

Congress is being asked to appropriate $2,000,000 for this purpose. Thus this procurement planning, this scheme of accustoming plants to war-time production and this mobilization of the economic resources of industry provide a systematic way for immediately putting plants upon war-time production.

In time of war, we have already provided the framework of Boards for controlling labor, prices, raw materials, and the like. We will not have to wait six months to a year to organize these Boards and get them functioning as we did during the last war with the General Munitions Board and the War Industries Board.

The cost in money and lives due to neglect of these principles is stupendous. The great cost of war is not in preparation for it, but in making up for lack of preparation after we have once entered it, and the strange thing is that, the greater the preparation, the less likelihood there is that we will ever have to use it.

Excerpts from the paper: "Industrial Mobilization and Its Contribution to the Making of War," by H. A. Toulmin, Jr., Toulmin and Toulmin, presented at the Dayton Section Meeting of the Society, Dayton, O., May 9, 1938.

Torsional Vibration of In-Line Aircraft Engines

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A DISCUSSION is presented of the two-node as well as the single-node characteristics for various crankshaft systems both with regard to frequency and to severity of vibration. The effect of frequency on harmonic torque input, which is a measure of vibration severity, also is discussed. The paper shows that, with a careful selection of the vibration frequency, the harmonic torque input for various vibration orders can be reduced considerably.

A method is presented for the selection rather than the determination of vibration frequencies. It shows that, by the careful selection of frequencies, certain harmonic orders both for single-node and two-node vibration can be minimized appreciably and/or eliminated from the normal operating range of the engine. The friction-type and the dynamic-pendulum-type vibration damper are discussed.

This paper brings out the fact that a definite restriction on vibration amplitude (degrees) cannot be used to limit the severity of torsional vibration. The allowable magnitude of vibration for safe operation depends upon the type of system being used and the relative specific power output of the engine. The stresses brought about by vibration must be considered before an intelligent restriction of vibration amplitude can be made.

THE continued refinement of aircraft towards the ultimate goal of a flying wing has resulted in the powerplant accounting for an increasing percentage of the total drag. It is natural, therefore, that interest in the in-line type of aircraft engine is increasing, particularly in the larger powers, on account of its low frontal area, inherent balance, adaptability to special low-drag installations, and the possibility of using a large number of cylinders in a single unit.

In-line engine development has, with the exception of the low-power field, resolved itself almost entirely around the use of a six-throw crankshaft. It is well known that this is the shortest type to give inherent smoothness due to balance of reciprocating and centrifugal forces and couples, suitable overlapping of power impulses, and ease of manifolding for good mixture distribution. The twelve-cylinder engine, on account of its low ratio of maximum to average torque, low specific weight due to efficient utilization of crankcase and crankshaft structure, and its useful shape in a fuselage or nacelle, has long been the favored model of the in-line type engine.

The existence of torsional vibration in six-throw crankshafts was recognized early due particularly to the custom of driving accessories from the free or anti-propeller end of the engine. Practically every step in the development of the modern aircraft powerplant has tended to make the torsional-vibration problem more severe until it is now engineering suicide to design a new engine without predetermining the torsional characteristics and providing for essential damping. In the old days of wood propellers, low specific power outputs and direct-drive engines, speed ranges existed free from resonant vibration for all normal operation. Very reliable engines could be built without any provision for torsional damping by using these speed ranges or even by strengthening parts sufficiently to operate in some of the lesser periods. The improvements in propellers going from wood to fixed-pitch metal, then to two-position and finally to the completely variable-pitch propeller with gear-reduction drive, have spread greatly the full-throttle operating speed range of the engine. The addition of take-off rating and the economy of low-speed full-throttle cruising have contributed further to this spread. Add a dive-test requirement to these items, and the modern engine should have satisfactory torsional characteristics *at full throttle* anywhere from 1000 r.p.m. to 3500 r.p.m. or more. With resonant harmonics on a four-stroke-cycle engine possible at ½, 1, 1½, 2, 2½, . . . and so on, orders of the fundamental frequency, the position of the designer becomes none too happy. Add to this situation the fact that each improvement in fuel results in higher compression ratio and/or mean effective pressure, both of which add to the vibration amplitude and the increase of which have been largely responsible for bringing two-noded vibration into the picture, and the problem becomes fairly complete. On the favorable side it is fortunate that several of the harmonic orders cancel out, that the exciting forces are lower at low engine speed, that considerable damping exists in the engine itself, that certain favorable frequency ranges can be designed into an engine without appreciable cost in weight, and that torsional-vibration characteristics can now be predetermined with reasonable accuracy in regard to both frequency and severity.

Both single- and two-node vibration occur in high-output in-line engines, whereas the radial engine usually encounters only the single-node type. The single-node vibration exists adjacent to the propeller and usually evidences itself at the

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., Mar. 10, 1938.]

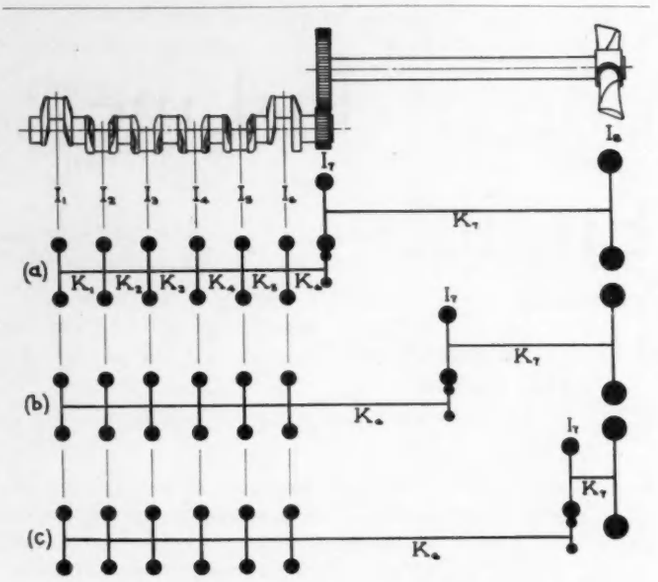


Fig. 1 – Three Types of Equivalent Crankshaft Systems with Reduction Gear – The Type Shown in Fig. 1 (*a*) Is Used in the Standard Allison V-1710 Engine

propeller cone or hub where galling occurs or in the accessories and drives when they are taken from the anti-propeller end of the shaft. Propeller troubles often are traced to excessive vibration of this type. Two-node vibration also has a node adjacent to the propeller and, in addition, another not far from the center of the crankshaft, the location of the latter being affected appreciably by the location and mass of the reduction gear in the torsional system. Two-node vibration stresses the crankshaft highly at the node, and excessive amplitude is evidenced chiefly in crankshaft failure. Since relatively small amplitudes result in comparatively high crankshaft stresses, it is essential that careful calculations and analysis of torsiometer data from a reliable instrument be made of any new or changed engine design. The two-node vibration of in-line engines is very similar both as to frequency and effects to the single-node vibration on radial engines, except that it is less apt to cause propeller troubles.

Natural Vibration Frequencies

The calculation of natural vibration frequencies in aircraft engines has been discussed previously in other papers and text books. This paper presents the results of several studies and engine tests rather than the methods of calculation. However, it might be of interest to mention several of the methods that are used. The Holzer method sometimes known as the Gumbel method[1] is a "cut-and-try" procedure that is worked out in tabular form. The Lewis method[2], which is also a "cut-and-try" procedure is used commonly. The Tuplin method[3] solves the natural frequencies by direct procedure so that it is not necessary to use "cut-and-try" operations. However, this last procedure is rather complicated, and it is questionable whether it has any time-saving advantages over the two preceding "cut-and-try" procedures. The best method to use generally is determined by the individual preference of the engineer making the analysis.

In the solution of natural frequencies of in-line engines, no abridged procedures such as considering the crankshaft assembly and propeller as a simple two-mass system can be used. There are two reasons for this condition: (1) it is not possible to calculate the two-node frequencies and (2) the relative severity of the various harmonic orders cannot be ascertained. This latter item is very important since, by calculating the harmonic torque input of the various vibration orders, the relative severity of vibration can be determined. Some writers have advocated the use of empirical formulas in order to simplify the natural frequency calculations. However, in these modern times when results should be within ±10 per cent, empirical formulas are not accurate enough for practical use.

The range of natural frequencies encountered in in-line engines varies considerably due to the use of different gear ratios and shaft rigidities. In general, it may be said that single-node frequencies vary between 30 and 160 cycles per sec. Two-node frequencies usually exist between 160 and 350 cycles per sec.

A diagrammatical sketch showing three designs of crankshaft torsional equivalent systems is given in Fig. 1. These diagrams show the distribution of the masses and rigidities that occur throughout the system and which are used in calculating the natural vibration frequencies. Fig. 1 (*a*) is used on the standard Allison V-1710 engine. With this system the reduction gear (I_7) is located relatively close to the crankshaft (I_1 to I_6), that is, considerable flexibility is provided between the reduction gear and the propeller hub. This flexibility is readily obtained when a long nose is provided for aerodynamic reasons, as on the Allison Engine shown in Fig. 2, and when a large gear ratio is employed (since the rigidity of the propeller shaft varies inversely with the square of the reduction-gear ratio). Fig. 1 (*b*) shows the reduction gear approximately half way between the crankshaft and the propeller (I_8). In this case the reduction gear is at

[1] See "Mechanical Vibrations," by J. P. Den Hartog, McGraw-Hill Book Co., 1934.

[2] See Transactions of the Society of Naval Architects and Marine Engineers, Vol. 33, 1925, pp. 109-140: "Torsional Vibration in the Diesel Engine," by F. M. Lewis.

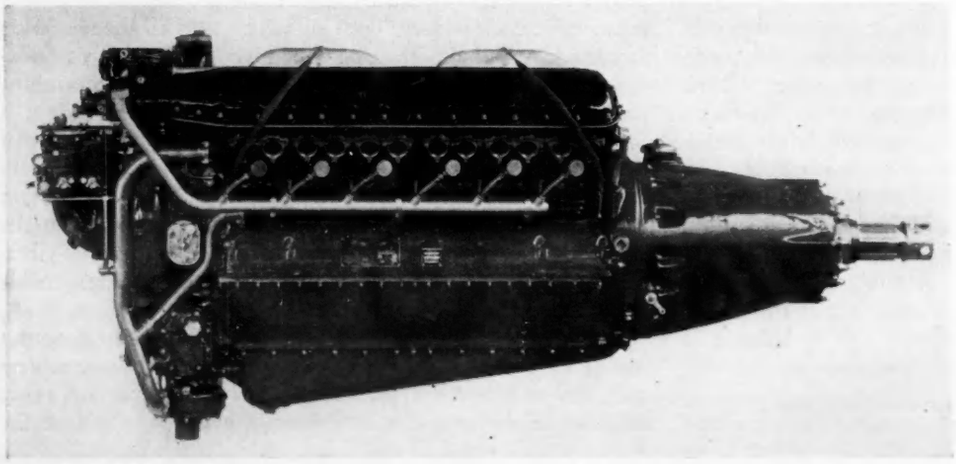


Fig. 2 – Side View of Allison Model V-1710-C (Standard) Engine

the anti-node or point of maximum amplitude with two-node vibration. Changing the location of the reduction gear consequently results in a change of the pinion-shaft rigidity (K_6) and the propeller-shaft rigidity (K_7) for a constant frequency. In Fig. 1 (*c*) the reduction gear is shown rather close to the propeller with the flexibility between reduction gear and crankshaft. This type of system is used for short-nose engines such as the Rolls Royce and Ranger geared engines. All three of the crankshaft systems shown in Fig. 1 result in different torsional-vibration characteristics both with regard to frequency and to harmonic torque input.

Harmonic Torque Input

The relative severity of torsional vibration can be expressed by a term known as the harmonic torque input. This factor, which is measured in inch-pounds, is a numerical indication of the amount of vibration energy induced in or acting on the torsional system. This term is also known by the total effective torque or the work input per cycle. In general, the vibration amplitude is proportional to the harmonic torque input, that is, in a given engine the greater the harmonic torque input the greater the vibration amplitude of the crankshaft.

The amplitude produced by a given harmonic torque input can be calculated only by making a number of assumptions. It is therefore believed to be a waste of time to attempt to calculate the torsional vibration amplitude of an engine for various harmonic orders. The amount of torque input that can be induced in a crankshaft system without severe vibration can be determined best from practical experience. Theoretically the vibration of a crankshaft at a resonant speed would be infinite if no inherent damping existed in the engine. However, there is always some damping present which will absorb a certain amount of harmonic torque input and prevent excessive vibration. However, if the torque input is too great, the inherent damping of the engine will not be adequate and either severe vibrations will be encountered or vibration dampers will have to be used.

In one engine with which the authors have had considerable experience, it has been found that a torque input of approximately 1000 in-lb. or 0.59 in-lb. per cu. in. piston displacement produced either a single- or two-node vibration of serious amplitude. However, this amount of torque input is not an indication of serious vibration in all internal-combustion engines. In large engines or in engines where a greater amount of inherent damping is present, it would take more harmonic torque input to produce serious vibrations.

In the investigation of torsional characteristics of an engine, the harmonic torque inputs are calculated for each critical speed. No attempt will be made in this paper to explain the procedure of calculation. The method generally is included in most books on torsional vibration [1 and 3]. By comparing the inputs with each other the most serious critical speeds can be determined.

Calculations and actual test experience show that the harmonic torque input for a given engine will vary with the natural frequency of the crankshaft system, that is, for a low frequency the harmonic torque input will be different than for a higher frequency with the same vibration order. Fig. 3 shows the harmonic torque input for various vibration orders plotted versus frequency for both single-node and two-node vibration of a twelve-cylinder engine. The curves show that the torque input for single-node frequencies increases rapidly with an increase in frequency. The 1½ order is the most severe. (On a six-cylinder engine the third order is the most troublesome). If a high-frequency crankshaft system is desired, it is necessary to make sure that the serious resonant speeds come in far above the engine operating range in order to eliminate serious vibration trouble. This precaution is essential because the value of the harmonic torque input is also a measure of the speed range over which an order is noticeable. For torque inputs of 5000 in-lb. in an engine of 1710 cu. in. piston displacement, undamped torsional vibration is severe over a speed range of approximately 400 r.p.m.

The characteristics of harmonic torque input for two-node vibration in a twelve-cylinder engine are also shown in Fig. 3. These characteristics are considerably different than those for single-node frequencies. The 4½ and 7½ orders show very little change in harmonic input with frequency. The 6th order shows both increasing and decreasing characteristics with an increase in frequency and is eliminated when the node exists in the center of the crankshaft. This fact shows that 6th-order two-node vibration can be eliminated by designing the crankshaft system so that the node exists close to the center of the crankshaft.

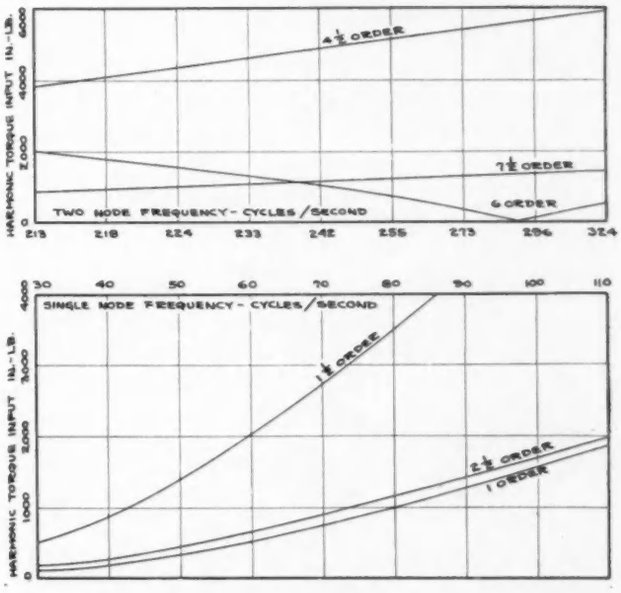


Fig. 3 – Harmonic Torque Input Characteristics Plotted Vs. Frequency for Both Single-Node and Two-Node Vibration – Only a Few of the Most Important Harmonic Orders Are Shown

The value of harmonic torque input also is affected appreciably by the following factors: indicated mean effective pressure, compression ratio, firing order, and reciprocating masses. In general, anything that affects the shape of the indicator card also affects at least one of the harmonic orders. Fig. 4 shows the variation of harmonic torque input with indicated mean effective pressure. Fig. 5 gives similar data for the effect of compression ratio. Both sets of data were calculated for a twelve-cylinder engine and are fairly well confirmed by test results.

Selection of Vibration Frequencies

In the design of modern aircraft engines, the torsional vibration frequencies generally are selected or controlled by predetermining the constants of the equivalent system. This practice differs from that employed several years ago when in-line engines often were designed with little regard for torsional frequencies. After the engine was built, the frequencies generally were checked with a torsiometer. Using this procedure, it was not uncommon to find that the most severe critical periods were in the desired operating range, even with a fixed-pitch propeller.

[3] See "Torsional Vibrations," by W. A. Tuplin, John Wiley & Sons, 1934.

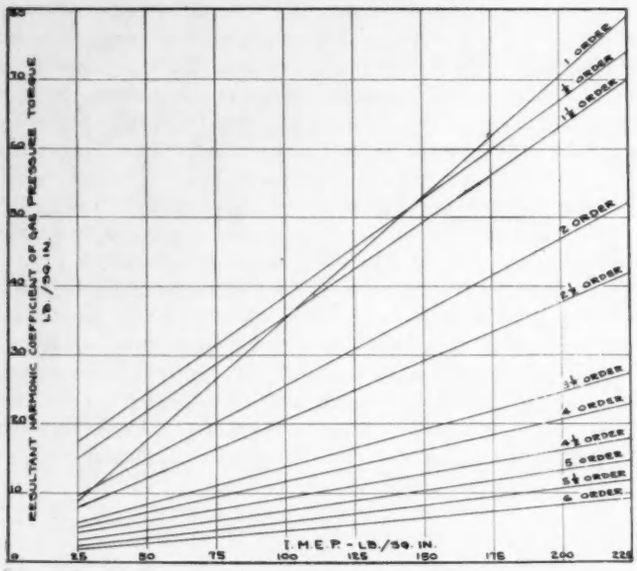


Fig. 4 – Resultant Harmonic Coefficients of the Single-Cylinder Gas Pressure Torque Curve Vs. I.M.E.P.

In the original design of an engine, the crankshaft and attached masses are not changeable through a wide range on account of other than torsional vibration considerations, such as cylinder size and spacing, bearing loads, and so on. In a new design considerable latitude can be provided for the selection of the pinion-shaft or propeller-shaft rigidities. These torsional constants are shown by K_6 and K_7 in Fig. 1. By the suitable selection of these rigidities it is possible to minimize the number of harmonics that come within the normal operating range of the engine. In general, the most severe harmonics can be brought outside the normal operating range of the engine so that only a few minor vibration periods are left which can be damped. The relation of these constants to each other and also the reduction gear mass (I_7) determine the relation of the single-node frequency to the two-node frequency.

The method used to make an analysis of the frequency characteristics of an engine is carried out conveniently by the chart shown in Fig. 6. In this chart the propeller shaft rigidity (K_7) was varied over a wide range which resulted in the frequencies shown by the abscissa of the graph. Both the single-node and two-node frequencies are shown. The dotted lines indicate the vibration orders that will cancel out and will not be encountered in a twelve-cylinder 60-deg. vee engine. The heavy solid lines represent those orders which are normally severe enough to require attention on a twelve-cylinder engine, whereas the light lines show the orders which can be neglected. The severity of these vibration orders was determined by the harmonic torque input and verified by actual tests. The frequency selected for the crankshaft system should be such that the most severe vibration orders will come outside the normal operating speed range of the engine. By drawing a vertical line through any frequency the harmonic orders which occur in any selected engine operating speed range may be determined readily.

Take the 1½ order for discussion. If the engine has a single-node natural frequency of 75 cycles per sec., this order will be most severe or will peak at 3000 r.p.m. At 60 cycles per sec., it will peak at 2400 r.p.m. and is still within the normal operating range of the engine. Consequently, in order to remove this vibration order from the operating range, it is necessary to employ a single-node natural frequency of 90 cycles per sec. or more, or a single-node frequency of 35 cycles per sec. or less. In the first case, the vibration order will occur above the normal operating range of the engine and, in the second case, it will exist below the cruising speeds of the engine. In an extension shaft system where it is possible to get a small amount of rigidity without involving high stresses, the 35-cycle-per-sec. system could be employed. However, in the event that a short compact reduction-gear housing is desired on an engine, it would be preferable to use a system with a single-node frequency of 90 cycles per sec. or more.

As has been said before, it is necessary to consider the two-node frequency as well as the single-node frequency in an in-line type engine. With a stiff or 90-cycle-per-sec. system as mentioned previously and with the particular one-node to two-node relation shown on Fig. 6, it will be observed that only the 7½ order two-node and 6 order two-node vibrations will exist in the operating range. However, with a flexible system of 35 cycles per sec., the 4½ order two-node critical speed comes within the normal operating range. Further examination of the curves in Fig. 6 shows that there is no frequency in which all vibration orders can be eliminated, consequently, some damping is required if the specific output of the engine is high.

In contrast to Fig. 6 the curves in Fig. 7 are drawn which show the frequency characteristics for a system with variable pinion-shaft rigidities (K_6). In some engine designs this shaft section is controlled easily as when an external gear box is employed. Probably the biggest differences to be observed in comparison with Fig. 6 is the fact that this variation of rigidity has very little effect on the two-node frequencies. Figs. 6 and 7, therefore, reveal two facts regarding in-line engines: the variation of K_6 maintains the two-node frequency practically constant but changes the single-node frequency and the variation of K_7 changes the two-node frequency as well as the single-node frequency. These facts suggest that, with the proper selection of K_6 and K_7 that most of the critical speeds can be removed from the operating range of the engine. It is obviously a laborious process to determine the effect of all the variables in a given engine design, but one which more than pays its way in freedom from unexpected torsional vibration difficulties. Probably the most desirable torsional system to employ is one in which the single-node frequency brings all one-node vibrations of importance below the normal operating range while the two-

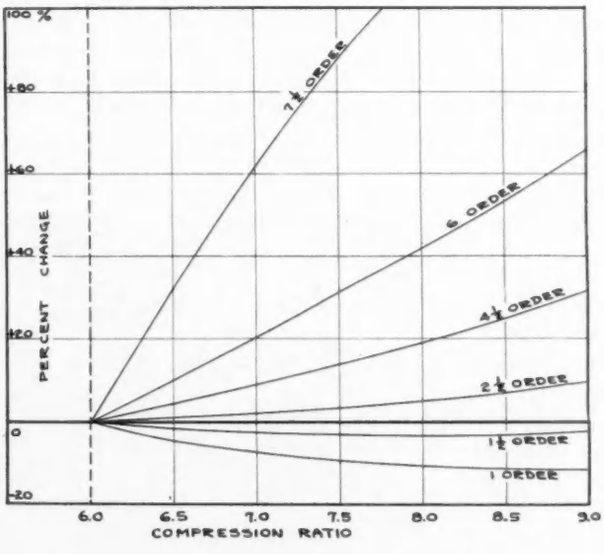


Fig. 5 – Chart Showing the Effect of Increasing the Compression Ratio from 6:1 on Harmonic Torque Input or Relative Torsional Vibration Severity – It Is To Be Observed that Different Harmonics Have Different Characteristics

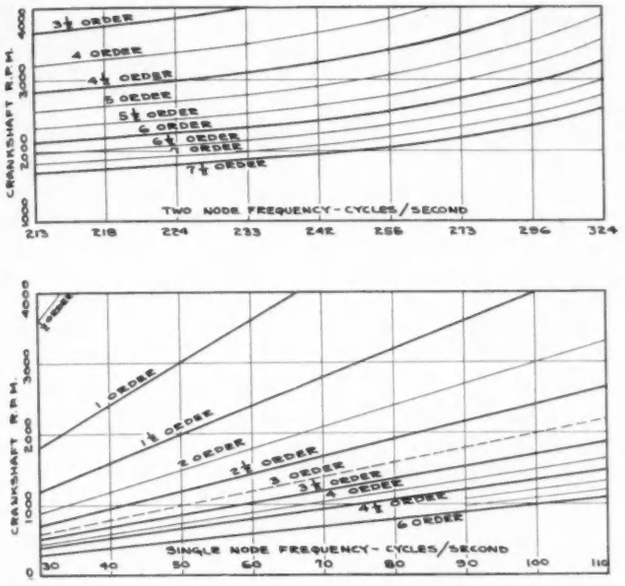


Fig. 6 – One- and Two-Node Resonant Speeds Vs. Frequency for Variable Propeller-Shaft Rigidity

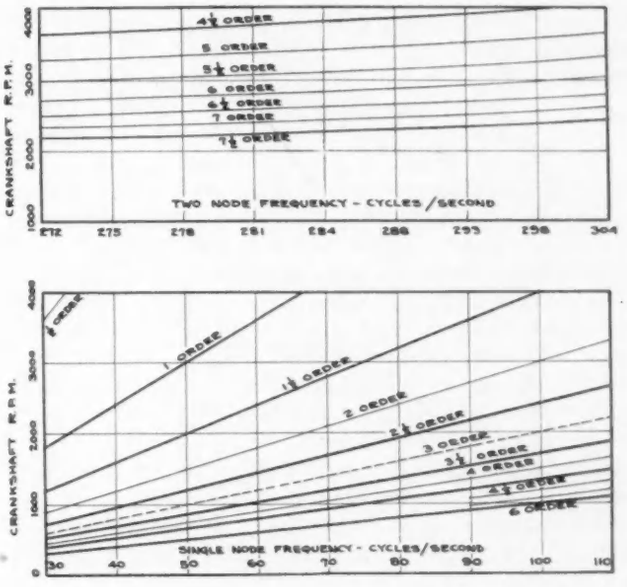


Fig. 7 – One- and Two-Node Resonant Speeds Vs. Frequency for Variable Pinion-Shaft Rigidity

node frequency is adjusted so that the major related harmonics are above the operating speed. An equivalent torsional system to produce these characteristics is shown by Fig. 1 (*c*). In this figure K_6 has very low rigidity while K_7 has a rather high rigidity. This arrangement is somewhat difficult to obtain in conventional engine design but has definite future possibilities.

Torsional Vibration Dampers

In the old days torsional vibration dampers generally were considered a nuisance and were to be avoided wherever possible. They still fall in this same category but, due to the wide speed ranges and high specific outputs involved in modern aircraft engines, vibration dampers have become a necessity just as they did on automobile engines some years ago. Two types of dampers are used generally on in-line engines, namely, the friction and dynamic types. It has been stated before in this paper that vibration amplitude generally is caused by harmonic torque input acting on the crankshaft system. In order to reduce or eliminate the vibration amplitude, this harmonic torque input must be dissipated in one way or another. The energy can be absorbed in heat by a friction damper or removed from the crankshaft system by a dynamic-type damper.

The friction damper is normally a multiple-disc type of alternate steel and bronze plates. One set of plates is connected to the vibrating member and the other set, to a stabilizing or flywheel mass with the least possible angular lash or play. Sufficient oil supply is required to lubricate the plates and remove the heat energy resulting from the harmonic torque input. The chief advantage of the friction-type damper is that it tends to become effective on any kind of vibration as soon as the unstabilized member starts to vibrate. This effective mass is limited somewhat by the slipping torque necessary to absorb the worst vibration in the operating range without excessive amplitude, it being apparent that, as long as the plates do not slip, the flywheel mass or the stiff connecting members become an integral part of the vibration system and affect both its normal characteristics and its amplitude. An objection to this type of damper is that some torsional vibration must exist in order to have energy absorption. It therefore is placed desirably at the point of maximum torsional oscillation which is normally the rear of the crankshaft for both single-node and two-node vibration. This arrangement is hardly compatible with the equally important requirement of attaching one set of plates to a member of considerable mass rotating at constant velocity, the major example of which is the propeller at the opposite end of the engine.

Two fairly satisfactory compromises have been worked out. One is illustrated in Fig. 8 (*a*) in the standard Allison V-1710 engine in which the propeller is used as the flywheel mass by connection to a stiff outer shaft while the engine torque is transmitted through a very flexible inner shaft at the rear of which the clutch is located. If the inner shaft is flexible compared with the crankshaft, the amplitude at the friction damper is high enough to give good damping on single-node vibration without excessive amplitude at the rear of the shaft. The damper also is at an effective point for two-node vibration since it is near the anti-node which may, in a flexible system, be better than at the rear of the crankshaft. However, a friction damper is not particularly effective for two-node vibration since the stresses in the crankshaft

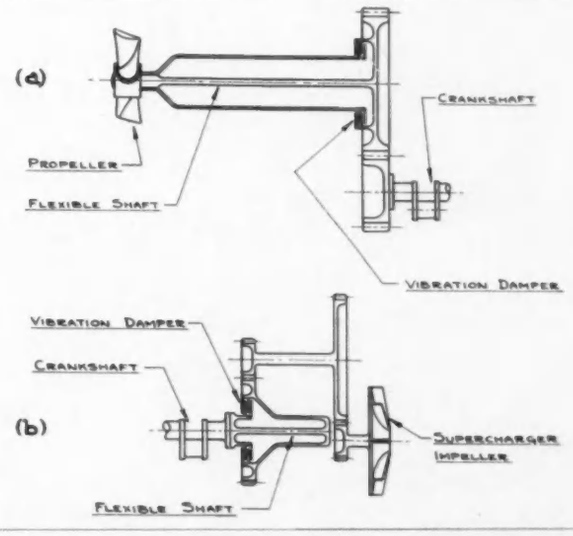


Fig. 8 – Two Types of Friction Damper Applications – In One Case (*a*) the Crankshaft Is Stabilized by the Propeller Inertia and, in the Other Case, (*b*) the Supercharger Impeller Rotating at High Speed Stabilizes the Crankshaft Through the Friction Damper

become high for comparatively small angular amplitudes. It, therefore, can be useful chiefly on those orders where the harmonic inputs are not high. The other compromise used on at least one foreign engine (see Fig. 8 (*b*)) is to attach the damper at the rear of the crankshaft and to stabilize one set of plates by connecting them to the supercharger impeller, which has a marked flywheel effect due to its rotational speed, and to other accessories. For example, a certain 9½-in. diameter supercharger impeller has a moment of inertia of 12 lb-in.[2] When driven through a gear ratio of 8:1, its effective moment of inertia, which varies as the square of the gear ratio, becomes 64 times as much as it would be at crankshaft speed, sufficient to provide a high degree of stabilization for the vibration damper. This arrangement, of course, assumes small backlash in the clutch and a fairly stiff connection from the inertia mass to the damper.

The dynamic-type damper is familiar as the result of its wide application on automotive engines. The requirements for aircraft engines are similar but, of course, much more severe. The pendulum-type damper, so successfully worked out by E. S. Taylor and R. Chilton[4] on radial engines, has an equally good application on in-line engines for eliminating two-node vibration. Like friction dampers, this type should be located in the crankshaft system so that it is subjected to the maximum amplitude of vibration. If the crankshaft system is designed so that the node is approximately in the center of the crankshaft, the damper may be located on either end of the crankshaft. The dynamic-pendulum type damper has the advantage that it can be tuned to almost any harmonic order rather than a particular vibration frequency. When more than one resonant speed occurs within the operating range of an engine, the weights on a dynamic damper can be tuned to the various harmonic orders which cause these critical speeds.

Engine Damping

It has been stated before in this paper that all engines can absorb a certain amount of harmonic torque input due to inherent engine damping and thus automatically mitigate the less serious vibration orders. The magnitude of engine damping is rather complicated to predict mathematically. It is subject to a number of engine variables which do not lend themselves readily to mathematical analysis. In general, it may be said that anything that affects friction horsepower of an engine regulates the internal engine damping. Friction horsepower is generally due to piston, ring, and bearing friction, lubrication-system design, and clearances of the various moving parts in the engine. From test experience it has been found that the engine damping increases with the crankshaft vibration frequency, that is, a crankshaft frequency of 100 cycles per sec. will have a larger damping factor than a system of 35 cycles per sec. The curve shown in Fig. 9 is based on test experience and shows the magnification factor (reciprocal of damping factor) of an in-line engine plotted versus resonant crankshaft frequency. The magnification factor is used conveniently to calculate the torsional vibration amplitude from the harmonic torque input.

[4] See S.A.E. TRANSACTIONS, Vol. 31, March, 1936, pp. 81-89. "Eliminating Crankshaft Torsional Vibration in the Radial Aircraft Engine," by E. S. Taylor.

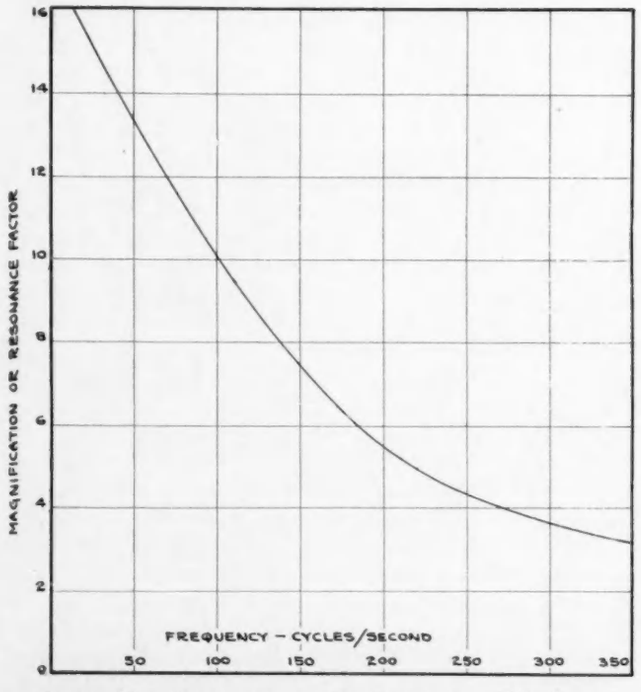


Fig. 9 – Magnification Factor Vs. Frequency – This Curve May Be Used To Calculate the Torsional Vibration Amplitude from the Harmonic Torque Input

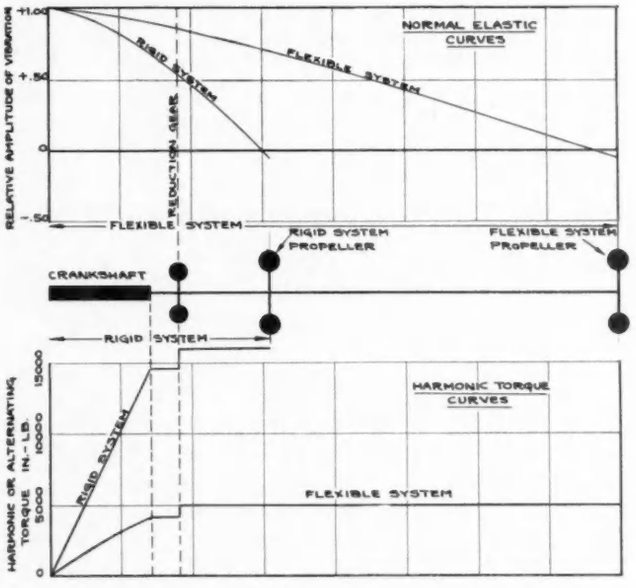


Fig. 10 – Normal Elastic and Harmonic Torque Curves for a Rigid Torsional System and also for a Flexible Torsional System

Interpretation of Torsiometer Data

The chief reason for taking torsional amplitude data with a torsiometer is to determine whether amplitudes exist of a nature which may be destructive to some part of the mechanism, particularly the crankshaft and propeller. It has been customary to limit the torsional amplitude to a definite plus or minus value for a certain type of vibration. With the various grades of fuel available it is possible and becoming perhaps more desirable to have models of the same engine with considerable differences in the power-output and compression-ratio ratings. Various reduction-gear ratios and shafting arrangements are being required. All of these variations may apply to a single basic crankshaft and connecting-rod arrangement for interchangeability and economy reasons. It is obvious, therefore, that, for example, ±1 deg. amplitude at the rear of the crankshaft in single-node vibration cannot mean the same thing in stresses in all of these cases.

Probably the most important variable affecting the torsional amplitude permissible is the frequency of the system used. It is well known that a short stiff rod cannot be twisted as far as a long slender rod to produce the same torsional stress. This rather simple analogy is applicable to various crankshaft

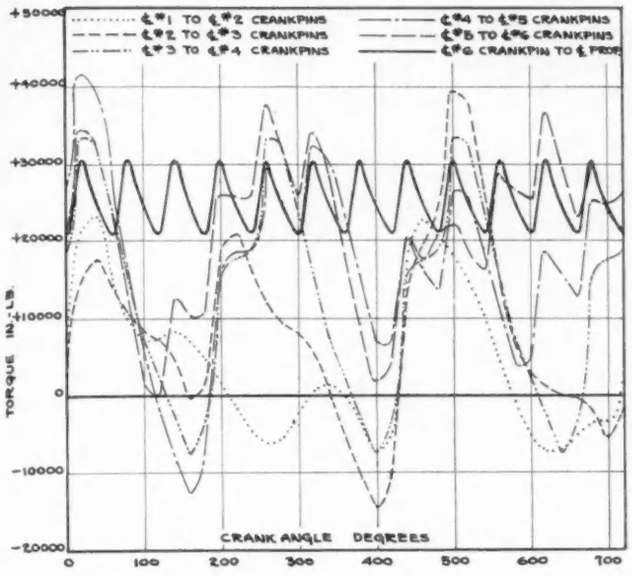


Fig. 11 – Power Torque Variations at Various Places Along the Crankshaft of an In-Line Engine

torsional designs in aircraft engines. When a stiff system is used to produce a high frequency, the engine has what you might call a rather short equivalent system but, when a flexible system is employed, the torsional system is a fairly long one. This point is illustrated by Fig. 9 which shows a flexible and stiff equivalent crankshaft system. These curves are known as the normal elastic curves which show the relative amplitude of vibration at every point in the crankshaft system. The curves are drawn for 1 deg. torsional vibration at the rear of the crankshaft. The corresponding harmonic torque curves also are drawn which show the torque due to this twist at every point in the crankshaft system. It will be observed that, although the twist or amplitudes are the same for both systems, the torque is about three times higher in the stiff system. The harmonic or alternating torque produces a harmonic stress in the crankshaft system which adds directly to the stresses due to power and inertia torque. If the mean torque in both systems were, say 25,000 in-lb., the alternating torque in the flexible system represents a stress increase of 20 per cent while in the stiff system it represents approximately 60 per cent increase for ± 1 deg. amplitude at the rear of crankshaft in single-node vibration. The stress range from maximum to minimum is in the ratio of 1:3 which is even more serious.

Although the two-node frequency is not varied readily through as wide a range as the single-node frequency, the same comments in general apply. The location and inertia of the reduction gear and the ratio of K_6 to K_7 (Fig. 1) have an important influence on both the frequency and location of the nodal point in the crankshaft. A node at the center of the crankshaft for two-node vibration is the equivalent of a short or stiff system in one-noded vibration, whereas a node at any point further from the free end of the crankshaft gives the equivalent of a more flexible system. For example, $\pm 1/4$ deg. amplitude with a node at No. 4 main journal (counting from free end of shaft) will represent a 33 per cent increase in the alternating torque over that for the same amplitude if the node is at No. 5 main journal.

Another factor having an important bearing on the permissible amplitude at the rear of the crankshaft is the way the torque builds up in the crankshaft due to the firing order. Fig. 11 shows the torque variation along the crankshaft as calculated from indicator cards and inertia forces for a twelve-cylinder 60-deg. vee engine. It is interesting to note that negative torque exists at certain times from the rear of crankshaft forward to No. 5 crankpin, and that the torque variation at the worst point varies from —8 per cent to 160 per cent of the engine mean torque. The flywheel effect of the centrifugal and inertia masses in the crankshaft probably explains the fact that this variation does not show up on the usual torsiometer cards. Since the alternating torque from resonant vibration is superimposed on the torques shown in Fig. 11, it is apparent that the phasing of the alternating torque is an important item as is the location of the node or nodes during vibration. It is also obvious that any general rules as to permissible amplitude of vibration are extremely difficult to apply as regards the in-line type engine. After some study the writers conclude that the limits of torsional amplitude with reliability can be set at present only by the individual engine manufacturer and may vary for each model of a given engine type, as the result of experience with a particular crankshaft system design. After once establishing a safe operating limitation for a particular model engine, the effect of changing any of the many variables can be predicted rather closely by the methods discussed in this paper.

Amplitude curves of a twelve-cylinder engine for both single-node and two-node vibration are shown in Fig. 12. Curve (*a*) shows the calculated harmonic torque input. Curve (*b*) shows the actual test results from torsiometer data on the engine when no dampers were used. Curve (*c*) shows torsiometer data when suitable damping has been provided on the engine.

As an example of what can be done with the torsional factor, Allison engines have been built with variations of 56 in. in the distance from the center of the engine to the center of the propeller hub without change of either the single-node or two-node frequencies. This construction permits the same crankshaft and damper arrangement to be used on two or more widely differing models with a resultant high degree of interchangeability.

Acknowledgment is made to the Materiel Division of the U. S. Army Air Corps for permission to publish the information contained in this paper and also for its assistance in working out several torsional-vibration problems encountered in the design of the V-1710 engine.

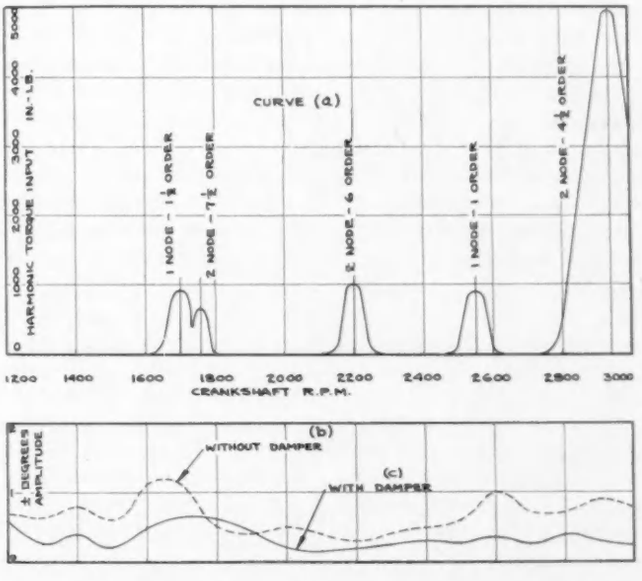


Fig. 12 – Curve (*a*) Shows the Calculated Harmonic Torque Input for Several Important Harmonic Orders – Curve (*b*) Depicts the Resulting Amplitude Curve Obtained from a Torsiometer During Actual Engine Test Without a Vibration Damper – Curve (c) Was Obtained with a Vibration Damper Installed on the Engine

Truck Performance

By Merrill C. Horine

Sales Promotion Manager, Mack Manufacturing Corp.

A PROBLEM well analyzed is half solved. Truck performance certainly seems to be one of highway transport's most pressing problems, if we are to judge by interest in its various phases exhibited by manufacturers, operators, and legislators alike. He would be bold indeed who, in view of the many minds which have wrestled with this problem, dared to step forward at such a gathering as this claiming to hold the answer.

The Society of Automotive Engineers is a deliberative body; that is where engineers differ from legislators. An engineer endeavors to learn and to analyze the facts concerning a matter of importance and, from these facts, to deduce conclusions upon which a profitable course of action may be based.

Legislators, on the other hand, completely reverse the process, starting with a desired course of procedure (not always a profitable one), arbitrarily making rules calculated to lead to such a course of action, and then, by induction, endeavoring to parade an array of facts, prejudices, and impressions in support of the predetermined conclusion.

As this is an engineering society, I shall endeavor to proceed according to the first-described process, and it is my hope that, by conforming to the adage first cited, I shall so analyze the problem of truck performance that its solution, if not half attained, at least shall be furthered to such an extent as my limited ability, plus the wisdom which I feel certain will be displayed in the discussion which I confidently expect and solicit, may produce.

The first step in analyzing any subject is its definition. My definition of performance is the rate of doing work, "work" being defined as the transport of a load over the highway, and the "rate" at which it is done being miles per hour. According to my conception, "performance" is quite distinct from capacity. Were capacity alone required, there would be no motor trucks, for the warehouse, coal pocket, storage tank, gravel pit, loading platform, or wherever else the load originates, offer all of the capacity required. Performance is concerned entirely with the manner in which the load is made to move from the point of origin to the point of destination.

Of course, capacity has a very definite practical bearing upon performance. A boy on a motorcycle can provide performance which is undoubtedly superior to anything possible with a motor truck, but only at a very limited capacity for load. It may be accepted without argument, therefore, that a given standard of performance becomes difficult to maintain in proportion to the amount of load to be moved.

This seems such a simple proposition that it might seem puerile to mention it here. Continual contact with motor-truck salesmen and a type of prospect more typical than the decidedly superior sort found at S.A.E. meetings has convinced me, however, that this fundamental relationship between capacity and performance is, by no means, as generally appreciated as might be desired. In fact, there are large institutions within this industry who constantly, in their sales talks, their sales literature, and their advertising, affirm or imply that, by the application of additional structure to a standard motor-truck chassis, it may be enabled to transport considerably increased loads at no sacrifice of performance. Operators prevail upon or are prevailed upon by salesmen to overtire truck chassis for the purpose of increase in lading, while expecting a degree of performance identical with that of the chassis as originally rated.

Although strength has no direct bearing upon the subject of this paper, it is useful at this time to reflect that the structural strength of the chassis also has a definite bearing upon the performance which is practicable. Just as vainglorious skippers, racing their stern-wheelers on the storied Mississippi, tied down their safety valves so that their boilers exploded, so also modern truck operators occasionally exceed prudent performance standards by tampering with governors, over-inflating tires, and unduly compounding gear ratios, with similarly dire results.

Performance has several phases which might be reduced to factors in a general performance equation. One of these phases is speed on the level road. Of course, by this phrase is not meant the speed in high gear at which the governor throttle closes, for this is not the maximum speed of which the vehicle would be capable if geared for the maximum at which the engine power could move the truck. Such speed maximums, in fact, almost always would be higher than safety and reasonable law abidance would permit. A more familiar phase of performance is grade-climbing ability, this phase alone constituting in reality a comprehensive index of performance, and is so generally accepted. Another performance phase is that of drawbar pull on the level and, like grade-climbing ability, represents the excess tractive ability possessed by the vehicle over and above that required to maintain a given speed upon a level roadway of normal tractive resistance.

Another phase of performance which is given more serious consideration by operators of buses than of trucks is that of acceleration. This, too, is in reality equivalent to grade-climbing ability, as is indicated by Table 1: "Tractive Effort Equivalents."

All of these phases are, in reality, different forms of excess tractive ability over and above that required to maintain a uniform speed on a level road of normal tractive resistance. Were there no such excess of tractive ability then, unless a down-hill start were given, a motor truck could never even attain a normal highway speed, much less climb a grade, pull a trailer, or accelerate.

The significance of performance at this time arises from several considerations, the foremost of which is the strong probability that, in the near future, legislation, either National or State or both, will be enacted, or regulations adopted requiring certain minimum-performance ability of commercial vehicles, either as a prerequisite to registration, or as a requirement in operation. By direction of Congress, the Bureau of Public Roads, in cooperation with the Highway Research

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Board and the National Research Council, as well as the Bureau of Standards, the Interstate Commerce Commission, the Bureau of the Census, the Connecticut Department of Motor Vehicles, the Michigan State Police, the National Safety Council, the Bureau for Street Traffic Research of Harvard University, and many others, conducted an exhaustive research into traffic conditions and rendered a report to Congress on June 30, 1937. On page 59 of this report appears the following significant statement:

"In the case of freight vehicles, there is urgent need of a better adaptation of the power of the engine to the loaded weight of the vehicle and their conditions of use. Vehicles supplied by their manufacturers with power intended for the propulsion of a definite maximum weight are being used to transport loads which greatly exceed the designed limit. Vehicles designed and powered to be trucks are converted without essential change into tractors and hence, as such, are used to tow trailers and semi-trailers, imposing tractive demands far in excess of the ability of the motor. The crawling speeds to which such excessively loaded vehicles and combinations are reduced on even moderate grades are a serious strain upon the patience of the drivers of following passenger cars, and are often the provoking cause of hazardous attempts to pass.

"Except as such excessive loading is encouraged by competitive sales policies, there is a condition for which the manufacturers of vehicles are not directly responsible and cannot directly control. *It is apparent, however, that greater hill-climbing ability is an outstanding need of freight vehicles.* This ability may be obtained by providing the greater motive power necessary to move the present maximum loads or by reducing the loads to limits fixed by the ability of the present motors. But, in either case, it will be necessary, if the present annoying and dangerous uses are to be prevented, that the performance ability of trucks and tractors shall, in some terms, be exactly defined by their manufacturers and that, by adequate legal regulation and enforcement, the use of such vehicles shall be kept within the defined limits."

In a report to the American Association of State Highway Officials, the Committee on Highway Transport at Boston, Mass., on September 30th, 1937, stated:

"Accurately measured speeds of truck units, particularly of the tractor-trailer type in the 30,000- to 40,000-lb. class, operating on 6 per cent grades, indicate a quite uniform maximum speed of 5 m.p.h. A conservative normal speed of 40 m.p.h. for the lighter units presents a speed differential of 35 m.p.h. between these and the heavier units observed, an impossible operating condition on the principal highway routes.

"In line with previous reports of this Committee, the studies of the past year emphasize the acute necessity for establishing and enforcing minimum speeds and braking requirements with particular reference to the heavier classes. . . . The Committee holds that, in enforcement of performance requirements, methods that could be put into general use by the highway policemen are essential."

In November, 1937, the Interstate Commerce Commission issued an order for a study and report on these requirements as to "the sizes and weights of motor vehicles and combinations thereof in so far as they affect the safety of operation." In view of this situation, the Automobile Manufacturers Association already has requested the Society of Automotive Engineers to renew its efforts at evolving a sound performance formula which would be useful in these ways and, although there are many in the industry who look with misgiving upon the present tendency toward requiring that all motor trucks shall be capable of a given minimum performance, contending that the basic premise upon which the demand for such performance is based is fallacious, nevertheless none will deny that a better understanding of performance and more reliable means for gaging it will be distinctly beneficial, not only to users of highway transport vehicles, but also to the majority of manufacturers of highway vehicles and equipment who endeavor to sell them upon a straightforward basis.

The trend toward such regulation is far from having crystallized into definite form, but it seems probable that it will be either by formula, prescribing a certain definite relationship between cubic inches of piston displacement and permissible maximum gross vehicle weight; limitation of tire equipment, according to piston displacement; type-testing and rating of various models of various makes; or the development of instrumentation for actual road tests. Research embracing all phases of this matter has been going forward for some time under the direction of the Bureau of Public Roads of the Department of Agriculture, and more or less at the behest of the Interstate Commerce Commission, by Prof. J. Trueman Thompson of Johns Hopkins University, which he reported to the Society at its Annual Meeting in Detroit last January[1]. It certainly seems desirable, in view of this condition, that the Society of Automotive Engineers take due cognizance of this matter and proceed with all feasible dispatch to complete the work started several years ago by its Rating Committee.

Another reason why performance should be of widespread interest at this time and at all times is its significance from the operating standpoint. The value of performance to the operator is chiefly in that it permits a higher average speed to be maintained within a given maximum speed, thus per-

[1] See S.A.E. TRANSACTIONS, May, 1938, pp. 209-214: "The Outlook on Legal Performance Requirements," by J. Trueman Thompson.

Table 1 – Tractive Effort Equivalents

Tractive Effort, lb. per ton	Per Cent* Grade Ability	Acceleration† on the Level, m.p.h. per sec.	Tractive Effort, lb. per ton	Per Cent* Grade Ability	Acceleration† on the Level, m.p.h. per sec.
100	3.5	0.77	800	38.5	8.46
125	4.75	1.04	825	39.75	8.73
150	6.0	1.32	850	41.0	9.01
175	7.25	1.59	875	42.25	9.28
200	8.5	1.87	900	43.5	9.56
225	9.75	2.14	925	44.75	9.84
250	11.0	2.42	950	46.0	10.01
275	12.25	2.69	975	47.25	10.04
300	13.5	2.95	1000	48.5	10.06
325	14.75	3.25	1025	49.75	10.09
350	16.0	3.52	1050	51.0	11.2
375	17.25	3.79	1075	52.25	11.5
400	18.5	4.06	1100	53.5	11.75
425	19.75	4.37	1125	54.75	12.05
450	21.0	4.62	1150	56.0	12.3
475	22.25	4.39	1175	57.25	12.6
500	23.5	5.17	1200	58.5	12.75
525	24.75	5.45	1225	59.75	13.15
550	26.0	5.72	1250	61.0	13.4
575	27.25	5.99	1275	62.25	13.67
600	28.5	6.26	1300	63.50	13.95
625	29.75	6.54	1325	64.75	14.22
650	31.0	6.82	1350	66.0	14.5
675	32.25	7.09	1375	67.25	14.8
700	33.5	7.36	1400	68.50	15.1
725	34.75	7.63	1425	69.75	15.33
750	36.0	7.91	1450	71.0	15.6
775	37.25	8.18	1475	72.25	15.9

* $\dfrac{\text{Tractive Effort, lb./ton} - \text{Tractive Resistance (30 lb./ton)}}{\text{Grade Resistance (20 lb./ton/per cent grade)}}$

† $\dfrac{\text{Tractive Effort, lb./ton} - \text{Tractive Resistance (30 lb./ton)}}{\dfrac{\text{Mass}}{\text{Gravity}}} \left(\dfrac{\text{sec. per hr.}}{\text{ft. per sec.}}\right)$

$= \dfrac{\text{Tractive Effort} - 30}{62.1} (0.682)$

mitting a greater number of miles per day, and hence permitting given routes and schedules to be covered with a smaller number of vehicles or, in the case of long-distance transportation, permitting faster schedules between termini – hence greater attraction to traffic and lower overall costs.

In construction work, greater performance has many advantages, including ability to operate under more difficult road conditions, thus reducing the expense of maintaining roadways, assuring maximum use of power-shovel time, permitting steeper and hence cheaper ramps out of excavations, and general speeding up of construction work.

In general, however, the value of performance ability in normal trucking work is more or less in ratio to the maximum speed permissible, and also in ratio to the prevailing grades upon the routes. Thus, in metropolitan delivery and drayage service, maximum speeds are so reduced by the congestion of traffic, that higher accelerative ability is of small moment. In flat terrain, such as in Southern New Jersey, Eastern Michigan, Southern Illinois, and Central Texas, for example, where hauls are long, stops infrequent, and grades virtually non-existent, performance capability of very moderate order seems to suffice. In hilly sections, such as in the States of Pennsylvania, New York, the Western Carolinas, Eastern Kentucky and Tennessee, and the Pacific Coast and inter-mountain States, just the opposite is the case. In these regions, more than ordinary performance ability is essential.

In those services in which motor transport is conducted in competition with other forms of transportation, or where motor freight lines compete, good performance may mean better service and, therefore, competitive advantage, and smaller fleets, which may mean lower cost for the entire operation.

One further advantage of good performance for the operator is the effect which it has upon driver morale. Every driver knows that a good-performing truck is far easier to handle than one whose inferior performance requires constant nursing, frequent changes of gear, and a continual nervous strain. It is well recognized by experienced operators that pride in his mount inspires the driver to more considerate operation. Drivers are less apt to abuse trucks which they admire than those whose performance is a continual disappointment.

From the safety standpoint, the contention is at least tenable that a good-performing truck is less apt to be involved in accidents, through offering the driver less temptation to attempt to make up time by reckless coasting, or to attempt to "beat the lights." Certainly, if a truck be governed to a safe maximum speed, the effects of better performance ability will not be to increase this maximum, but rather to permit faster acceleration to such speed, and the maintenance of closer to that speed when negotiating grades.

Of perhaps greatest current interest among operating advantages of performance is improved public relations. Irrespective of our personal opinions regarding the effects of better performance upon safety, it is certain that, as reflected in the official attitude of the American Automobile Association, the general motoring public regards slow-moving trucks upon two-lane highways with great disfavor and, unquestionably, a more sympathetic public attitude toward motor transportation would redound greatly to the benefit of all concerned with it.

Without doubt, nevertheless, extreme performance ability involves greater operating costs although, where the operation is such that improved performance means increased earning capacity, overall results actually may show economy. Certainly, other things being equal, the greater the daily mileage, the lower the cost per mile. Despite the popular impression to the contrary, it is not true that the biggest engine always consumes the most fuel. Every engine has an economical range of speeds and loads. An engine of adequate power for the work it is to do may operate the greater part of its time within this economical range, whereas an engine of insufficient power may be forced to operate most of its time closer to its maximum output, at which its economy will be appreciably less. On the score of maintenance cost, too, a truck of greater power is quite apt to prove more economical because the unit pressure on its bearings, as well as its lower average speed, results in a lower *PV* factor and, therefore, greater durability of bearings. To a certain extent, in all other respects, its durability and reliability will be increased.

High power has a similarly beneficent influence upon transmission maintenance, since a greater part of its operation may be in direct drive and there should be fewer gear changes. Higher power probably will mean less rear-axle reduction, which will result in a more favorable proportion of pinion-to-ring-gear, more teeth in mesh, and greater resiliency of torque from the engine, due to more moderate reduction. There will be less flywheeling with an amply powered engine as compared with an under-powered one, and thus all parts of the driving line will be relieved from the shocks incident thereto.

As against all of this argument, larger engines, larger clutches to transmit their torque, bigger transmissions, and heavier rear ends unquestionably will cost more to build, and therefore to buy, and will involve more tare weight in the vehicle. If improved performance means more ton-miles per hour, then we may certainly expect to burn more fuel per ton-mile. Just how these opposing influences will balance out cannot be determined theoretically, but it is certainly reasonable to suppose that the most economical vehicle probably will be neither a drastically under-powered nor an extravagantly over-powered one.

With this conception of the nature and significance of performance, we are next concerned with its measurement since, if performance is a desirable thing, then we must be able to measure it in order to determine the relative fitness of different types and specifications of vehicles, or to determine from an analysis of a transportation project just what degree of performance ability will be required. Performance ability actually is rear-wheel horsepower, the factors of which are tractive effort in pounds and speed in miles per hour, as shown in Table 2: "Rear-Wheel Horsepower Equivalents." Thus, 50 rear-wheel hp. at 33,000 ft-lb. per min. is equal to 1,650,000 ft-lb. per min. A speed of 20 m.p.h. at 88 ft. per min. per m.p.h., is 1760 ft. per min. Dividing, we find that 50 hp. at 20 m.p.h. represents 937 lb. tractive effort. With

Table 2 – Rear-Wheel Horsepower Equivalents

Rear-wheel Hp.	Miles per Hour							
	5	10	15	20	25	30	35	40
40	3000	1500	1000	750	600	500	428	375
45	3375	1687	1125	843	675	562	482	422
50	3750	1875	1250	937	750	625	535	469
55	4125	2062	1375	1031	825	687	589	515
60	4500	2250	1500	1125	900	750	642	562
65	4875	2437	1625	1219	975	812	696	609
70	5250	2625	1750	1312	1050	875	750	656
75	5625	2812	1875	1406	1125	937	803	703
80	6000	3000	2000	1500	1200	1000	857	750
85	6375	3187	2125	1594	1275	1062	911	797
90	6750	3375	2250	1687	1350	1125	962	843
95	7125	3562	2375	1781	1425	1187	1018	890
100	7500	3750	2500	1875	1500	1250	1070	937
105	7875	3937	2625	1969	1575	1313	1125	984
110	8250	4125	2750	2062	1650	1375	1178	1031
115	8625	4312	2875	2156	1725	1437	1232	1078
120	9000	4500	3000	2250	1800	1500	1286	1125
125	9375	4687	3125	2344	1875	1562	1339	1172

this computation as a basis, we may determine easily net grade ability at any given speed, maximum speed on any given grade, or the theoretical maximum acceleration, given the gross weight of the vehicle and the tractive resistance of the road surface over which it is moving. All of this computation naturally ignores air resistance or wind resistance, and also assumes that the gearing is such that the engine will be capable of developing its maximum horsepower at the speeds taken.

This method sounds simple enough, but it is when the *assumptions* are investigated that the problem commences to become complicated.

Real rear-wheel horsepower is the difference between the net brake horsepower of the engine and the mechanical losses in the transmission of this power to the rear tires. It is customary to set up an overall mechanical efficiency in terms of percentage, to approximate the magnitude of this loss. If this scheme has been satisfactory in the past, it has been because, like all rules of thumb, it has been accompanied by other assumptions equally specious, whose errors to some extent have mutually corrected one another, so that the terminal result has approximated accuracy within practical limits. As a matter of fact, it would appear that almost none of the various assumptions entrained in conventional formulas for performance is to any degree scientific or accurate.

For example, the basic conception of mechanical efficiency as a loss in per cent of engine brake horsepower is a fallacy since these losses are a combination of rubbing and fluid friction whose magnitude is largely inherent in the transmission and driving system, and not to any appreciable extent affected by the input horsepower. Throughout the driving system of a conventional motor truck, there are, on the average, 28 ball or roller bearings and 2 plain bearings, to which must be added 4 anti-friction bearings on the front wheels, making 34 in all. The drive is taken through but 2 gears in a single-reduction drive in direct drive; or 8 gears in a double-reduction in any other forward speed; or 10 in reverse. In addition to this condition, and far more important in so far as power losses are concerned, there are two closed boxes filled with heavy viscous lubricants, in one of which there are at least 10 and, at most, 18 to 20 gears constantly turning, acting as churning paddles. To all of this as a source of friction, we must add not less than 8 sealing rings which, particularly in a new truck, exert a considerable constant braking effect; and the internal friction of the tires. All of this discussion takes no account whatever of possible brake dragging; or windage on such parts as the clutch, shaft brakes, universals, wheels, or brake drums.

Researches have shown that all of these losses tend to remain more or less fixed at any given speed, regardless of the amount of power being transmitted. Modern anti-friction bearings in some cases actually have lower friction drag under heavy load than when lightly loaded. Modern hardened and highly-finished gears produce surprisingly little friction so long as they are not loaded beyond reasonable limits. In general, the friction loss tends to increase with speed, this condition being particularly true of the oil-churning losses.

Obviously, then, the time-honored method of assuming the lowest mechanical efficiency in the slowest gears and the highest mechanical efficiency in the fastest gears will not bear analysis, and it has been proved over and over again that, in a given chassis, a change from one engine to another of greater power results in a net gain in rear-wheel horsepower almost exactly equal to the increase in engine power.

The tractive resistances of different types and conditions of roads have been the subject of a great many careful studies from which some general ratings have been developed. A close study of all of these researches shows that any general classification of types of road surfaces in terms of tractive resistance is subject to exceedingly large errors. The time-honored figure of 30 lb. per ton, although a very convenient figure very widely used, is certainly not representative of prevailing pavements on main traveled routes.

Turning next to the brake horsepower of the engine, the majority of performance calculations are based upon torque rather than horsepower, the torque figure ordinarily used being the laboratory peak, which usually is assumed cavalierly to be a fixed quantity, available under any and all conditions of operation. The facts, of course, make such assumptions ridiculous. In the first place, it is the usual custom in publishing engine test curves, to show output without accessories, and to correct such output to a normal barometer and an assumed normal intake air temperature of 60 deg. fahr. Some justification for these corrections is afforded in the fact that dynamometer tests usually are made on comparatively new engines whose frictional horsepower may be accepted as somewhat higher than would be the case after the engine had operated for, say, 250 hr. But, inasmuch as the accessories commonly used on motor trucks, including the fan, water pump, and generator alone, will account for from 6 to 9 hp., the net power available at the flywheel certainly will be noticeably less than shown by the curves, even where the curves have been drawn honestly directly from the actually observed dynamometer-beam scale readings.

There is also a distinct fallacy in basing performance calculations upon the peak of the torque curve, since, as a rule in heavy-duty types of engines, the torque peak occurs at relatively slow speed, speeds slower than a skilful driver will use ordinarily in attempting a difficult pull. In giving performance predictions, it is customary to quote both grade ability and speed. In most instances, however, the grade-ability figure is based upon the peak torque, whereas the speed is figured from the governed maximum. Seeing that the torque at governed speed may be in the neighborhood of 80 per cent of peak torque and that the miles per hour at peak torque in any gear will not be faster than half that at governed speed, the disparity is at once evident.

The truck-operating public seems to have an almost pathetic faith in the efficacy of gears and, what is still more curious, a belief that the arrangement of reduction gears, rather than the total reductions which they provide, is the controlling influence on performance.

Arguments are heard on every side as to the relative merits of over-geared versus direct-in-high transmissions, auxiliary transmissions versus two-speed rear axles, single-stick versus two-stick shifting arrangements. All sorts of fanciful claims are advanced respecting the fuel economy to be gained as a result of a geared-up high; the quick-return principle in truck transmissions; the mysterious two-range effect; and the poesy of overlapping ratios. It seems that the industry always has been beset by credos of this nature. A few years ago, truck buyers were far more interested in the form of gearing used in the rear axle than in the gear ratios which it provided. We heard much of the virtue of applying the power near the rim of the wheel, as in the internal-gear-drive axle, as distinguished from applying it near the hub, as in the worm- or double-reduction type.

Through all of this aura of mystery and magic, the simple facts with respect to gears remain clear and simple for all to understand. All that gears can do is to provide a mechanical advantage in the form of a reduction in speed, thus serving to multiply the torque and divide the speed. No type of gearing, no matter where it is located or how it is constructed, can do more than this; nor is there any way of relaxing the inexorable law that speed and torque must forever be reciprocal.

Of course, as some of my friends have been at great pains to impress upon me, there are often weighty commercial considerations which enter into the choice of gear arrangement. It is certainly true that a lighter and smaller gear box for a given range of ratios and a given torque capacity is possible by means of over-gearing than where the top speed is direct. Undoubtedly still further gains in these directions might be made if the top two or the top three speeds were to be over-geared. Much might be said, too, concerning the penalty of such gear-box design, a penalty paid by the universal joints because of being required to operate at higher than crankshaft speed; by the rear-axle gears which must have smaller bevel pinions with fewer teeth, and flatter ring-gears to produce the abnormal reductions necessitated by the over-gear; the higher tooth speeds of the gears making shifting more difficult; and the noticeably higher oil-churning losses. It might be pointed out that, inasmuch as top gear in any event almost always will be used a greater part of the time regardless of whether it be direct or over-gear, that the over-gear involves an excessive amount of running through the gears, but all of this is irrelevant to the subject of this paper.

Gearing certainly has a vital bearing upon performance for, although the basic and intrinsic performance ability of a vehicle consists of the ratio of its rear-wheel horsepower to its gross weight and although from this ratio a specific grade ability may be obtained for any speed, the road speeds at which the engine may produce its full power are dependent upon the total ratios provided. If, for example, the vehicle had but two ratios, one of which afforded a maximum speed of 40 m.p.h. and the other a maximum speed of 5 m.p.h., a speed of 20 m.p.h. could be attained only in the faster ratio but, if the engine were governed at 2000 r.p.m., this speed would be at 1000 r.p.m., at which speed the engine could develop only slightly over half its horsepower; and a speed of 50 m.p.h. would be impossible. Consequently, the first requirement for the transmission and driving gearing is that it shall provide a series of total ratios, which shall encompass a range adequate to meet maximum speed requirements at the top and maximum tractive effort required at the bottom; and that between these extremes there shall be intermediate ratios of sufficient number and at such intervals as will provide the speed and tractive effort which operation requires.

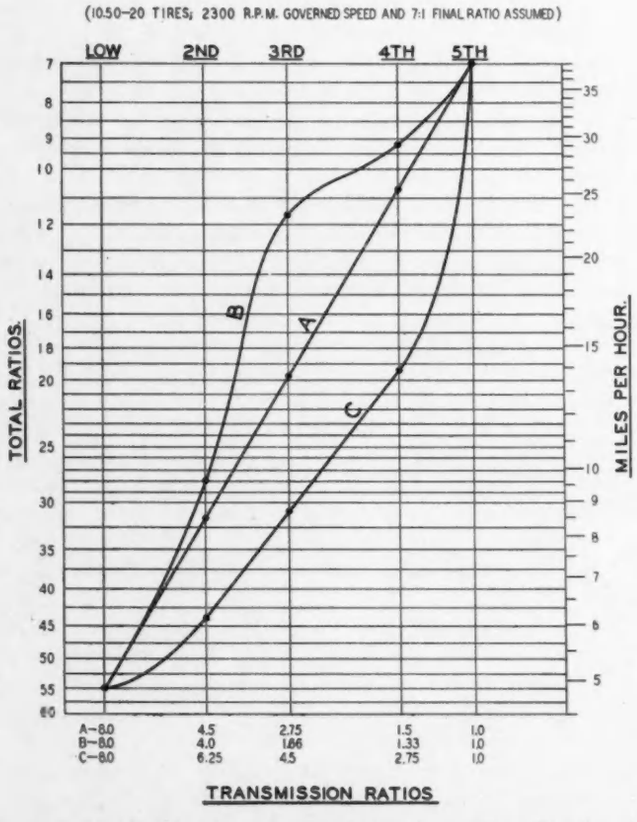


Fig. 1 – Gearshift Ranges and Intervals vs. Total Ratios and Speeds

In straight selective shifting, five forward speeds generally are considered the maximum which is practical, a greater number than this requiring either two ranges of countershaft speeds, an auxiliary shift either ahead of or behind the transmission, or a two-speed final drive.

In normal truck operation, both the number of and the intervals between ratios are dependent upon the horsepower-to-weight ratio. Thus, in light package cars, where the power-to-weight ratio is practically the same as in passenger cars, three speeds generally prove adequate. In light trucks, up to 14,000 to 16,000 lb. gross rating, four speeds generally are satisfactory whereas, in the heavier sizes, five speeds are needed.

If, however, through the use of six-wheel attachments or semi-trailers, the gross weight is increased out of proportion to the horsepower in any of these sizes, then the number of speeds required should be increased and the intervals between them reduced. The range provided by any transmission or combination of transmissions and auxiliaries, and the intervals provided, remain constant so far as ratios within themselves are concerned but, when compounded with the final ratio, they tend to expand as the rear axle reduction increases, and vice versa. They also expand and contract with differences in tire size. For example, a standard transmission which provides an interval of 2.9 between fourth and fifth speeds with the fastest rear-axle ratio for this model, expands this interval to 4.07, with the slowest ratio. Similarly, with the same ratio in the rear axle, the interval in m.p.h. between fourth and fifth speeds in the same truck varies from 11.3, with the smallest standard tire, to 12.5 with the largest and, with the same-sized tire, the speed interval as between the slowest and the highest rear-axle ratio varies from 13.6 to 10.8 m.p.h. Thus it is seen that, although greater rear-axle reduction tends to increase the spread of ratios and widen the intervals, it decreases the range of speeds in miles per hour, and decreases the intervals.

From this fact arises the problem of how to arrange the order of progression through the different ratios in a transmission. For many years, the so-called geometric shift has been accepted as ideal. Fig. 1 illustrates such an ideal geometrical shift as Curve *A* which, on logarithmic ordinates, results in a straight line. Remembering what we have just found concerning the effect of ratios upon speed intervals, we find that this shift provides rather extreme differences in successive speeds as between the top gears and, for this reason, particularly in motor freight operations, there has been a demand for closer steps at the top. The fact that over-geared transmissions, as ordinarily built, have provided these closer steps at the top accounts in no small measure for the popularity of this otherwise undesirable type of transmission. The fact that there is nothing inherent in either the over-geared or the direct-in-high transmission which necessitates this difference in the pattern of progression, is beside the point at this time.

Curve *B* shows an ultra-geometric progression in which the wide intervals in terms of miles per hour at the top have been brought closer together, thus permitting more close adjustment of ratio to slight variations in grade, and thereby permitting higher average speeds over gently rolling terrain.

In dump-truck operation, oil-field work, and other applica-

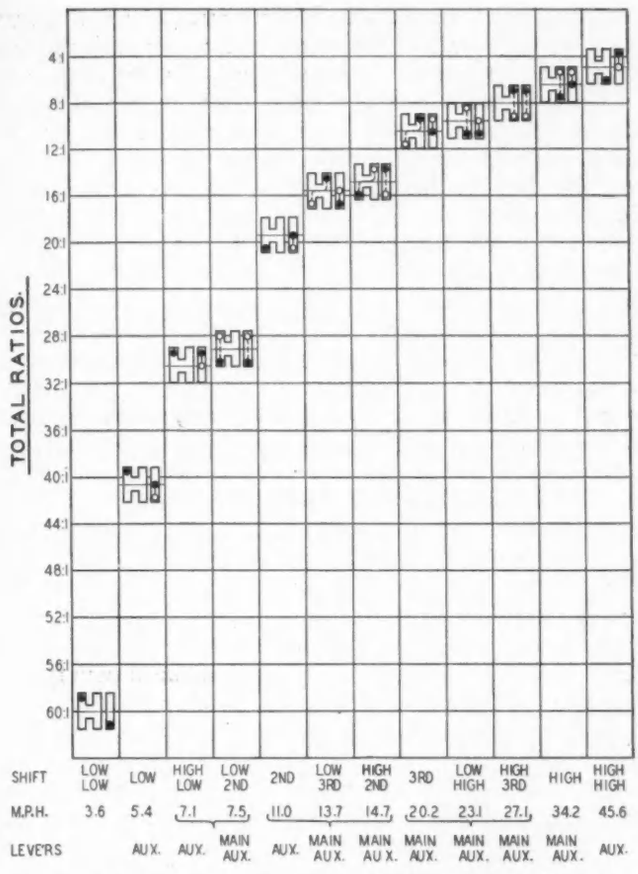


Fig. 2 – Diagram of Shifts – Four-Speed Plus Three-Speed Auxiliary – Showing Progression Through Full Range

tions where extremely hard pulling is encountered much of the time, the widening of the intervals at the bottom of Curve *B* introduces a highly undesirable situation at the lower end, whereas the greater adaptability at the upper end serves hardly any useful purpose. In this type of operation, we are more interested in ratio intervals than in speed intervals, and we are distinctly more interested in the pattern of progression at the lower end of the range than we are in that at the upper. As trucks of this description are prone to use the slower rear-axle ratios, the difficulty is aggravated still further because of the expansion of the interval between speeds already noticed. Accordingly, Curve *C* has been plotted upon the same chart to show the characteristic shape of a curve in which the progression is more nearly arithmetic. Here we find the intervals much closer together at the bottom and farther apart at the top.

Observation of actual working conditions encountered by both highway freight and dump trucks, logging trucks, and oil-field trucks indicates that this wide difference in characteristics is most decidedly desirable. However, the number of trucks requiring these characteristics usually has been considered too limited to warrant the production of these special types of transmission, the conventional type, as indicated on Curve *A*, being considered a compromise between the two which most nearly would fit the majority of truck applications.

Because of this situation, auxiliary transmissions and two-speed rear axles are being offered to afford an approximation of the ratios required in these two fields. In order to effect such additions, over-geared auxiliaries may be used to provide a faster overdrive slightly higher than direct drive, and under-geared auxiliaries to supply a greater reduction than the regular low.

Recently there appeared in the public press an article describing the alleged advantages of a three-speed auxiliary transmission, affording both an over-gear and an under-gear, in addition to the straight-through drive. In this article, an ideal pattern of ratios was indicated, and with it, the positions of the two gearshift levers necessary to make the full progression. To illustrate better what this arrangement involves, I have prepared a chart, Fig. 2, indicating that, in almost every case, each shift requires the movement of both levers. Only one who has never driven a heavy vehicle on the highways or in an excavation pit can miss the humor of the contortions thus prescribed.

This chart also shows further how many of the allegedly split speeds are actually equivalent. A glance at the chart shows that the high-low and low-second are virtual duplicates, as are also low-third and high-second, and third and low-high. As a matter of fact, second, low-third, and high-second are virtual triplicates, as are third, low-high, and high-third. The arrangement actually seems to provide seven usable speeds. This overlapping of ratios is generally typical of attachment-type auxiliaries. A close study of Fig. 3 will indicate that the ratios in the main transmissions afford, within their range, a reasonable interval and progression of speeds, and that all the auxiliary accomplishes is to provide an over-geared high above the regular speeds, and an under-geared low below the regular speeds. Used in this manner, such an auxiliary begins to make sense and we have, instead of 12 speeds as advertised, 6 speeds, which are fully adequate for any vehicle of sufficient power for the weight to be moved.

The great difficulty with auxiliary transmissions, however, is that, in any two-lever shift, the driver is far too apt to manipulate the levers unintelligently. Observation of drivers on the job showed that, in the majority of cases, the auxiliary is used for freak combinations of ratios where the straight

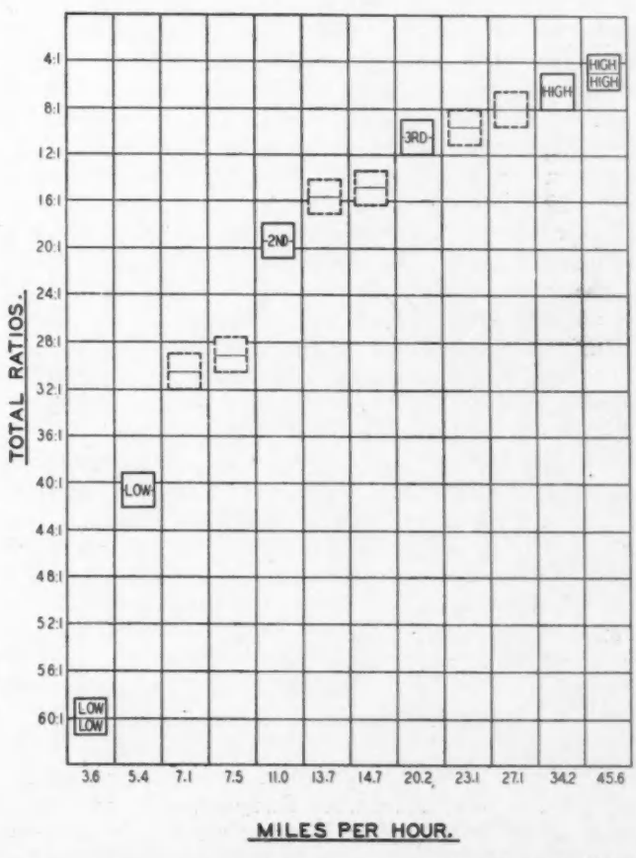


Fig. 3 – Diagram of Shifts – Four-Speed Plus Three-Speed Auxiliary – Showing Useful Ratios Provided

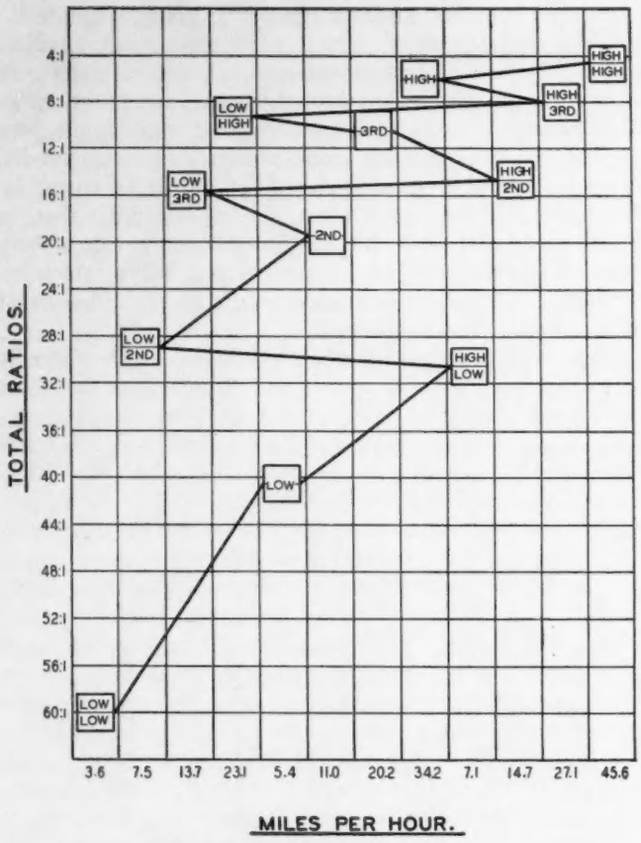


Fig. 4 – Diagram of Shifts – Four-Speed Plus Three-Speed Auxiliary – Showing Three Separate Ranges and Progressive Shifts Through Full Range

ratio in the main box would serve the purpose better. Fig. 4, for example, shows the three ranges in this combination as they would occur were a driver to go through the four main speeds in the low auxiliary first; then repeat with the auxiliary in direct position; and finally go through once more with the auxiliary in the over-geared position. The zig-zag line, incidentally, indicates the progression in order of ratios.

This subject introduces the next consideration in performance which, in many cases, is overlooked by engineers intent only upon providing a plentiful selection of ratios. This consideration is ease of shifting with which, of course, is bound up the matter of the speed with which shifts may be made. This consideration has been a major factor in the popularity of auxiliary transmissions and two-speed rear axles, although not generally appreciated as such either by designers or operators. An auxiliary transmission or the rear-axle shift is usually easily made because the steps are small and generally the two speeds of the gears are low. This was unquestionably one factor which militated against the two-speed countershaft arrangement which was formerly so popular, since the countershaft driving pinions always operate at full crankshaft speed. Another limitation of the two-speed countershaft was that, whereas the auxiliary at the rear actually doubles the number of gear changes, the two-speed countershaft afforded two direct drives, which has no gain.

A great deal has been accomplished in the ease and speed of shifting by the employment of constant-mesh gears with internal-gear shifting clutches. Syncromesh shifting, which has been such a boon to passenger-car drivers, as yet has made no headway in the truck field, although encouraging results are being realized in bus development. It is logical that the greatest emphasis up to the present time should have been placed upon the ease and speed of bus shifting, both because of the enormous number of gear changes involved in a bus-day's work, and also because of the desirability of maintaining a maximum acceleration rate against the frequency of stops for the benefit of faster schedules.

These then are the factors with which we have to conjure in constructing a logical doctrine with respect to performance and, from careful consideration, we may secure a conception of their proper relations. It is clear that the basic limitation of performance is rear-wheel horsepower in its relation to gross vehicle weight, which we may express, for convenience, as gross weight per horsepower. Regardless of all else, a given number of pounds per horsepower will produce a certain coefficient of performance, regardless of whether such a relationship is the result of more horsepower and light weight, or great horsepower and heavy weight. From the laws of inertia, we also may deduce that, for a given pounds per horsepower, equal ranges and intervals of gear changes will give equal results, except as affected by tire radius, which is really a part of the gearing of the truck.

In an attempt to reduce these considerations, or at least a part of them, to a form which would permit the purchasers and operators of motor trucks to select equipment with greater ease and intelligence to fit the actual conditions and requirements of operation, repeated attempts have been made to formulate an equation by which, from information readily available, reliable prognostications of specific performance might be obtained. The development of these formulas was traced in detail in the author's paper, "Fundamentals of Vehicle Performance," which he read at the Summer Meeting of the S.A.E. on May 6, 1937.[2] Without rehearsing all of this material in detail, it may be said that all performance ratings at the present time are based upon the tractive factor formula, originated by the late Cornelius T. Myers, being carried out to show the result in terms of ability factor, grade ability in high gear, grade ability at various speeds, or the S.A.E. Performance Factor.

Once more this entire matter of a performance rating, if I am informed reliably, is to be taken up by the Society of Automotive Engineers and, in closing these remarks, I wish to adjure every member of the S.A.E. who is interested in any way in the profitable operation or production of highway transport vehicles or equipment – to cooperate in every way possible. I do not yet know into whose capable hands this important mission will be placed; but it is my earnest hope that, when the work shall have been completed, the S.A.E. will present to the world a formula for the rating of motor-vehicle performance which will fulfill these requirements:

1. That it shall be based soundly upon the facts in so far as they may be determinable, and not upon traditional theory.

2. That it shall be simple, both in final coefficient of performance which it yields and in its forms of expression. That it shall be divested of all needless complication of calculation.

3. That it shall be flexible so that, in the future when changes in prevailing conditions or present mechanical limitations shall occur, the formula may be adapted to such changes by the simple process of altering the constants currently applicable while preserving the essential form and principle.

4. That it shall employ as factors in its equation only such quantities and measures as are unequivocal, readily ascertainable, and subject to positive verification.

5. And finally, that it shall constitute nothing more than a yardstick for measurement, accurate within commercial and practical limits, but in itself incapable of constituting standards of practice or of incorporation into law or regulation.

[2] See S.A.E. JOURNAL, August, 1937, pp. 13-19: "Fundamentals of Motor-Vehicle Performance," by Merrill C. Horine.

The Single Sleeve as a Valve Mechanism for the Aircraft Engine

By A. H. R. Fedden
The Bristol Aeroplane Co., Ltd.

THERE is no reason, Mr. Fedden predicts, why single-sleeve-valve aero engines should not eventually supplant completely existing types equipped with poppet valves with their inherent fundamental problems, especially for military and commercial craft.

The single sleeve valve has now passed from the experimental to the production stage with hundreds flowing through the Bristol factory, and the author sees ahead a vast field of research yet to be explored.

Among the nineteen inherent advantages claimed for the sleeve-valve engine over the poppet-valve type by Mr. Fedden are quieter operation, higher compression ratio with its attendant benefits, absence of maintenance except for plugs and magneto servicing, and complete enclosure of all working parts. Data comparing the two types are given in detail.

The paper starts by giving a historical background of the sleeve-valve internal combustion engine from 1905 when Charles Y. Knight first began work; then carries on with the development by different groups until the Bristol Co. entered the field.

Then follows a résumé of Bristol single-cylinder research, and subsequent main-engine development, giving particulars of type tests together with flight-testing of the Bristol sleeve-valve engines. The author announces that more than 10,000 hr. of main-engine running and flying have been attained on the Bristol sleeve-valve types to date.

THE sleeve-valve internal-combustion engine has an interesting historical background for the patent record goes back to 1908, although 1905 is accepted as the date when Charles Y. Knight of Chicago began his work on the subsequently famous double-sleeve-valve engine.

Although the honor of originating this novel design rests with the U.S.A., most of the attempts to interest the American automobile industry failed, and Great Britain – the Daimler Co. in particular – must be given the credit for pioneering this type, and for providing the financial and technical backing that was necessary to bring the sleeve-valve engine to a front-rank position in the automobile world.

In the spring of 1909, after completing successfully under official observation what was, in those days, an outstanding performance comprising a long continued endurance bench test of over 100 hr., followed by a 2000-mile trial on Brooklands Track, the Daimler Co. presented three standard models fitted with the Knight-type double-sleeve-valve engine at the London Motor Show that same autumn; and, subsequently, other established Continental automobile manufacturers followed the lead taken by the Daimler Co.

The success of the early Knight engine stimulated a plentiful crop of new ideas on the subject of positively operated valves and, of these, the outstanding solution was the single-sleeve-valve type for which patent application was made in 1909 by a Scottish engineer, Peter Burt, in conjunction with Argylls, Ltd., which was, at that time, a well-known motor-car manufacturing company near Glasgow, Scotland.

Almost simultaneously, an application of independent origin was lodged by a Canadian engineer, James McCollum, covering a sleeve-valve engine with the important feature of combined reciprocating and rotary motion.

Arrangements were made for an agreement between McCollum and the Argyll Co., and the single-sleeve-valve design of engine has been generally referred to since as the Burt-McCollum type.

Although the double sleeve valve preceded the single sleeve for road vehicle powerplants, both in origin and practical application, the first sleeve-valve aircraft engine was of the latter type, and was built by the Argyll Co. in 1913-1914 for the Naval and Military Aeroplane Engine Competition organized by the British Government in 1914. This engine was an in-line water-cooled six-cylinder, of 785 cu. in. capacity, with a maximum output of approximately 130 b.hp. It created an excellent impression despite continual trouble due to the crankshaft but, with the outbreak of the World War in August, 1914, this development work was shelved due to the necessity for the diversion of all available energy and material to the production of established models and, apart from an experimental adaptation of a twelve-cylinder 140-hp. V-engine produced by the original Royal Aircraft Establishment, then known as the Royal Aircraft Factory, no further work was carried out on this type of engine until after the Armistice.

In the years immediately after the War, there was widespread activity in the motor-car field with sleeve-valve engines

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 13, 1938.]

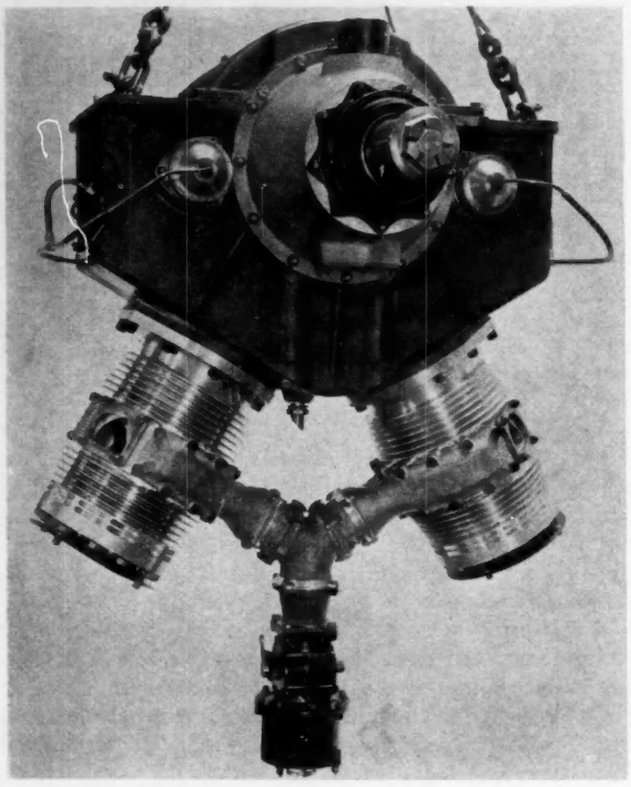


Fig. 1 – First Bristol Single-Sleeve Research Unit

of both types, and aircraft engines employing the double sleeve were produced by Panhard in France, and Minerva in Belgium, whereas the development of the single-sleeve type aircraft engine showed no progress until 1925. In that year the Continental Motors Corp., of Detroit, U.S.A., acquired the Burt-McCollum patents, and started in on the development of an aero engine employing this principle of valve mechanism and, early in 1927, this concern exhibited a nine-cylinder air-cooled single-sleeve radial engine of 787 cu. in. capacity, developing 220 hp., at the New York and Chicago shows.

The interest of the British aero-engine industry in the single-sleeve-valve engine was stimulated by the 1925-1926 report of the Aeronautical Research Committee, which referred in warm terms to the experimental work carried out by H. R. Ricardo, and drew attention to the definite advantages offered by this type of valve mechanism for aero engines.

By 1926 the Bristol Aeroplane Co. had consolidated a long-standing opinion that cylinder bores of over 5 in. were necessary to obtain the optimum size of radial engine in single-bank form for powers up to 1000 hp. and, owing to its experience up to that date on the four-valve head, was convinced that such mechanism was preferable to the two-valve type, of which it had some detail knowledge, for military requirements, and for the types of fuel available in England. At this date it became increasingly apparent that, although the two-valve head could be made lighter and more simple and it was free from the annoying and undesirable oil-leakage troubles, it would become a difficult problem to make the four-valve mechanism work satisfactorily and yet provide a relatively simple solution with enclosed rocker gear, except at a prohibitive increase in complication. Further, it was felt highly desirable to retain the compensation device then standardized on the Bristol four-valve design.

It was decided that a determined effort to find an alternative to the poppet valve must be made, and the Cuff type, rotary valve, and the double and single sleeve valve were all investigated. The rotary valve was discarded from considerations of operating mechanism and lubrication; the Cuff valve also suffered from the latter failings and from limitations of port area and combustion-chamber design.

There is a great difference between the two types of sleeve-valve operation and, whereas the double sleeve disappeared from motor-car powerplants when higher duty engines became standard practice owing to distortion, lubrication problems, and difficulties of operation, the single sleeve has, in my opinion, a vastly greater field, and it is submitted that it is misleading to judge its possibilities on the existing bias formed against the Knight type in automobile practice.

The single-sleeve mechanism of the Burt-McCollum design always has been attractive to the internal-combustion engineer and, although no serious work on air-cooled sleeve-valve engines, particularly of such large bore, had been undertaken at the time under discussion, namely ten years ago, a certain amount of preliminary research had been carried out by H. R. Ricardo, and the Barr & Stroud Co. of Glasgow had produced quite promising air-cooled motorcycle engines based on this principle, but which it had been forced to drop through lack of funds to cover such an expensive and fundamental development.

Further, the technique of sodium-cooled valves, originated at the Royal Aircraft Establishment, Farnborough, and subsequently developed in later years in the U. S. A.,[1] was still in its embryo state, and the excellent forging technique on hollow stems and heads, which also was brought successfully to fruition in America, was not then visualized as a practical production proposition in England.

As a result of this broad review of the requirements and possibilities of the breathing mechanism of large air-cooled aero-engine cylinders, which occupied some months, the Burt-McCollum, in spite of all its vicissitudes, emerged as the most promising and, toward the end of 1926, the Bristol Aeroplane Co. decided to commence experimental work along the line of the single-sleeve valve as a "long-term" research, and designed a twin-cylinder air-cooled unit. Thereafter, the company proceeded with experimental and research work on the various problems involved, from both the design and the manufacturing aspects of such an engine, on a reasonable priority, being determined to keep an open mind on the problem and to be prepared to lose several years' work if the promise given by the preliminary investigations was not fulfilled. In this work it was assisted greatly by the technical encouragement and financial assistance given by the British Air Ministry – in the first instance by the Directorate of Scientific Research and, subsequently, by the Directorate of Technical Development.

Many real difficulties were encountered and, in view of the varied early setbacks, it is only fair to say that it is somewhat problematical if the Bristol Co. could have pursued this work to a satisfactory conclusion without the assistance of the British Air Ministry.

Throughout these early teething troubles, the most encouraging aspect was the continued faith in, and enthusiasm for, the single-sleeve-valve mechanism as shown by all the Bristol staff who participated in this experimental work.

For the first four years efforts were concentrated on single- and two-cylinder unit research running, and an extensive

[1] In prepared discussion following presentation of this paper (see page 363, this issue) S. D. Heron contended that this statement is incorrect and that the sodium-cooled valve originated at McCook Field, U.S.A., and not at Farnborough, England, as stated by Mr. Fedden. In reply, Mr. Fedden retracted his statement, explaining that, in view of Mr. Heron's intimate connection with this work, he did not question the accuracy of his statements.

background was accumulated. Designs for a complete engine were then put in hand, culminating in the building of the first Bristol "Perseus" in 1932, this engine being a nine-cylinder air-cooled radial of 1520 cu. in. capacity on which, in the same year, a British Air Ministry 100-hr. service type test was completed successfully at a normal rating of 515 hp., with a maximum output of 638 hp., after a relatively short period of development running.

Subsequently, a small batch of these engines was produced for the British Air Ministry and, after extensive flight trials lasting some hundreds of hours of this basic engine by the Bristol Co. in military aircraft, five of these engines were loaned to Imperial Airways for installation in one of their Short "Scylla" 32-passenger, 32,000-lb. four-engined air liners. June 29, 1935, when the Perseus engines were put into commission on the London-Paris passenger-transport service, is a date worthy of record in the history of aircraft engines.

After a total of some 2500 hr. running on these five engines, during which they were transferred to another aircraft owing to a mishap to the first when on the ground in a gale, whereas the second aircraft was wrecked ultimately in the course of night flying operations – the accident not being in any way due to the engines – this flight-test work was abandoned as the engines were at that date superseded by a later and improved design. They had, however, "made their mark" as the commercial engines of the future, and the technical staff of Imperial Airways paid striking tribute to the lack of maintenance required, to their smooth and satisfactory running, and to their economy of fuel and oil.

By the end of the present year a new fleet of Imperial Airways' four-engined machines will be equipped with the latest series of Bristol Perseus engines.

In the meantime, the development of other Bristol designs, including double-row engines, was proceeding and, at the Paris Salon of November, 1936, the Bristol Co. was able to exhibit the "Hercules," a fourteen-cylinder double-bank sleeve-valve engine of 2360 cu. in. capacity, type-tested at an output of 1375 hp.

Other and larger engines are to follow, but the single sleeve valve now has passed from the experimental to the production stage, and today some hundreds are flowing through the Bristol production factory.

Following this short historical outline, I propose to deal in some detail with the research work at Bristol which has been carried out to date, the main engine development, and the most important advantages which it is considered are inherent to the single-sleeve-valve aero engine.

Single-Cylinder Research Work

The first single-sleeve research unit was constructed and run toward the end of 1927.

Fig. 1 shows the first Bristol unit, which took the form of a vee-twin, being in effect one-sixth of a proposed inverted twelve-cylinder engine layout, it having been designed in one of the recurring periods when the in-line air-cooled engine came into the picture and before the N.A.C.A. had consolidated its work on the cowling of radial engines. The bore and stroke were 5¾ in. x 6 in.

Considerable experience was obtained with this unit during 1928, and the then relatively high output of 135 lb. per sq. in. at 2000 r.p.m. was obtained. Difficulties were experienced, however, with nickel cast-iron sleeves through cracking and blowing out, and attempts to use steel cylinder barrels with semi-steel sleeves were unsuccessful due to high local temperatures.

The general problems became clear from this preliminary running in that they lay mainly with cylinder-head cooling and sleeve-material technique.

After due consideration, the unit was rebuilt in 1930 with forged KE.965 steel sleeves and aluminium-alloy barrels and heads, a different design of cylinder head, and generally increased cooling as indicated by the preliminary tests. One of the sleeves was run in the soft condition, whereas the other was nitrided.

Tests soon showed that the soft sleeve was unsatisfactory, but extremely promising results were obtained with the nitrided component. The design modifications had improved the cooling very considerably, some 50 deg. cent. reduction in head temperature being realized. A power output of 145 lb. per sq. in. b.m.e.p. at 2000 r.p.m. was obtained finally, but trouble was experienced with the crankcase. The unit was rebuilt again in 1931 and, as the large-bore in-line design of engine had been abandoned subsequently, it was run in single-cylinder form. During this year other sleeve-valve single-cylinder units incorporating these improvements were added to the Research Department equipment.

After some preliminary testing with this modified design, a 100-hr. endurance test, without stripping down or adjustment, was completed satisfactorily in March, 1931, at 2000 r.p.m. 120 lb. per sq. in. b.m.e.p. The successful realization of this test was a big step toward providing the final conviction that the single sleeve valve could be made satisfactorily in air-cooled form, and continuous research has proceeded since that date with every confidence and on a high priority.

Fig. 2 shows, in composite form, the advances made from year to year in terms of power output, b.m.e.p., and fuel consumption, which have been realized at Bristol on the 5¾-in.

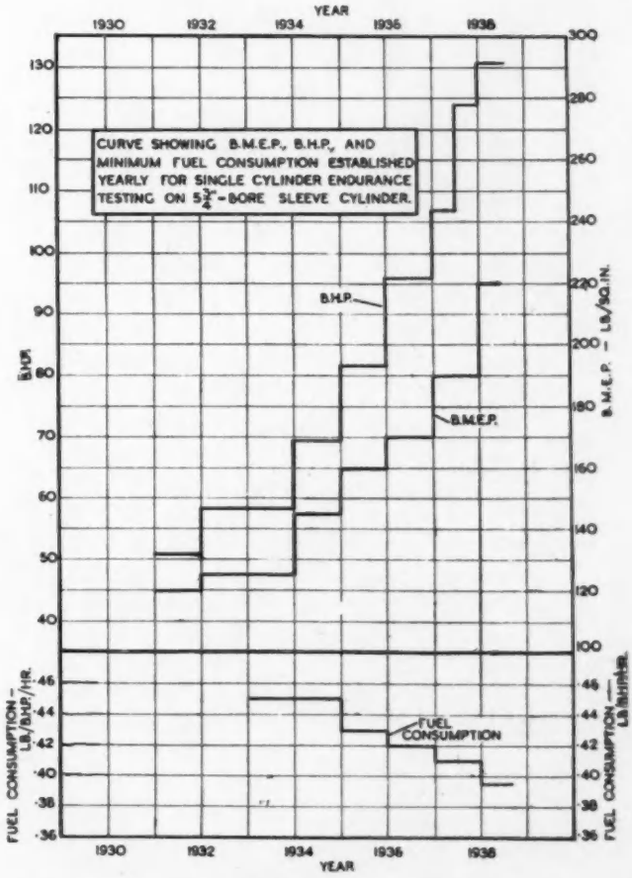


Fig. 2 – Advances in B.M.E.P., B.Hp., and Minimum Fuel Consumption for Single-Cylinder Endurance Testing on 5¾-In. Bore Sleeve Cylinder

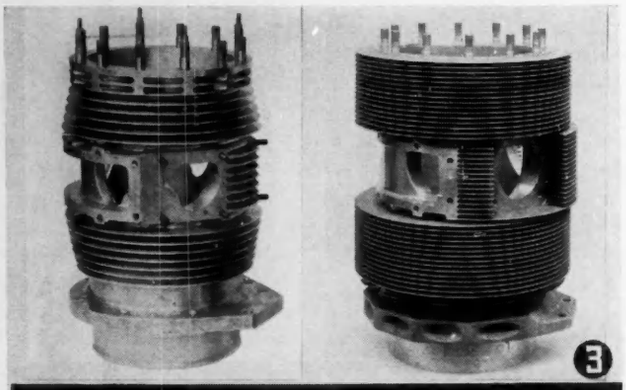


Fig. 3 – Original Aluminum Cylinder Barrel (*left*) beside Present Perseus Production Type

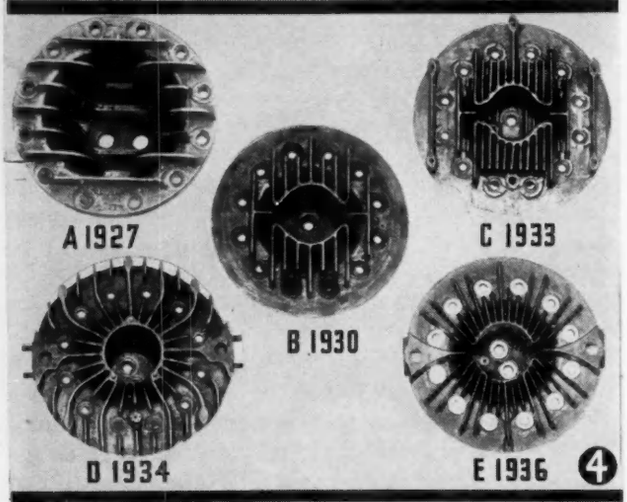


Fig. 4 – Stages in the Development of the Cylinder-Head

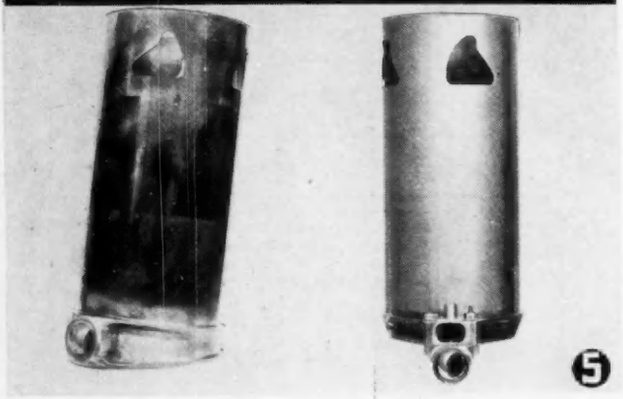


Fig. 5 – Original Nitrided-Steel Sleeve (*left*) beside Sleeve from Latest Production Batch

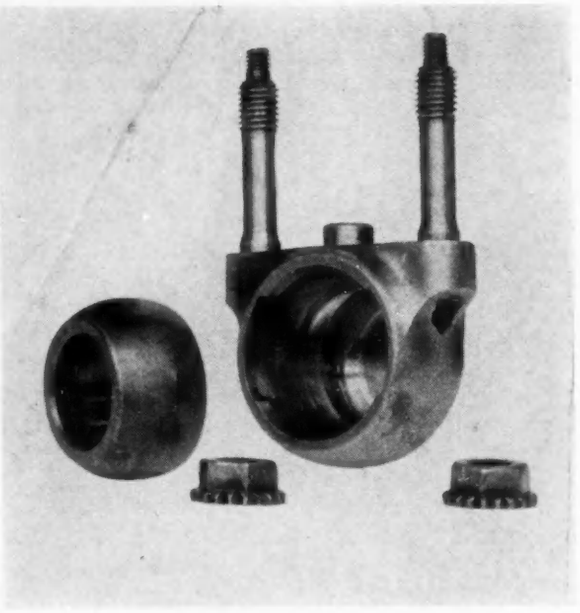


Fig. 6 – Standard Driving-Ball Assembly

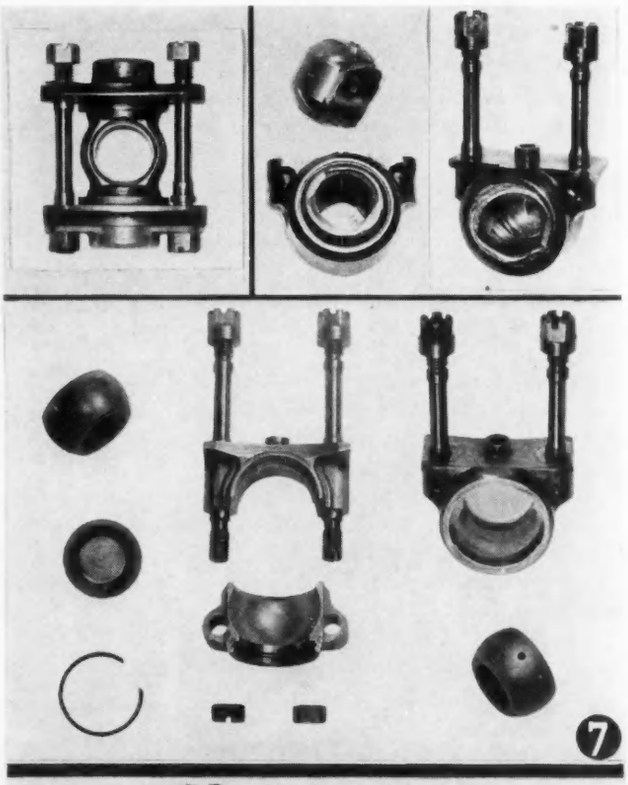


Fig. 7 – Experimental Types of Sleeve-Valve Drives

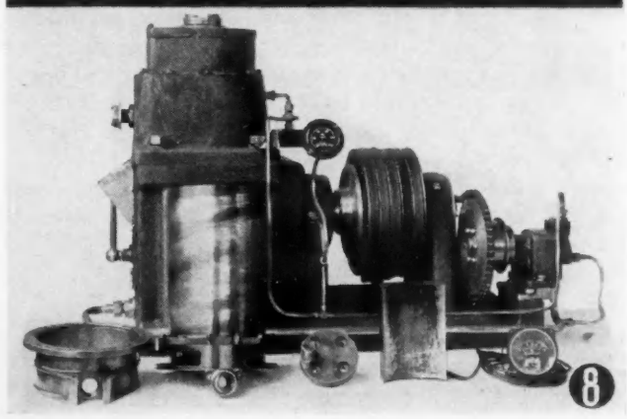


Fig. 8 – Rig for Testing Sleeve-Valve Layout

bore single-cylinder units. It should be appreciated that, in each of the three conditions shown on this figure, a standard of 100 hr. running without adjustment or breakdown has been established and, in no case, are they snatch readings taken for short periods.

It is now proposed briefly to touch on the main components forming the assembly.

Fig. 3 shows the original aluminum cylinder barrel on the left in comparison with the present production Perseus type.

Research on this component has led to improved cooling, adequate porting arrangement, and suitable proportioning to maintain parallelism and circularity under all running conditions.

Fig. 4 shows stages in the development of the cylinder-head. This component presents rather a different cooling problem than that of any poppet-valve arrangement in that cooling air has to be taken down inside the wall of the head to the crown. The original design provided plain fins and a simple deflector.

The first big advance was made in the rebuild of 1930, where consideration of the problem led to the incorporation of a fin arrangement giving an integral baffle together with

CYLINDER TYPE.		PERSEUS	MERCURY
VALVING.		SLEEVE	4 POPPET PUSH ROD
BORE & STROKE.		$5\frac{3}{4}$" x $6\frac{1}{2}$"	$5\frac{3}{4}$" x $6\frac{1}{2}$"
DISPLACEMENT.		168·8 CU.INS.	168·8 CU.INS.
CYLINDER ASSEMBLY WEIGHT.		42·1 LB.	46 LB.
COOLING FIN AREA.		2970 CU.INS.	2364 SQ.INS.
SUSTAINED PEAK POWER NATURALLY ASPIRATED 87 OCTANE FUEL.	B.M.E.P. LB/SQ.IN.	138	130
	R.P.M.	3200	3000
	B.H.P./CU.INCH.	·557	·492
SUSTAINED POWER FOR 100 HOURS 100 OCTANE FUEL.	B.M.E.P. LB/SQ.IN.	220	185
	R.P.M.	2800	2750
	B.H.P./CU.INCH.	·778	·642

Fig. 9 – Comparative Data of Bristol Cylinders

directed circulation for the crown of the head, and turbulent circulation within the baffle around the spark-plugs. This general layout has been adhered to ever since, and has progressed in the stages shown to the final form with radial finning giving circularity under running conditions, together with cooling adequate for all power outputs so far attained.

It is interesting that this head might be claimed as the forerunner of complete pressure baffling in that the whole layout was based on attaining a velocity head at the upper front section and allowing the air to leak through finned cells to the lower pressure air at the rear.

The present head shown at *E* in Fig. 4 is of die-cast Y-alloy, and is an excellent example of this method of manufacture.

The sleeve, as shown in Fig. 5 – the one on the left-hand side being an example of the original nitrided-steel series as compared with that on the right-hand side being from the latest production batch – apparently has not altered greatly but, nevertheless, it represents many detail changes, many years of patient development work, and some thousands of hours running and, in conjunction with the cylinder, increase of port area has been provided, and control of the swirl has been obtained.

The piston, a light-alloy drop forging, has followed normal Bristol lines and, similar to the practice in the "Pegasus" engine, carries two compression rings and two scraper rings, but this component also has been the subject of many thousands of hours development work.

The standard driving-ball assembly is shown in Fig. 6.

Certain experimental types which have been tested out are shown in Fig. 7.

This component has to be 100 per cent reliable and, owing to the length of time necessary and the arduous nature of proving out this unit, a test rig was employed as shown in Fig. 8, and considerable design and research effort has been spent in obtaining the present successful type, which fortunately also has proved to be the simplest which was tried.

On rig tests it has been found possible to transmit over 4 hp. through this component and, due to suitable detail design, the selection of materials, and the nature of the motion, marked absence of wear exists on the production series.

Fig. 9 shows a tabulation of the horsepower which it is possible to maintain for a full-throttle 100-hr. life without distress with the weights and fin areas of these cylinders – the "Mercury" being the Bristol four-valve poppet type, and the Perseus the Bristol single sleeve valve, both of the same cubic capacity, 168.8 cu. in., $5\frac{3}{4}$ in. bore by $6\frac{1}{2}$ in. stroke.

Fig. 10 shows comparative power curves of the Mercury and Perseus units taken at full throttle under naturally aspirated conditions at various r.p.m., and adjusted to take full advantage of the fuel.

Fig. 11 shows the comparative breathing efficiencies of a modern two-valve American poppet valve, the Bristol Mercury four-valve, and the Perseus sleeve valve.

Fig. 12 shows photographs of the Bristol Mercury four-valve poppet-valve cylinder (center), a typically modern American all-enclosed two-valve (left) – given by the courtesy of the Wright Aeronautical Corp. – and the Perseus sleeve-valve cylinder.

Fig. 13 shows the cylinder weights and displacement of the previously mentioned representative poppet-valve and sleeve-valve cylinders.

The first research work on a size of cylinder smaller than the $5\frac{3}{4}$-in. Bristol standard was commenced in 1931, when a small cylinder of $4\frac{1}{4}$-in. bore x $4\frac{1}{2}$-in. stroke was built into a single bank of six cylinders, with a view to ascertaining its

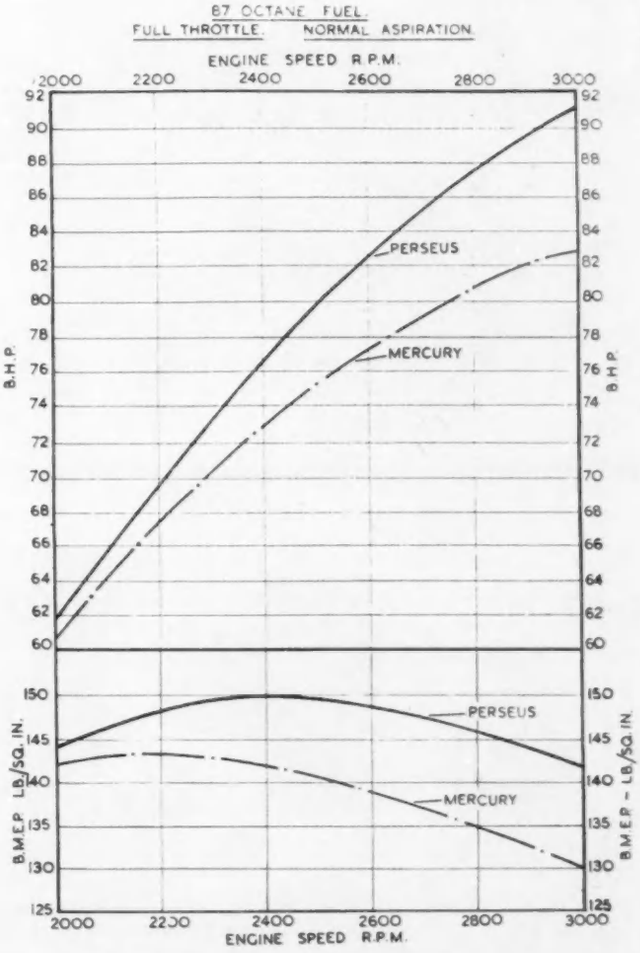


Fig. 10 – Comparative Power Curves for Perseus and Mercury Units – Compression Ratio Adjusted To Give Similar Detonation Characteristics

possibilities for a V-12 or cross engine. Two views of this unit are shown on Fig. 14. Research work proceeded on this experimental unit for two years, but was dropped eventually in favor of a return to the radial engine for high power, on which engine it was felt necessary to employ the largest bore of cylinder that conveniently could be built into such a design.

Work also was commenced in 1933 on a 5-in. bore cylinder of 5⅜-in. stroke, it being intended to use this cylinder for a small nine-cylinder 350-hp. civil engine. The early power outputs obtained with this cylinder were, however, so promising that the engine – subsequently known as the "Aquila" I – was type-tested in 1934 at a normal rating of 420 hp. with a maximum power of 500 hp. on 74-octane fuel. The cylinder referred to is shown in Fig. 15 (left) side by side with a Perseus cylinder, and was developed originally on 74-octane fuel with marked success.

Although main-engine work on the Aquila size has been suspended largely owing to British re-armament needs, research work on the preparation of this cylinder, in its existing and slightly modified size, has proceeded on a reasonable priority with a view to its incorporation in small-diameter double-row engines.

Fig. 16 shows power curves of the Aquila size cylinder, naturally aspirated, with 87- and 100-octane fuels, the com-

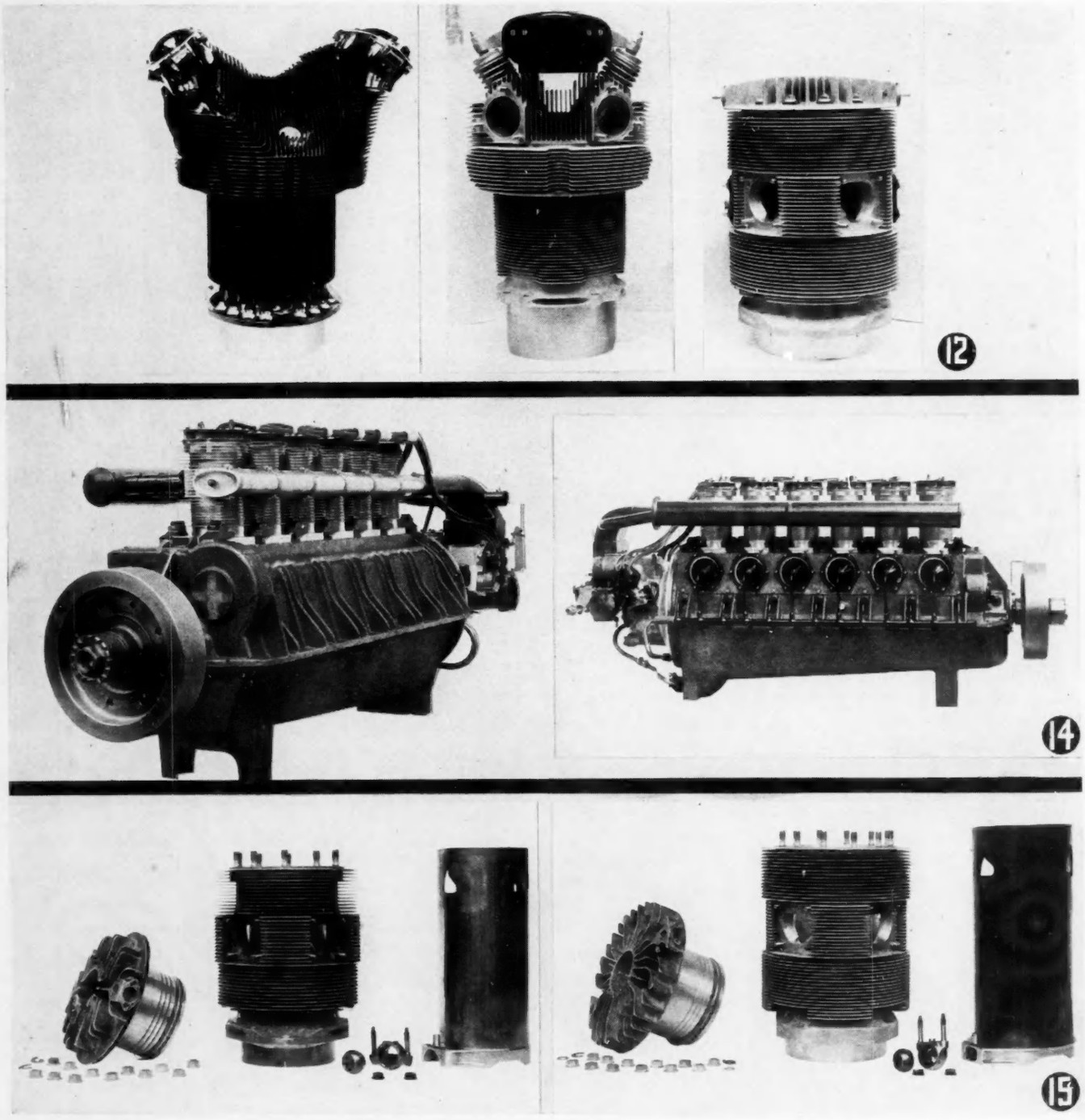


Fig. 12 – Modern American Two-Valve Poppet-Valve Cylinder (*left*), Bristol Mercury Four-Valve Poppet-Valve Cylinder (*center*), and Perseus Sleeve-Valve Cylinder

Fig. 14 – Experimental Unit Comprising a Single Bank of Six 4¼ In. by 4½ In. Cylinders

Fig. 15 – 5 In. by 5⅜ In. Aquila Cylinder (*left*) beside a Perseus Cylinder

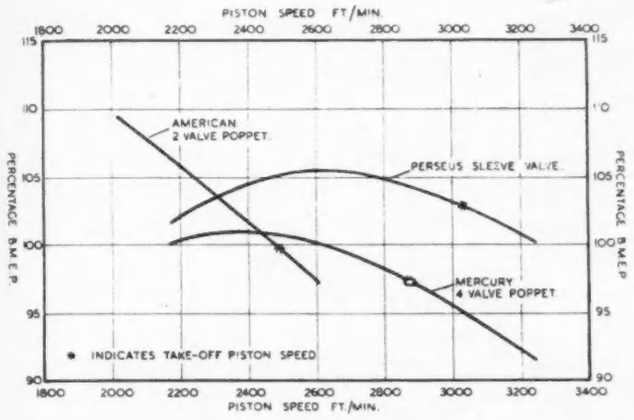


Fig. 11 – Comparative Breathing Efficiencies of Two-Valve, Four-Valve, and Sleeve-Valve Cylinders

pression ratio being adjusted to give similar detonation characteristics for main-engine operating conditions at appropriate ratings.

A year or two previous to the development of the 5-in. bore sleeve-valve cylinder, the Bristol Co. had embarked upon a considerable amount of research work on a 5-in. bore two-valve cylinder with overhead camshaft. A complete double-row engine actually was made and tested, two views of which are shown on Fig. 17, and exhaustive single-cylinder tests

CYLINDER TYPE.	PERSEUS.	MERCURY.	AMERICAN.
VALVING.	SLEEVE.	4 POPPET PUSH ROD.	2 POPPET PUSH ROD.
BORE & STROKE.	$5\frac{3}{4}$" X $6\frac{1}{2}$"	$5\frac{3}{4}$" X $6\frac{1}{2}$"	$6\frac{1}{8}$" X $6\frac{7}{8}$"
DISPLACEMENT.	168·8 CU. INS.	168·8 CU. INS.	202·6 CU. INS.
WEIGHT.	42·1 LB.	46·0 LB.	47·576 LB.

Fig. 13 – Cylinder Weights

were carried out with cylinders having single and double overhead camshafts. The volume of this cylinder, namely, 98.2 cu. in., was selected carefully to allow the maximum efficiency for two valves at high speed, and modern technique was incorporated in regard to valve operation mechanism, and sodium-cooled valves.

Fig. 18 shows four views of the single and double overhead valves and cylinders. Although this engine provided some interesting and valuable data, this overhead camshaft cylinder was superseded finally by the sleeve-valve type which the Bristol Co. felt held greater promise of future development.

These few details of the Bristol overhead-camshaft engine are given because the bore is the same as the smaller size sleeve-valve cylinder which the Bristol Co. later developed, and Figs. 19 and 20 provide interesting data showing comparative power curves of the 98.2 cu. in. poppet-valve cylinder and the 105.6 cu. in. sleeve-valve cylinder. It will be noted that Fig. 19 gives comparative power curves with the compression ratios adjusted to give similar detonation characteristics, whereas Fig. 20 gives comparative power curves at the same compression ratio.

Finally, Fig. 21 gives interesting comparative data between the two sizes of cylinder developed by the Bristol Co., operating on poppet valve and single sleeve valve. It is wished to emphasize again that the last two columns are not snatch readings.

Fuel-consumption loops obtained from the sleeve-valve type of cylinder, in both sizes, are remarkable for their stable shape. An example is shown in Fig. 22, which reflects favorably on the smoothness of main-engine running under extreme-economy conditions.

Altogether a total of some 26,000 hr. has been run by the Bristol Co. on sleeve-valve single-cylinder research to date and, although this work represents an achievement that has enabled the single-sleeve-valve engine to go into considerable production for the British Air Ministry, it is now apparent that there is a vast field of research on this form of valve mechanism as yet unexplored, and that much promising development can be expected confidently from this type of cylinder construction.

Main-Engine Development

As mentioned in the introduction, the first complete Bristol sleeve-valve engine, a nine-cylinder radial of 1520 cu. in. capacity, known as the Perseus I, was type-tested in 1932, after which five Perseus II engines of modified design were produced. Considerable setbacks were encountered during the development of the Perseus II type, due both to manufacturing difficulties and an incomplete understanding of the fundamental problems involved as the duty of these engines was increased. In the meantime, overload testing on the first Perseus engine was carried out successfully at 700 hp., still with very promising results, and the engine was endurance-tested as a fully ground-boosted engine of 610-hp. rating in a single-seater Bristol "Bulldog" aircraft, as shown in Fig. 23, being demonstrated at the Royal Air Force Pageant and S.B.A.C. Displays of 1934.

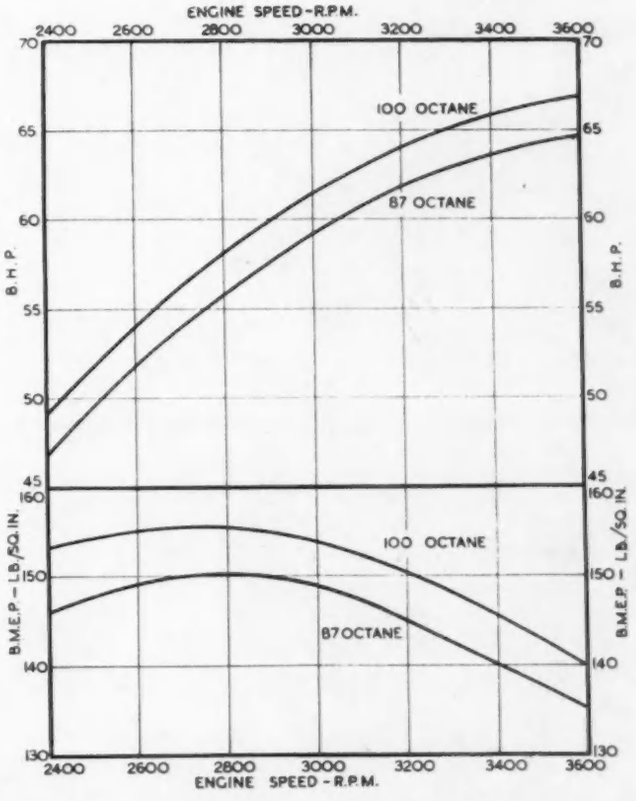


Fig. 16 – Power Curves of the Aquila Unit – Natural Aspiration with 87-Octane and 100-Octane Fuels – Compression Ratio Adjusted To Give Similar Detonation Characteristics for Main-Engine Operating Conditions at Appropriate Ratings

The Perseus II finally was type-tested in the year 1934, and this type test was followed by a 250-hr. cruising test at a power of 420 hp. at 2150 r.p.m., with a consumption of 0.435 lb. per b.hp-hr. The satisfactory condition of this engine at the conclusion of this run convinced the Bristol Co. that the engine had great possibilities for commercial operation, and it became apparent that the troubles being encountered by poppet-valve engines with ethylized fuels would be eliminated totally by the adoption of this type.

The five Perseus II engines referred to as being loaned by the British Air Ministry to Imperial Airways subsequently gave excellent service. Their flight trials were, however, as previously stated, interrupted by the complete loss of the "Syrinx" aircraft at Brussels aerodrome due to it being turned over in a gale while taxiing. This accident delayed the engine testing, but they were then put into a Vickers "Vellox" freight machine, which also was lost unfortunately in a night flying accident. Sufficient running had, however, been accomplished to demonstrate the reliable service to be expected from this type of engine in commercial operation, and the last engine received back at Bristol for strip had carried out 400-hr. running since the previous overhaul, at a power of 480 b.hp. with, of course, the usual periods of high power at full throttle for take-off and climb. This engine stripped in first-class condition and, had not the flight tests been terminated by the unfortunate accident to the Vickers' Vellox, it had been intended to step up to overhaul periods of 500 to 600 hr.

The Perseus III engine was substantially identical to the Perseus II, but was of an intermediate type rated at 5000 ft. instead of being ground-boosted as previously experimented with, and was intended for military trials. This engine also was type-tested successfully at the first attempt and, of the three engines of this type manufactured, one subsequently was flown in Canada in an adapted Hawker "Hart" aircraft, for winter trials of the sleeve-valve type. Operation under Arctic conditions was a bogy on which the Bristol Co. sought information at an early date as it was felt that, if undue difficulty was encountered, the universal usefulness of the engine might be affected somewhat adversely.

Through the courtesy of Joseph Lucas, Ltd., the Bristol Co. was enabled to carry out an extensive series of starting and torque tests in parallel on sleeve- and poppet-valve engines at a comparatively early date in the history of sleeve-valve development, in that company's cold chamber normally used for automobile starting tests, with interesting results as shown summarized on the curve on Fig. 24.

Although these cold-chamber tests alleviated the immediate apprehension on the possible starting difficulties of the sleeve-valve engine as a type, it was gratifying when the Canadian authorities reported favorably upon the absence of any real problem encountered with the operation of the Hawker Hart aircraft with Perseus III engine during two Canadian winters, and we were glad to know that our apprehensions were dispelled completely.

The encouraging results on the single-cylinder units with 87-octane fuel led the British Air Ministry to order a development batch of higher rated Perseus engines, known as the Perseus VIII for squadron trials, and this engine was successfully type-tested in 1936.

A squadron of Perseus Vickers "Vildebeests" then was

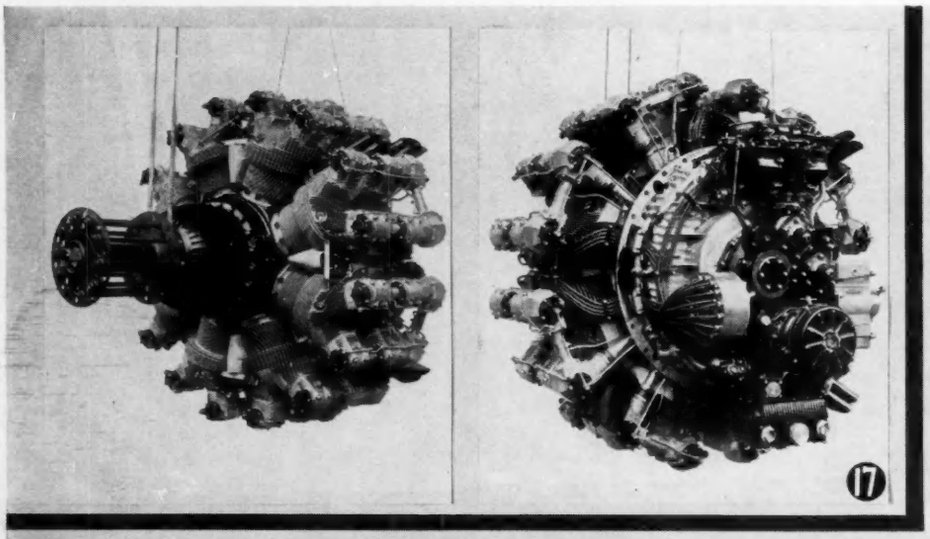


Fig. 17 – Double-Row Engine with 5-In. Bore Two-Valve Cylinders and Overhead Crankshaft

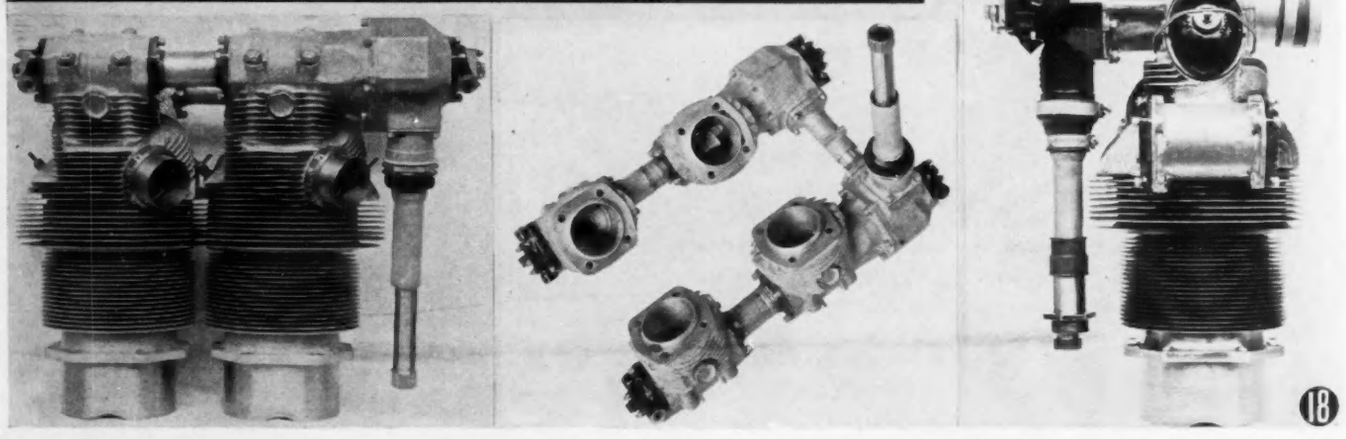


Fig. 18 – Single and Double Overhead Valves and Cylinders

ordered to provide service trials of the type of engine, the installation being shown in Fig. 25. This aircraft, although hardly of the most modern vintage, is one which has given excellent service to the Royal Air Force, and a large number of squadrons have been in commission throughout the Empire for a number of years with Bristol Pegasus poppet-valve engines. The Perseus Squadron, which has now been in service for over a year, has proved a thorough testing ground for the sleeve-valve engine, and has given the most satisfactory results as regards reliability, ease of maintenance, and continued serviceability, this latter feature being the most marked.

Subsequently, the Perseus V and VI engines were type-tested in 1936, and a number of modern aircraft were designed to utilize this powerplant. Many of these aircraft were put straight into large-quantity production owing to the inception of the British Air Force Expansion Scheme and, due

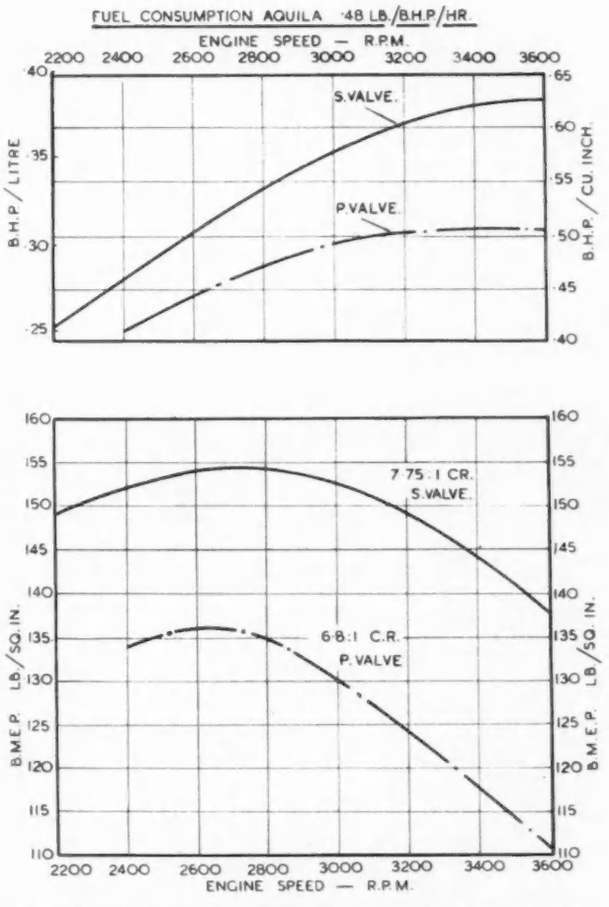


Fig. 19 – Comparative Power Curves for 98.2 Cu. In. Poppet-Valve Unit and 105.6 Cu. In. Sleeve-Valve Unit, the Compression Ratio Being Adjusted To Give Similar Detonation Characteristics – 87-Octane Fuel

to the time lag which then became inevitable before engines would be required, opportunity was taken to bring the specification of these engines in line with the new policy of the Bristol Co. This policy was to remove all aircraft accessories from the engine and drive them from a separate engine-driven gear box normally mounted on the aircraft bulkhead, leaving only those accessories essential to the operation of the engine still in position on the back cover. This policy has permitted of aircraft ancillary equipment being left in position when engines are changed, also facilitating rapid removal of the engine from the airframe.

Fig. 26 shows three views of the latest production series of

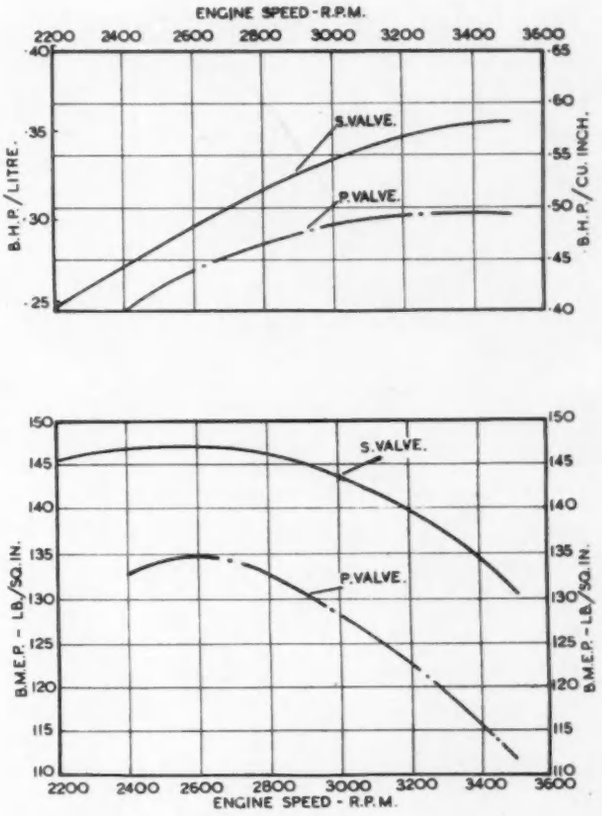


Fig. 20 – Comparative Single-Cylinder Power Curves for a 98.2 Cu. In. Poppet-Valve and a 105.6 Cu. In. Sleeve-Valve Unit at a Compression Ratio of 6.5:1 – 74-Octane Fuel

Perseus engine which is in considerable production at the present time for the Royal Air Force.

Fig. 27 shows details of the standard Perseus accessory gear box.

The Hercules, a double-row, fourteen-cylinder engine of 2360 cu. in. capacity, embodying the same cylinder size as the Perseus, now has completed two successful official type tests and a considerable amount of development testing. Two views of this engine are shown on Fig. 28. This engine was first type-tested in 1936 at a rating of 1375 hp. and 2400 r.p.m., and is the forerunner of other engines of this capacity at considerably higher power on both 87- and 100-octane fuel.

Endurance flight trials of the Hercules are proceeding in a special Northrop aircraft, photographs of the installation being shown on Figs. 29 and 30, and it is planned to be fitted to several new prototype aircraft this year.

CYLINDER TYPE & VALVING.		SLEEVE.	2 POPPET OVERHEAD CAMSHAFT.
BORE & STROKE.		5"X5⅞"	5"X5".
DISPLACEMENT.		105·6 CU.INS.	98·2 CU INS.
CYLINDER ASSEMBLY WEIGHT.		31·5 LB.	42·8 LB. DOUBLE CAMSHAFT. 44·2 LB. SINGLE CAMSHAFT.
COOLING FIN AREA.		1920 SQ. INS.	1660 SQ. INS.
SUSTAINED PEAK POWER NATURALLY ASPIRATED 87 OCTANE FUEL.	B.M.E.P. LB./SQ.IN.	138.	121.
	R.P.M.	3600.	3300.
	B.H.P./CU.INCH.	·628.	·505.
SUSTAINED POWER FOR 100 HOURS 100 OCTANE FUEL.	B.M.E.P. LB./SQ. IN.	240.	NO DEVELOPMENT ON 100 OCTANE FUEL.
	R.P.M.	3300.	
	B.H.P./CU.INCH.	1·0.	

Fig. 21 – Comparative Data of Bristol Cylinders

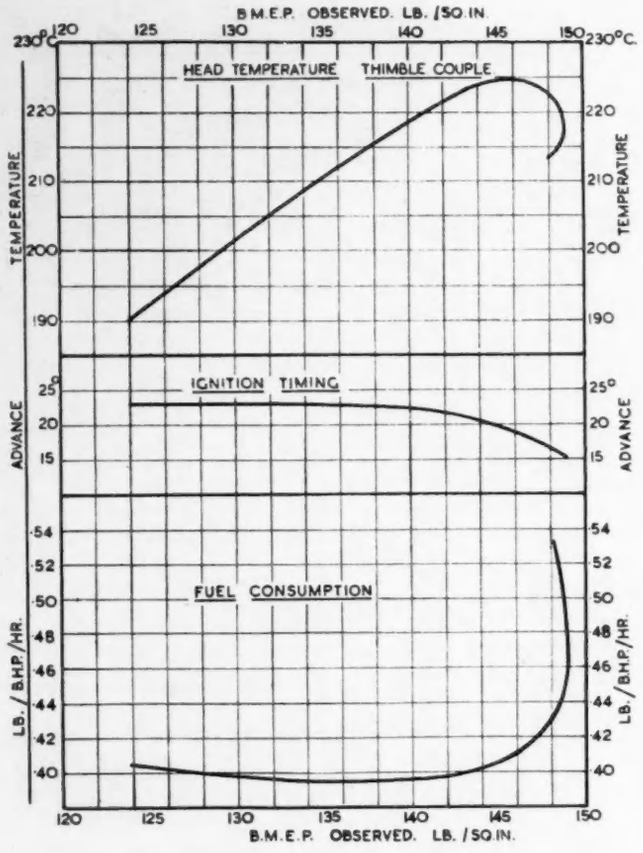


Fig. 22 – Perseus Unit – Fuel-Consumption Loop – Normal Aspiration, Variable Ignition Timing

Simultaneously with the development of the 5¾-in. bore cylinder, as outlined previously, the Bristol Co. commenced work on the Aquila engine of 950 cu. in. in 1933 and, after only 38 hr. of calibration work, carried out a type test on the Aquila I rated at 420 b.hp. normal, and 500 b.hp. maximum on 74-octane fuel, in 1934.

A successful 50-hr. overload test at 2650 r.p.m. and 500 b.hp. was then carried out, and two engines were submitted to some hundreds of hours of flight development but, owing to the need to devote our main attention to the requirements of the Royal Air Force, work on the 87-octane engine was suspended, but has just been taken up again with a type test at 500 hp., with a take-off power of 600 hp.

A considerable field can now be seen for this engine, which is shown on Fig. 31 (left) in comparison with the Perseus (center) and Mercury series and, on Fig. 32, installed in two modern aircraft – the Vickers "Venom" Interceptor Fighter (left), and the Bristol Type 143 civil aircraft, which latter machine has had to be held in abeyance owing to the exigencies of the British rearmament program.

Fig. 33 shows a view of the Aquila initial flight-test installation in the Bristol "Bullpup," a now obsolete fighter type which, nevertheless, proved very useful at the time for this relatively small engine, having a fuselage well suited to the engine's diameter.

The Bristol Co. has endeavored to adopt a conservative attitude in the initial stages of the sleeve-valve production, and the engine ratings given, although competitive, are by no means the maximum that can be obtained from the types which I have outlined.

Altogether a total exceeding 10,000 hr. main-engine running and flying has been attained on the Bristol sleeve-valve types to date.

Inherent Advantages

The advantages of the sleeve-valve engine as compared with the various poppet-valve types are numerous, and a number of them are given below:

1. Total absence of maintenance except for plugs and magneto servicing.

2. Complete absence of any hot-spot in the combustion-chamber and better temperature gradients over the whole cylinder, reducing distortion and stresses.

3. Use of higher compression ratio with consequent decrease in fuel consumption or, alternatively, of higher boost pressures and greater b.m.e.p.

4. Complete enclosure of all working parts with no possibility of oil leakage from the cylinder end, and absence of external oil leads to working end of cylinder.

5. Greater effective valve areas and reduction of restrictions to the gas flow. Better opening and closing diagrams, and improved volumetric efficiency.

6. More silent operation.

7. Freedom from hot lead corrosion as no parts of the cylinder reach temperatures which permit this corrosion and all, except the combustion space, are lubricated fully and continuously.

8. Cooler exhaust and, consequently, easier conditions for the exhaust manifold.

9. Centrally situated plugs giving optimum effective running on single ignition where necessary.

10. Any desired control of cylinder turbulence with its possible application to stratified charges and abnormally weak mixtures, with gasoline injection. It is felt that this may be a profitable avenue of development as recent improvements in variable-pitch propellers and increases in compression ratios form an important argument in favor of fuel injection,

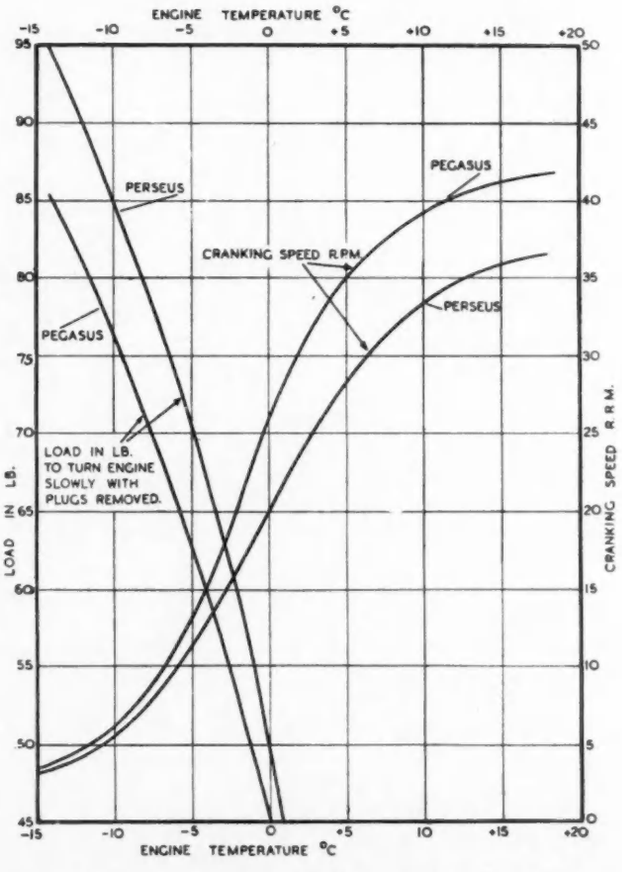


Fig. 24 – Comparative Starting Characteristics for Poppet-Valve and Sleeve-Valve Engines

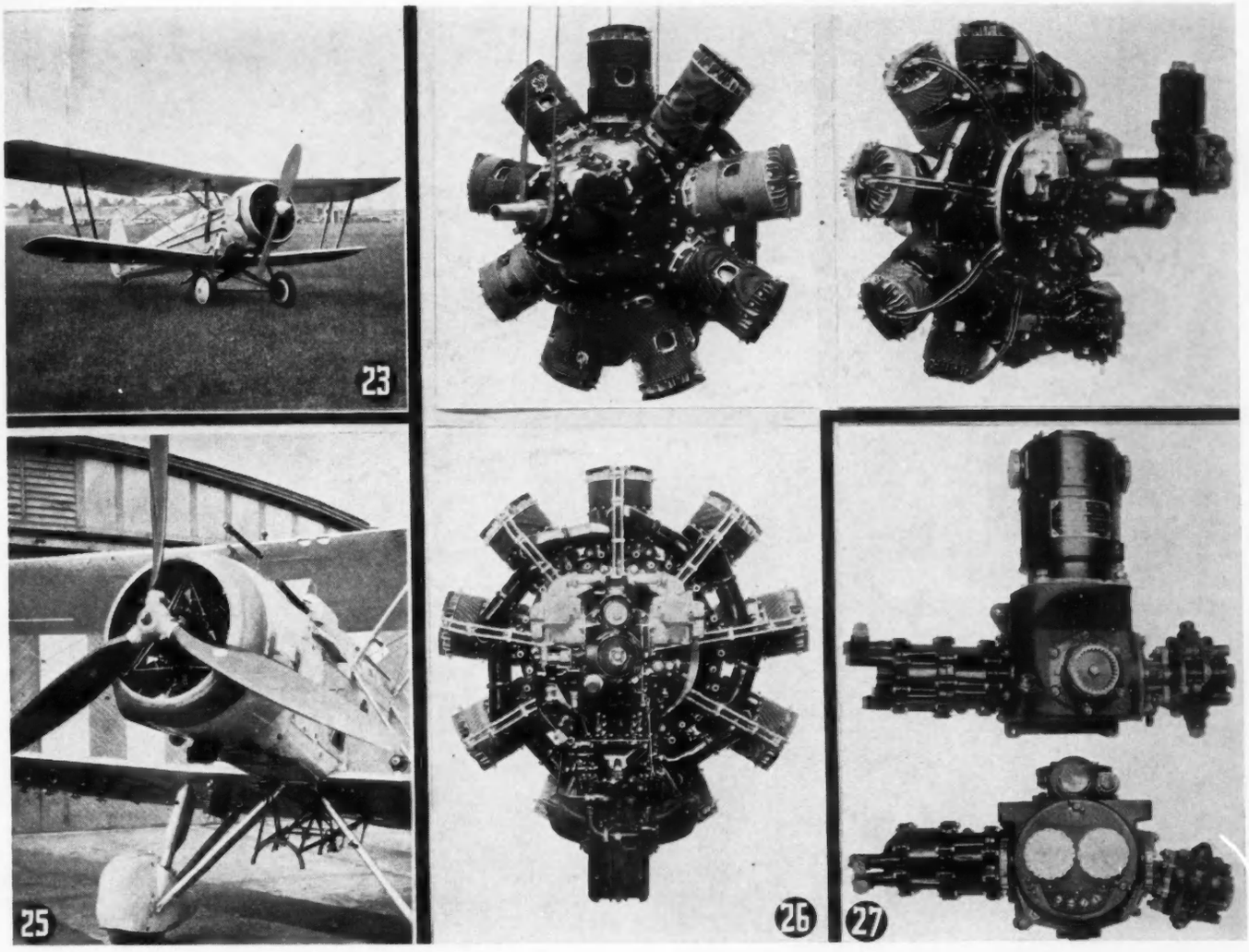


Fig. 23 – First Perseus Engine in a Single-Seater Bristol Bulldog Aircraft for Endurance-Testing

Fig. 25 – Perseus Engine Installation in Vickers Vildebeest

Fig. 26 – Three Views of Latest Production Series of Perseus Engine

Fig. 27 – Standard Perseus Accessory Gear Box

whereas the experience now available with small high-speed Diesels has taught us that direct fuel injection with small quantities of fuel present no insuperable difficulty. Fuel-air mixtures so arranged should be much more determinate over all conditions of operation, while injection also allows the use of fuels which are not carburetable.

It is understood that there are considerable supplies of fuel of high octane value but too low a volatility for use in normal carburetors, which might be employed for this purpose.

The control of turbulence mentioned previously is possibly the most important factor in this probable development.

It would be interesting to know why direct fuel injection has not made greater strides in America in view of the considerable start given to this project by the Army some years ago and, if it is in order, some authoritative statement on this subject would be welcomed.

11. Extremely flat mixture loops permitting smooth running under conditions of extreme economy.

12. Smooth running due to the good combustion-chamber shape, to the more accurate and simpler valve timing, and to the elimination of cam, tappet, and valve wear.

13. Regular cylinder shape permitting the simplest form of pressure baffling.

14. Marked decrease in number of parts in any comparable size of engine, with consequent effect on production as to times and prices.

15. Relative simplicity of all major parts, permitting accurate repetition machining when properly tooled. It is considered that the possibilities of tooling this engine for a national emergency are considerably greater than with any other type of equal efficiency which has been brought to the notice of the author.

16. Good accessibility and a clean exterior appearance.

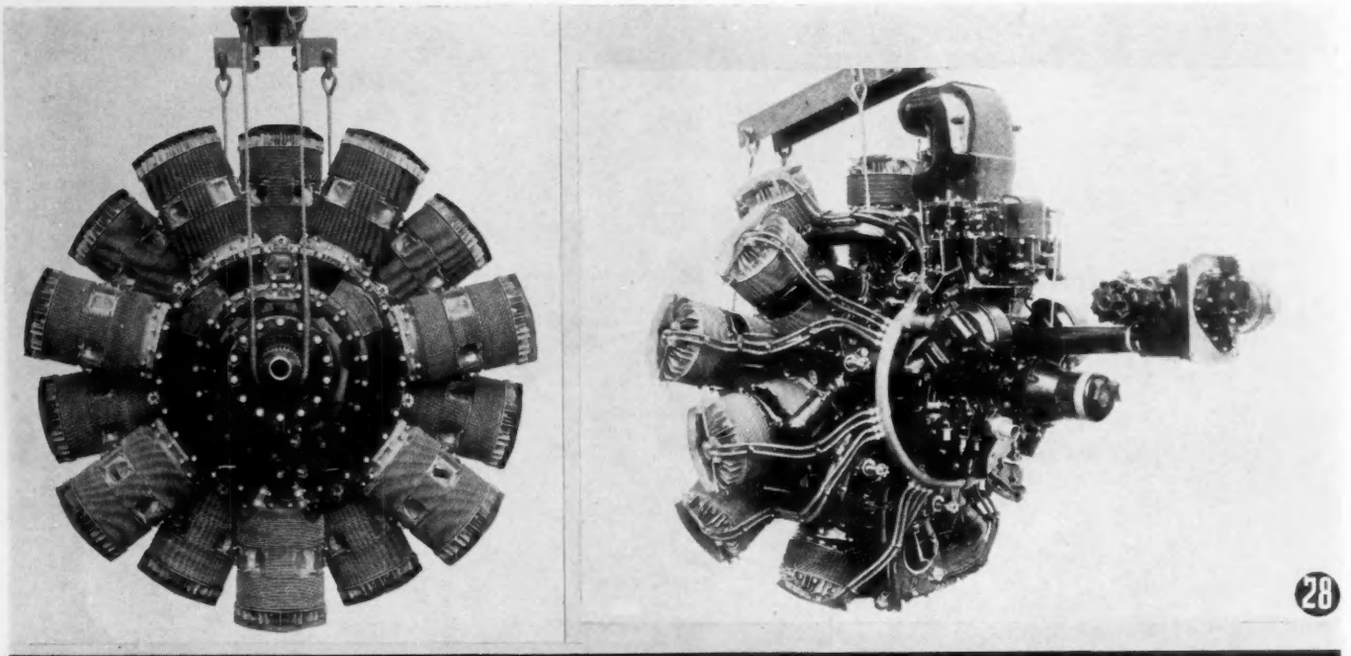
17. Probability of considerably less trouble with resultant back pressure than on poppet-valve types when using an exhaust turbo supercharger.

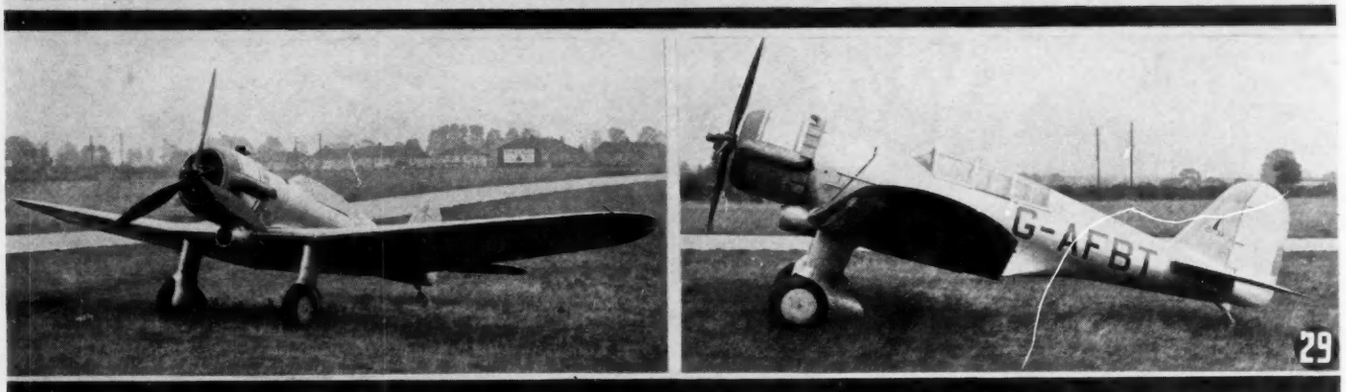
18. Greater freedom from cold corrosion. All surfaces wiped with oil, except the combustion-chamber which is of simple form, with absence of pockets, and easily doped with any anti-lead corrosive medium for storage.

19. Greater reliability due to any of the causes just mentioned, and to the elimination of parts such as valve springs.

Conclusions

Since the early successes of the Bristol Co. on the single-sleeve-valve aero engine, there has been a marked increase in interest in the type, both air-cooled and liquid-cooled form, by several prominent manufacturers. It is this mass experience that is desirable to establish further the successful devel-





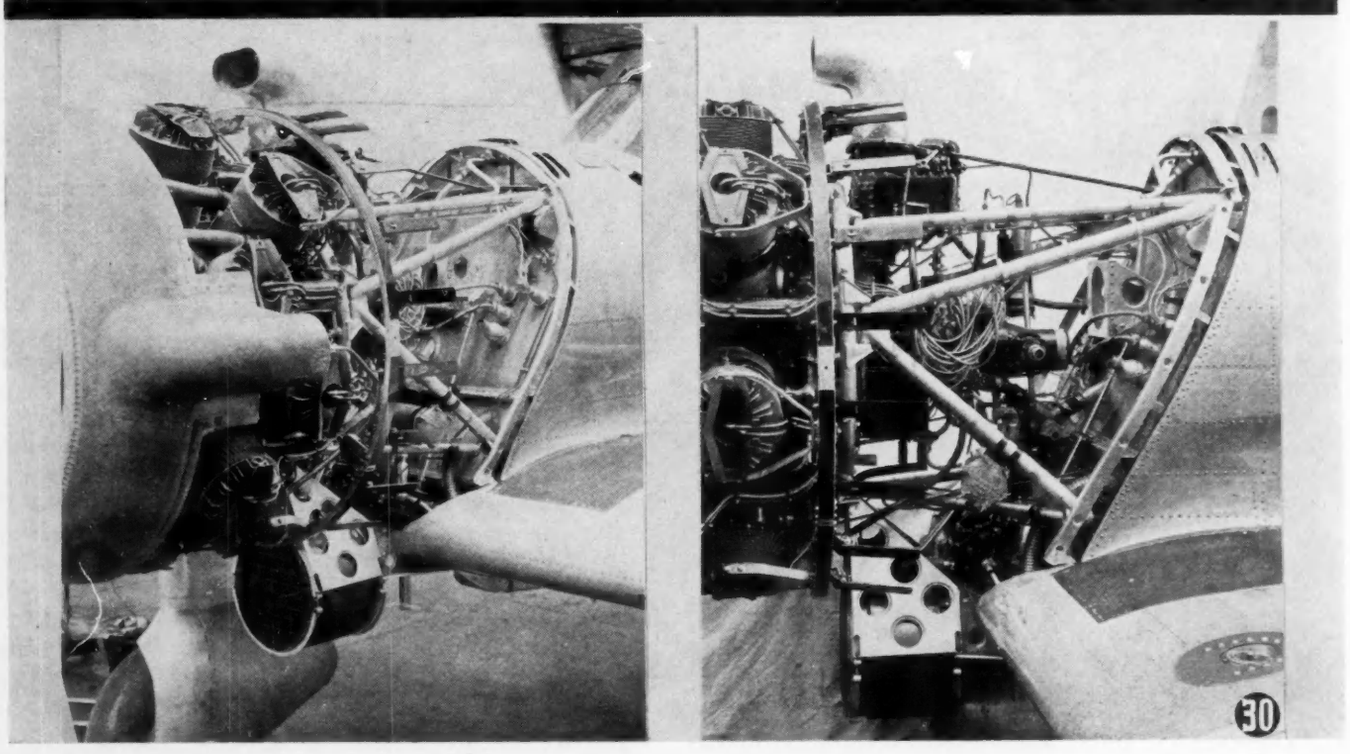


Fig. 28 – Hercules Double-Row Fourteen-Cylinder Engine

Fig. 29 – Hercules Engine in Special Northrop Aircraft for Endurance Flight Trials

Fig. 30 – Installation of Hercules Engine in Northrop Aircraft

opment of this design of valve mechanism and which, personally, I am so glad to see.

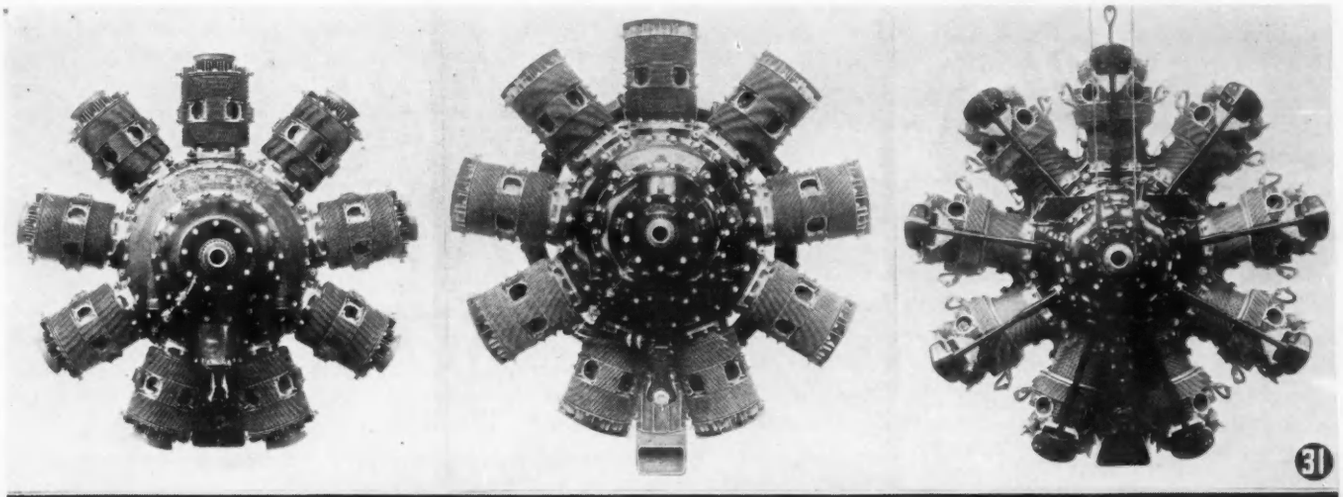
Many people have leveled what would, at first sight, appear to be a serious criticism against the sleeve-valve type, and that is, that, in the event of any internal failure, the resultant wreckage is greater than in the case of the poppet-valve design. Originally, I also subscribed to this view, and all aspects of the problem, such as the provision of "weak links" in any parts of the sleeve drive, and so on, were investigated thoroughly. However, experience with a number of engine failures which have occurred during development and overload testing has shown us that the preceding theory cannot be supported on any particular ground. On two occasions, with experimental sleeve drives, we have had failures which have been localized almost entirely to the particular cylinder in trouble, and I feel that, in the case of a piston failure, both sleeve-valve and poppet-valve engines are likely to be equally difficult once the rod becomes free. From this aspect, therefore, I must conclude that there is no hidden bogy which will prevent the completely successful adoption of this principle.

I feel that there is no reason why the type should not eventually completely supplant existing poppet-valve engines which have served us so well in the past, but which definitely have certain fundamental major problems inherent in the mechanism involved.

Although there are a few engine layouts which cannot be met conveniently owing to the inability to provide space for the existing type sleeve drive, most of the promising geometric forms can be accommodated, and the engine undoubtedly will make its appearance in multibank radial, X, and H forms.

None of the high-power running which we have completed to date has shown any indication that we are approaching any fundamental difficulties or limitations with this type of cylinder.

From both military and commercial aspects, I consider that there is every reason to suppose that it will find more favor



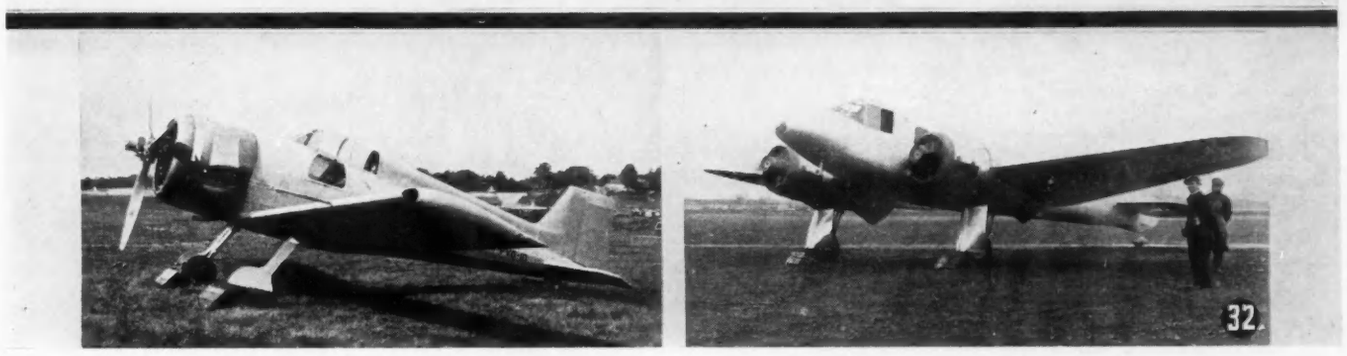


Fig. 31 – Aquila Engine (*left*) Shown beside the Perseus Engine (*center*) and the Mercury Engine

Fig. 32 – Aquila Engine Installed in the Vickers Venom Interceptor Fighter (*left*) and the Bristol Type 143 Civil Aircraft

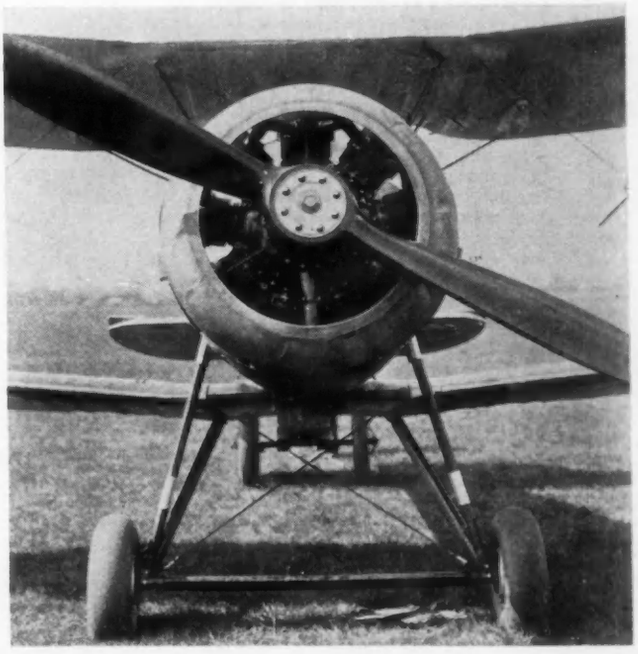


Fig. 33 – Initial Aquila Flight-Test Installation in Bristol Bullpup

than the poppet-valve type due to the relative absence of maintenance, in fact one may go further and say that there will be no maintenance other than plug maintenance, and strides already are being made toward plug-overhaul times of 100 hr. and more in service.

During our eleven years of single-sleeve development for aero engines, numerous engineers have warned me against this form of valve mechanism, pointing out the number of firms throughout the world who have tried to make a success of it and eventually have abandoned the design, and some people have gone so far as to suggest that there is a "hoodoo" upon it. I am glad to say that I have no fears on this score and, although I do not claim that it is the only solution for the valve mechanism of high-power reciprocating internal-combustion aero engines of the future, I feel convinced that it is going to be a classic one.

I must admit, however, that, looking back over the years which have elapsed during the Bristol development of the single sleeve valve, there were several times when failure to realize complete success after long periods of promising run ning gave me some qualms and called for a good deal of hard thinking, and I then began to realize why other firms had given up the conflict, and I became convinced that the successful solution of the single sleeve valve was a long and deep furrow to hoe.

I have no hesitation in saying that, had it not been for the great patience and encouragement from the Board, the loyal support of a first-rate enthusiastic design and experimental team, and the continued backing of the British Air Ministry, the present Bristol sleeve-valve development might quite well have had to be discontinued some years ago.

I feel that my conclusions are somewhat scant, but I would suggest that they may be likened to the remark that can usually be made about the condition of a sleeve-valve engine when stripped, after a hard run: "Well, what can we say about it? Very little! There's nothing wrong with it!"

This paper is intended to be chiefly historical, and is a précis of the single-sleeve development of the Bristol Co. over the last decade and, therefore, no attempt is being made to discuss derivations of the Burt-McCollum mechanism such as the open-ended sleeve, or to investigate possible solutions for the future, as this would be far too wide a field to attempt to cover in this present review.

In conclusion I wish to thank the Bristol Co. and the Air Ministry for permission to read this paper, and certain members of my staff for the assistance they have given me in preparing the data presented.

Discussion of Fedden Paper

Challenges Many Advantages Claimed for Sleeve Valve

– Arthur Nutt

Wright Aeronautical Corp.

MR. FEDDEN outlines a number of inherent advantages of the sleeve-valve design upon which I will comment.

In my opinion there should be no difference in the maintenance required on either type of engine – sleeve valve or poppet valve. With the lubricated poppet-valve gear it is the usual practice to check tappet clearances at 100-hr. periods, and no change in clearance is necessary. The checking actually is a habit rather than a necessity, although admittedly the habit remains because of an affliction long since overcome. Since the sleeve-valve engine has no tappets and the mechanism is inaccessible when assembled, no check is made on clearances.

In commenting on the comparison of the sleeve valve and poppet valve engines as outlined in paragraphs 2 and 3 of the list of advantages, I believe that Mr. Fedden has made his comparison of these engines between two engines of Bristol design and, in defense of the modern two-valve poppet-valve engine, it is hardly fair to use a comparison of a four-valve engine without enclosed valve gear with the sleeve valve as giving a fair comparison with the more modern designs. A more favorable comparison might have been obtained if first-hand information had been available to the author on a modern two-valve engine with particular reference to hot-spots, temperature gradients, and the ability to use higher compression ratio with a given fuel. Advantages under paragraph 4 also are present in the poppet-valve engine, namely, the absence of external oil leads to the working end of the cylinder, enclosure of all working parts in oil-tight compartments.

Under paragraph 7 freedom from hot-lead corrosion is claimed which indicates that the author believes the modern two-valve engine is subject to difficulty in this respect. Valves which have run for 5000 hr. appear to explode this theory.

Paragraph 10 – It seems reasonable that similar advantages can be obtained by the use of low-volatile fuel in poppet-valve engines as is claimed to be possible with the sleeve-valve engine.

Regarding Mr. Fedden's question as to why direct fuel injection has not made greater strides in this country, I believe that this condition is primarily because of the higher cost, reduced reliability, and the low state of development of a satisfactory fuel-air ratio device. The introduction of new carburetors which have eliminated carburetor icing has tended further to discourage the use of a more complicated means for supplying fuel to the engine. As I have often said, if we had started using the fuel-injection system originally and someone gave us a developed carburetor as a substitute, we would welcome it with open arms because of its simplicity, low cost, and comparative freedom from troubles. The use of low-volatility fuels may revive the development of the fuel injection system. Also, the development of larger engines with more cylinders may mean that the distribution problem will require fuel injection.

Regarding paragraph 11, can Mr. Fedden advise us whether he considers the flatter mixture-control loops peculiar to the sleeve-valve engine? Should it not become possible to obtain similar characteristics in the poppet-valve engine with controlled turbulence?

According to paragraph 12, the comparison with the sleeve-valve engine has been made with an engine which has had trouble with cam, tappet, and valve wear. Two-valve engines which run for 500 hr. with practically no attention to valve mechanism indicate that this item is not as serious as the author believes.

Paragraph 14 – Although I do not know the exact number of parts in a sleeve-valve cylinder, there are 44 parts of different design and a total of 128 parts in the modern two-valve cylinder not including the piston-connecting rod assemblies and not including the cylinder hold-down nuts. How does this number compare with the sleeve-valve cylinder?

Paragraph 15 – Indicates again a comparison of a rather complicated four-valve design with a simple sleeve-valve design. The relative simplicity of major parts is a matter of detail design and the two-valve engine is much simpler than the Bristol four-valve engine.

Paragraph 17 – It has been demonstrated that the poppet-valve engine is not affected by the application of the exhaust turbo supercharger. Messrs. Berger and Chenoweth discuss this detail in their paper on "Supercharger Installation Problems," presented at the 1938 Semi-Annual Meeting of the Society.

Regarding paragraph 18, which mentions greater freedom from cold corrosion, there might be more difficulty in adequately protecting the surfaces between cylinder and sleeve than would be the case with the two valve engine. Oil is not entirely satisfactory as a protection against cold corrosion, and it would, therefore, be necessary to cover the surfaces with some anti-corrosion material which would be difficult.

Paragraph 19 – Greater reliability can be determined only by actual service. Until sufficient time is obtained in scheduled airline service, no definite statement can be made as to the greater reliability of the sleeve-valve engine.

Although I have questioned over half of the advantages listed, there are still a considerable number of items left in support of the development outlined in Mr. Fedden's paper, and they are more than enough to justify the development.

The performance figures quoted in this paper are excellent and, when translated into full-scale figures, are enviable.

We would be interested to know how long any one sleeve-valve engine has been run in airline service without overhaul, and also the total number of hours including overhauls. Also, what cruising b.m.e.p. and crankshaft r.p.m. were used during this operation?

It is by such means that comparisons can be made with engines of current design and production.

More on Fuel Injection in Sleeve-Valve Engine

– Frank C. Mock

Bendix Products Corp.

I WOULD like to contribute a minor point to Mr. Fedden's recital, with particular reference to the mutual suitability of the sleeve-valve engine and fuel injection. As Mr. Fedden stated, this type of engine gives just the type and control of cylinder turbulence that is required with injection, and it permits locating the nozzle and spray delivery centrally in the cylinder space, with a minimum tendency toward getting liquid fuel on the cylinder (or sleeve) inner walls.

I might add that localization of the cooling effect of the fuel spray seems particularly advantageous in holding down the temperature of the one indirectly cooled combustion-chamber boundary of this type of engine, namely the piston-head. Experiments with "Esso" safety fuel sprayed against a hot surface have shown that it exerts a heat subtraction effect of about 450 B.t.u. per lb. of fuel sprayed. Perhaps this value does not seem appreciable, nevertheless, it equals about 8 per cent of the total heat loss to the cylinder walls, which explains why it has been found in practice to be quite important. For instance, directing less than one-half of the fuel spray against exhaust valves which overheated when a carburetor was used, resulted in the valves not even reaching dull red when the fuel was fed by injection. Similar but less striking effects were demonstrated when fuel was injected against the piston-head, the temperature limits being measured approximately by fusible plugs of different metals set in the piston-head.

It, therefore, seems possible that either higher compression ratios or larger cylinder displacements, or both, may be achieved by proper application of injection to the sleeve-valve engine.

The Sleeve Valve in In-Line Air-Cooled Engines

– A. T. Gregory

Ranger Engineering Corp.

THE sleeve valve appears to offer some attractive advantages for the in-line air-cooled engine. For one thing, it would permit the elimination of the overhead camshaft and housing, thus greatly facilitating servicing. The improved valving action of the sleeve valve would make possible both higher speeds and brake mean effective pressures, full advantage of which can be taken by the in-line engine.

The gain in serviceability of the engine by the use of the sleeve valve would be made, however, at the cost of compactness of the engine as a whole. Six-cylinder barrels with a crankcase on one end and a camshaft housing on the other make a pretty solid structure. Removing one or two cylinder banks as supporting structures may make the problem of crankcase design quite serious in an engine of extremely high output.

There seems to be no possibility of obtaining as good valving action with the poppet valve as with the sleeve valve. But the poppet valve also has not reached its ultimate in performance. Although we may see more sleeve-valve engines in the future, we probably will see also improvements in present valve gear and port design with the poppet valve.

Record of American Aviation Poppet Valve

– A. T. Colwell

Thompson Products, Inc.

DURING the past few years, great strides have been made in the design and lubrication of the valve gear, and in the design of the valve itself. It is a matter of record that exhaust valves have operated 5000 hr. and more, and show surprisingly little wear on the seat, stem or tip, and are being continued in service.[a] The ultimate life of these valves is as yet unknown. Enclosed valve gear with pressure lubrication, excellent head design, and other refinements, coupled with valve design, have changed completely the operating life of the parts.

The development of aircraft valves has been one of cooperation. The engine builder has contributed greatly with head and valve gear design. The oil companies have given proper lubricant and fuel. The Army contributed the clever method of measuring sodium content of valves electrolytically. The valve manufacturer has contributed the technique of fabrication and the research on materials. S. D. Heron has made numerous contributions, including the salt- and sodium-cooled valves, eutectic cooling, the suggestion of the hollow-head valve, experimental work on the technique of welding Stellite and other materials, and the electrolytic bath for cleaning used valves. The present state of the art indicates even better valve performance when it is demanded by the engine builder.

Regarding hot-spots in the head, so far as the modern hollow-head valve is concerned, we feel that one of them should be referred to as a "warm-spot," not a "hot-spot." Within the last month, one of the leading engine builders tried to get this temperature with an optical pyrometer, and reported that the valve was not hot enough to register a reading. The long-time performance of modern valves shows that they operate cool – formerly 1500 hr. was exceptional life.

I wish to refer to the remarks on valve gear in the paper given by R. N. DuBois and Val Cronstedt, entitled: "High Output in Aircraft Engines":[b]

"The valve gear is one of the few details of an engine where high operating speeds add to the stresses resulting from high gas pressures. It is true, therefore, that the valve operating mechanism offers an obstacle to extremely high-speed engine operation.

"There is, however, not the least question but that the valves themselves, if adapted to high temperatures, are likewise exceedingly well adapted to extremely high-speed engine operation. The hollow-head valve dictated by temperature is rigid and withstands the more violent decelerations and the additional valve-head stresses caused by deflections in the valve gear at high speeds. The valve seat and tip materials that have been evolved for operation at high temperature function admirably at high speed. The large-diameter stem necessary for adequate heat dissipation, provides an excellent 'cross-head' for guiding the valve with high linear velocities. Rocker side thrust, increased by the heavy springs required for high speed, is also better handled by the large stem. By using fully lubricated valve gear, the designer may reduce his chances of failure, and may further take advantage of unsymmetrical cam profiles, a device which has not seen wide use so far."

Regarding cold corrosion, this item should be no worse in the poppet-valve than in the sleeve-valve motor for, in American motors of high output, all parts also are supplied with oil and are enclosed. We believe that it is an advantage to have the valve mechanism easily available for inspection.

We do not infer that there is never any trouble with poppet valves, and Mr. Fedden probably does not intend to convey this impression of the sleeve valve. We would like to see the parts compared after 5000 hr. of similar operation.

Establishes Origin of Sodium-Cooled Valve

– S. D. Heron

Ethyl Gasoline Corp.

IN the preprint of the paper it is stated: "Further, the technique of sodium-cooled valves, originated at the Royal Aircraft Establishment, Farnborough, and subsequently developed in later years in the U. S. A., was still in its embryo state, and the excellent forging technique, which was also brought to fruition in America, on hollow stems and heads, was not then visualized as a practical production proposition in England."

This statement is definitely incorrect with respect to the work of the U. S. Army Air Corps at McCook Field which produced the salt- and sodium-cooled valve as it is known in this country. I was associated with the work on internal valve cooling at both Farnborough and McCook Field, and am in a position to state definitely that the work with non-volatile cooling liquids, such as alkali nitrates or alkali metals, at McCook Field was in no way the result of knowledge of any similar work at Farnborough.

To the best of my knowledge the use of an exhaust valve with internal cooling and with the coolant sealed in the valve *did* originate at Farnborough, at least as regards aircraft engines. This work, I believe, was carried out in 1914. (The use of internal valve cooling with an external supply of cooling water is, of course, nearly as old as the internal-combustion engine.)

This work of the R.A.E. contributed to the evolution in the United States of the salt- or sodium-cooled valve. The contribution, however, was largely that of showing that internal cooling could be very effective in large exhaust valves used in big-bore air-cooled cylinders. When the investigation was started in 1921-1922 at McCook Field it was known that mercury, water, and aniline had been used at Farnborough. Tests had been carried out with mercury-cooled valves made in accordance with the Kettering patent (U. S. Patent 1,663,709) in

[a] See S.A.E. TRANSACTIONS, this issue, pp. 366-370: "Wear Reduction of Valves and Valve Gear," by A. T. Colwell.

[b] See S.A.E. TRANSACTIONS, June 1937, pp. 230-231: "High Output in Aircraft Engines," by R. N. DuBois and Val Cronstedt.

which an amalgam is used to cause the mercury to wet the valve interior, and Kettering had supplied data showing that cooling efficiency was controlled largely by the wetting property of the liquid filling.

Known difficulties in completely sealing the valves against leakage of either mercury or water led to the idea that a non-volatile heat-transfer agent should be used if possible. The filling materials selected for test all had high boiling points and all were known to wet steel. It was thought that heat transfer from the head to the stem could be secured by the motion of the valve throwing the coolant from end to end of the cavity containing it. Cooling with mercury and water-filled valves always had been thought to occur as a result of boiling in the head of the valve and condensing in the stem portion. This conception of the mechanism of heat transfer with a volatile liquid was probably in error since mercury-cooled valves gave satisfactory operation in the bottom cylinders of radial engines.

The first test at McCook Field of valves partially filled with a non-volatile liquid showed that exhaust-valve temperatures could be reduced by about 500 deg. fahr. by this means. This test was followed by investigation of cooling liquids and of methods of construction. An entirely satisfactory construction was not obtained until the late Robert Jardine of the then Rich Tool Co. evolved the method of swaging the tip end of the stem.

The Air Corps was thus responsible for more than merely evolving means of using salt or sodium cooling. A. H. Gibson and H. Wright Baker, in describing the work on internal valve cooling at the R.A.E. and elsewhere[a] mentioned the then recent use of valve cooling with sodium nitrate-potassium nitrate at McCook Field and apparently regarded it as novel. They made no mention of any use of non-volatile liquids at the R.A.E. or elsewhere in England up to the time of their paper.

Salt- or sodium-cooled valves may have been developed at the R.A.E. without knowledge of the prior or parallel developments in this country. No claims for such development have been made until recently, however, despite the issue in 1928 of a British patent on the work of the Air Corps. Furthermore, the work of the Air Corps appeared in the S.A.E. JOURNAL in 1924[b] and was reprinted in England in that year.

The foregoing indicates that the Air Corps is to be credited with evolving and pioneering an original development and not merely with sponsoring an investigation of manufacturing methods.

The final development of the sodium-cooled valve into the production stage is in no small measure due to the Wright Aeronautical Corp. It endured more than the usual crop of troubles which are inevitable when an experimental article finally goes into production.

Asks How Oil-Control Problem Was Solved

— V. C. Young

Wilcox-Rich Division,
Eaton Manufacturing Co.

WE do not feel that the poppet valve and its mechanism has, by any means, reached their limits, and we believe that, by very careful attention to detail, higher outputs and satisfactory service can be obtained. Great improvements have been made in valve life by concentrated effort on the seating problem, both in connection with better cooling and resistance to distortion.

Ford L. Prescott in his paper on "High-Output Poppet Valve Cylinders" presented to the International Automotive Engineering Congress of the Society of Automotive Engineers in 1933, pointed out that "valves do not appear to be the limiting factor in pushing output to higher values."

Another factor, the possibilities of which have not been explored thoroughly in connection with valve mechanism, is the adaptation of the automatic-type lash adjuster, which was mentioned in Mr. Fedden's paper covering the "Possible Future Developments of Air-Cooled Aero Engines."

It always has been my belief that the inherent characteristic that held back sleeve-valve development has been the matter of oil control. When stated quickly, it sounds like a relatively simple problem and one that should not be insurmountable. When we think of all the development work that has been done with piston-rings to control the flow of oil by reciprocating parts and think that it is only within recent years that we have really accomplished a reasonable control, it brings up the question of how it can be kept within bounds of plain reciprocating surfaces such as are utilized in the sleeve-valve type of engine. Granted that the relative speed is lower, nevertheless the problem is similar and I would be very interested in hearing just how this problem in the development work was overcome.

[a] See Proceedings of the Institution of Mechanical Engineers, Vol. II, 1923, p. 1091: "Exhaust Valve and Cylinder-Head Temperatures in High-Speed Petrol Engines," by A. H. Gibson and H. Wright Baker.

[b] See S.A.E. TRANSACTIONS, Vol. 19, 1924, Part II, pp. 118-145: "Exhaust Valves and Guides for Aircraft Engines," by S. D. Heron.

Valve Opening Areas Compared

— Carlton Kemper

Senior Mechanical Engineer,
National Advisory Committee for Aeronautics

A STUDY of Table *A* shows that, as regards weight per horsepower, the design of present radial poppet-valve engines does not suffer by comparison even though the power is obtained at a considerably lower piston speed. For future engines, however, the use of the sleeve-valve mechanism indicates two outstanding advantages: first, increased cylinder charging efficiency at higher rotative speeds and, second, improvement in combustion resulting from complete scavenging of the combustion-chamber by overlapping the timing of the inlet and exhaust sleeve valves.

The improvement in valve opening area to be obtained by using 2 valves, 4 valves, and sleeve valves is illustrated in Fig. *A*. This analysis has been based on a cylinder having a displacement of 202 cu. in. The coefficients of discharge for poppet valves have been established, but there are no data available giving the coefficients of flow for sleeve ports. Fig. *B* illustrates the form of the flow passage when using a sleeve valve. For this analysis it was assumed that the flow passage approximated a sharp-edge orifice and the coefficients given by Marks[a] were used. These valves would be expected to be less than those of the sleeve valve so that the estimated values are conservative.

Table *A* (Kemper Discussion) — Comparison of Poppet-Valve and Sleeve-Valve Engines

ENGINE	BORE IN.	STROKE IN.	DISPL. CU. IN.	R.P.M.	B.H.P.	FUEL OCTANE NO.	ALTITUDE, FT.	PISTON SPEED, FT. / MIN.	HP / CU. IN.	WT / POWER
BRISTOL SLEEVE										
PERSEUS	5.75	6.50	1520	2750	870	100	15,000	2980	0.572	1.17
HERCULES	5.75	6.50	2360	2750	1450	100	15,000	2980	.615	1.12
AQUILA	5.00	5.375	950	3325	650	100	15,250	2980	.684	1.18
TAURUS	5.00	5.625	1565	3225	1050	100	13,500	3020	.671	1.19
BRISTOL POPPET PEGASUS	5.75	7.50	1750	2600	900	---	17,750	3250	.514	1.12
WRIGHT POPPET CYCLONE	6.125	6.875	1823	---	950	100	5,200	---	.521	1.34
	6.125	6.875	1823	2400	1220	100	SEA LEVEL	2750	.670	1.05

The summary table on Fig. *A* shows the large improvements in exhaust and intake areas and time areas that can be obtained by the use of the sleeve-valve mechanism. The arrangement of ports may be somewhat different from that found to be best by the Bristol Co., but the order of the gains to be expected should be correct. If Mr. Fedden's company has determined the coefficients of discharge for the sleeve

[a] See A.S.M.E. Transactions, 1936, AER 58-7, p. 596: "Square-Edged Inlet and Discharge Orifices for Measuring Air Volumes in the Testing of Fans and Blowers," by Lionel S. Marks.

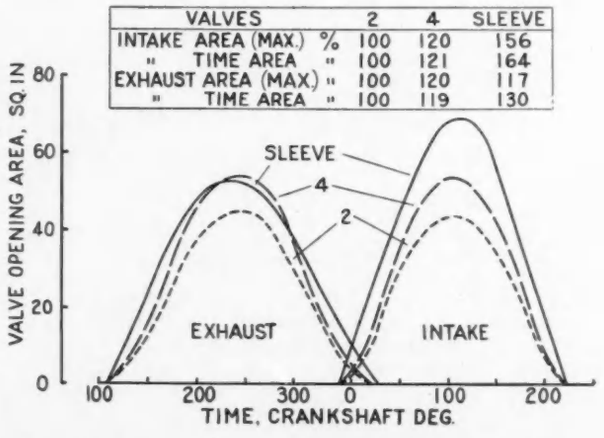


Fig. *A* (Kemper Discussion) — Sleeve-Valve and Poppet-Valve Opening Areas — Cylinder Displacement, 202 Cu. In.

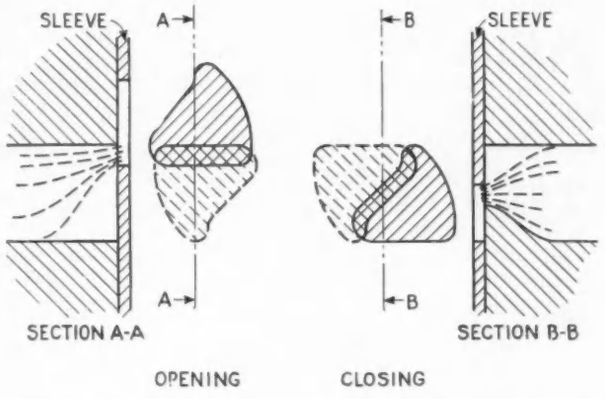


Fig. *B* (Kemper Discussion) – Form of the Flow Passage when Using a Sleeve Valve

valves, any information that he could give at this time regarding the values obtained would be appreciated.

With these large flow areas available and an almost infinite arrangement of exhaust- and inlet-valve timings, complete scavenging of the cylinder can be obtained with relatively low manifold pressures. The results obtained at this laboratory with air-cooled poppet-valve cylinders having the clearance volume scavenged by using valve overlap and low manifold pressure show a large increase in mean effective pressure and improvement in cylinder and combustion-chamber cooling to be obtained by this method. Best results will be obtained by injecting the fuel into the cylinder instead of using a carburetor. Perhaps someone will give Mr. Fedden an authoritative statement as to why fuel injection has made slow progress when the advantages resulting from its use have been demonstrated amply.

If fuel injection is used, then the volatility of the fuel is no longer of extreme importance and the way is opened to the use of fuels of low volatility (safety fuel) and Diesel oil. Although the limiting speed of the poppet-valve mechanism is considerably above present engine speeds, the results of the research of this laboratory on the uniflow type of two-stroke-cycle engine having piston-controlled inlet ports and cam-operated exhaust valves have indicated that the rotative speed of this type engine is limited by the loading of the valve gear to approximately 2400 r.p.m. This limitation in speed would be removed by the use of the sleeve valve.

Has the author's company conducted research work on the two-stroke or four-stroke-cycle sleeve-valve compression-ignition engine for aircraft? If so, are there any results of the work that can be given at this time, particularly as regards power output and fuel consumptions?

A very much-debated question has been the relative values of the friction mean effective pressure for the poppet-valve and sleeve-valve engines. Would the author supply figures for single-cylinder and multi-cylinder sleeve-valve and poppet-valve engines? These data also would permit the performance given in the paper to be placed on an indicated basis for comparison with that from other engines.

Points Out Additional Advantages

–N. N. Tilley

Continental Motors Corp.

IT will be noted that, for the same overall diameters as the poppet-valve engines, the sleeve-valve engine has longer connecting-rods, less piston side pressure, less rod angularity, and lessened differences in piston travel for the various cylinders which aids in balancing.

It will be noted also that the straightness of the bore or uniformity of the bore diameter is much better for the single-sleeve-valve cylinder than for the poppet-valve cylinder under operating conditions. There is less relative travel between the sleeve and piston on the power stroke so that the relative piston velocity is decreased when the side loads are maximum.

Survey of Special Running-Gear Lubrication

ALTHOUGH it is quite beyond the scope of this report to rationalize the problem presented or to suggest a practical solution, there are a number of basic facts that deserve specific mention and emphasis.

Most significant fact is that the manufacturers of chassis units intended for wide distribution aim at mechanical design which permits the use of high-grade lubricants generally available through the principal marketers. The incidence of passenger-car overdrives and automatic transmission which, in the opinion of well-informed people, will continue to expand in scope and application, has resulted and may continue to result in the requirement of specification lubricants possessing more closely controlled properties.

Despite this probable trend, it is encouraging to find that automotive engineers are most anxious to cooperate with the petroleum industry in the adoption of lubricants which are well within the commercial limits of widely distributed lubricants. In fact, there is an unquestioned willingness to effect compromises that will favor the most suitable commercial products rather than special or "ideal" materials.

Judging from off-the-record information gathered in this study, there is little doubt that the availability of the many widely distributed hypoid-gear lubricants, conceded to be so satisfactory for passenger-car hypoid rear axles, has presented a fresh and vexing problem to many automotive engineers. Reports from some sources indicate that, in many cases, hypoid-gear lubricant has been and is being used to lubricate conventional rear axles and all types of transmissions. Largely due to this tendency, you will note that some of the principal manufacturers have been forced to issue special bulletins warning against the use of the "active" or "powerful" E.P. materials.

One probable and perhaps undesired effect may be an unwarranted discrimination against the class of lubricants now termed "mild" E.P., which are suitable for so many applications.

This point brings us to a consideration of the types of chassis lubricants marketed at the present time and urges some clarification of these materials in the interest of everyone concerned. The three principal lubricants are:

1. Mineral gear oil.
2. Various types of so-called "mild" E.P. materials possessing extra film strength.
3. Powerful or active E.P. or hypoid lubricants.

So far as engineers are concerned, the mineral gear oils and "mild" E.P. lubricants are the *only* types that may be used in transmissions. Both the "mild" and "active" E.P. lubricants are used in axles, with exceptions noted previously in this study.

Definitely, there appears to be a need for a new and more specific nomenclature which will define the E.P. materials now being marketed and thus avoid the hazard of applying the "active" materials where they are not suitable. Perhaps it may be practicable to call the active or powerful E.P. lubricants – "hypoid-axle" lubricants, for example.

Excerpts from the paper: "Lubrication of Special Running-Gear Mechanism," by Joseph Geschelin, "Automotive Industries," presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 17, 1938.

Wear Reduction of Valves and Valve Gear

By A. T. Colwell

Vice-President, Engineering, Thompson Products, Inc.

THE factors affecting the rate of wear of valves and valve gear are summarized. Reasons for recent marked improvements in the life of aircraft valves are discussed. Surface finish developments are described that are designed to decrease the initial rate of wear. Ten means of getting close initial clearance are outlined. Progress of work in three laboratories on surface treatments is reported. The relative amount of wear with various material combinations is compared.

THIS symposium on the wear of engine parts* is to discuss the effects and the correction of wear, rather than the causes of wear. The causes are discussed quite widely in the literature, with considerable controversy, but we hope in this paper to indicate how the wear of moving parts may be diminished. Many factors affect the rate of wear of these parts, summarized briefly as follows:

Combination of materials
Rubbing speed
Pressure
Temperature
Oxidation due to heat
Surface finish of the parts
Alignment
Degree of oil film present; its ability to withstand temperature and pressure
Oil cleanliness
Corrosion due to acid attack (see Fig. 1)
Metallurgical structure of the parts
Operating clearance

Webster defines wear as follows:

To impair, waste, or diminish by continual attrition, scraping, percussion or the like; to consume or waste gradually; hence, to exhaust or lessen the strength of; fatigue; weary; use up.

In some examples of engine wear it would seem that one or two only of the above demons had participated; in other cases, apparently all of them, plus some of their relations that we do not know about, had held a seance with a chosen part, producing mystifying and annoying results. Not so long ago the metallurgist was ahead of the industry with materials perhaps better than needed. Today, the industry is waiting for the metallurgists in many lines and, in the meantime, it is necessary to do the best job possible with the materials available.

An accumulation of data from experience is likely to be more reliable than laboratory tests. There is no universal method for determining wear, nor an ideal test. Laboratory tests, at best, may indicate the trend that field tests should take. It is surprising to find how much test work has been done which gives no specific data, but indicates only in a general way that certain results may be obtainable, and many of these data are contradictory.

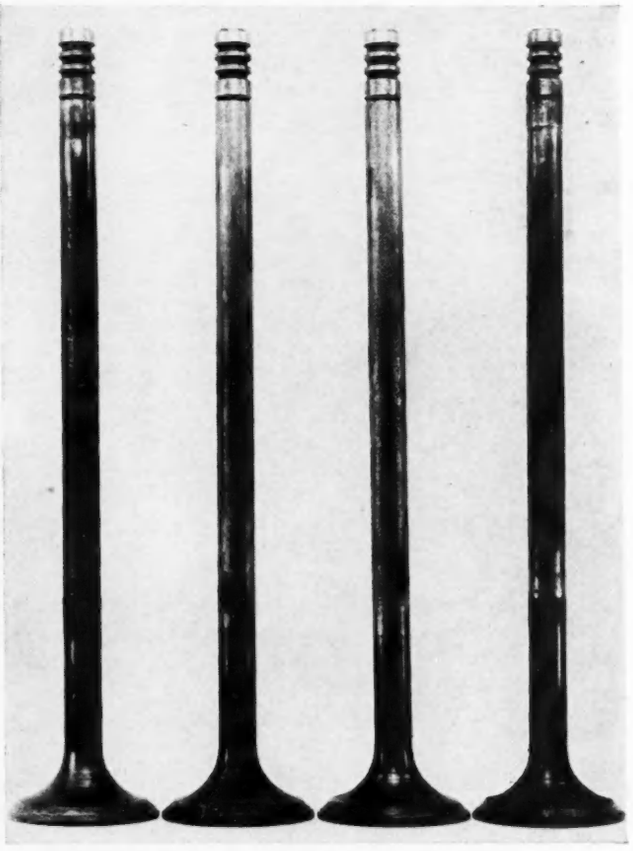


Fig. 1 – Example of Stem Corrosion

This condition is particularly true of valves and valve gear in the automotive industry, but quite the reverse is true in the aircraft industry. By well-engineered, but expensive, methods the life of aircraft valves has been extended beyond any possible anticipation of a few years back. It was not so long ago that valves finally passed a 300-hr. test to the gratification of all parties concerned. Today, aircraft valves are operating 5000 hr. and beyond in excellent condition, so that we do not know at this time their ultimate life. And these exhaust valves handle from 60 to 120 hp. each.

How was this improvement done? It was done by a combination of materials, clearances, lubrication, alignment, and surface finish. The exhaust steel used is TPA, a chrome-nickel-tungsten valve steel which is austenitic, working in

* [This paper was presented at the Symposium on Engine Wear at the National Tractor Meeting of the Society, Milwaukee, Wis., April 15, 1938.]

bronze guides. Forced and metered lubrication is fed through the rocker arms to the rocker boxes; in one case the valve stem is nitrided; exhaust guides do not project above the boss; a mirror finish is placed on the valve, and the stems are honed; seats are Stellite; seat inserts are TPA steel; a welded tip prevents wear; either a roller or ball is used on the rocker arm; the rocker-arm travel is proportioned carefully above and below center, and a roller or ball is used; alignment and selective fits give a perfect installation; the cylinder-heads are well engineered to keep the valves seated – in fact, extreme care is given to every detail, with the following results:

A hollow-head, nitrided-stem, sodium-cooled valve design has a stem diameter of 0.682 in.; it is installed with a minimum clearance of 0.0035 in.; actual operation of 3000 to 4000 hr. shows stem wear in many cases of only 0.001 in., the maximum wear seldom being over 0.004 in.; the guide life guarantee is 1000 hr., or about 180,000 miles, and the guides usually last 1200 to 1400 hr. See Fig. 2.

Another hollow-head, sodium-cooled, TPA valve design has a stem diameter of 0.619 in.; with the former rocker-arm grease lubrication, the valve was installed with a minimum clearance of 0.0075 in.; the guides were replaced at 400 to 800 hr.; the stems wore about 0.001 in. every 400 hr. With pressure lubrication the minimum clearance is now 0.0035 in.; the guides last at least 1500 hr.; maximum stem wear is 0.001 in. per 1000 hr. Incidentally, guide life is determined by either end bell-mouthing in excess of 0.005 in. See Fig. 3.

The preceding results are obtainable by better and continuous lubrication, close fits cutting down wear. These data are cited as an indication of what can be done with a wear problem where it is necessary to obtain dependable operation and freedom from frequent overhaul.

In the tractor field valve and guide wear occur, often to an alarming degree. This wear probably causes many troubles for which it is not blamed, as we seldom get definite complaints that valves wear out too fast. We know that excessive clearance develops too quickly and operation is continued too long with this condition. The complaints are directed usually to seating or breakage. We endeavor to give herewith, however, some information on improving wear in the valve train.

Surface Finish

The initial rate of wear is linked definitely with surface finish and clearance. Even a good valve-stem grind has fuzz remaining and, when such a valve stem is mated with a roughly reamed guide, the initial wear is rapid. After the surfaces wear smooth, the rate of wear is decreased due to surface condition; but it is increased due to the slippier condition as later data will show.

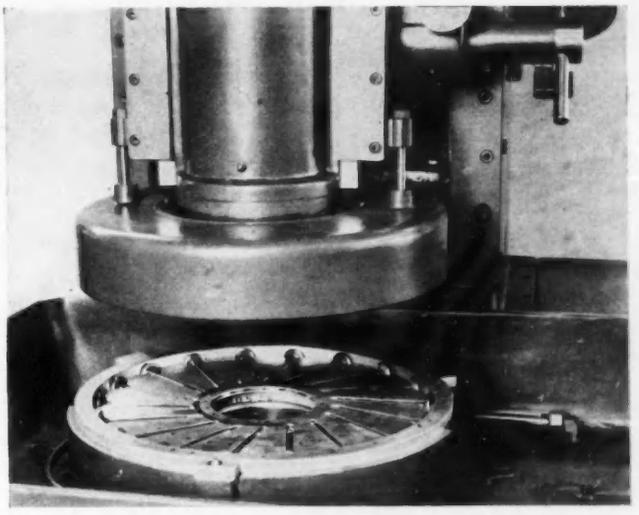


Fig. 4 – Valve-Stem Honing Machine

A new development is taking place in this line of work. The Chrysler Corp., in conjunction with the Foster Machine Co., has designed and is using new type honing machines on valve stems, tappet faces, and pistons. The valve stem is honed by placing a number of valves horizontally on a wheel as shown in Fig. 4, rotating the valves and the lower wheel, and giving an upper wheel both a circular and an oscillating motion along the valve stem. The finish produced is excellent, and the same is true on tappet faces and pistons. The finish on the tappet face is an immediate inspection of the grinding operation because inaccurate grinding is immediately visible. To our knowledge this is the first automotive application of honing such parts, although we have done it in a different way on aircraft valves for some time. The process is too new to have an accumulation of automotive field results and, without question, this type of finish will reduce initial wear. Honing machine data are shown in Table 1.

Clearance Data

Valve-stem clearance in the guide is important. The following data are enlightening:

Harry Smith, of the Buick Motor Co., ran a number of field tests to check the wear caused by various valve-stem-guide clearances. These tests were for a duration of 10,000 miles. The results showed that, for each 0.001 in. increase in initial

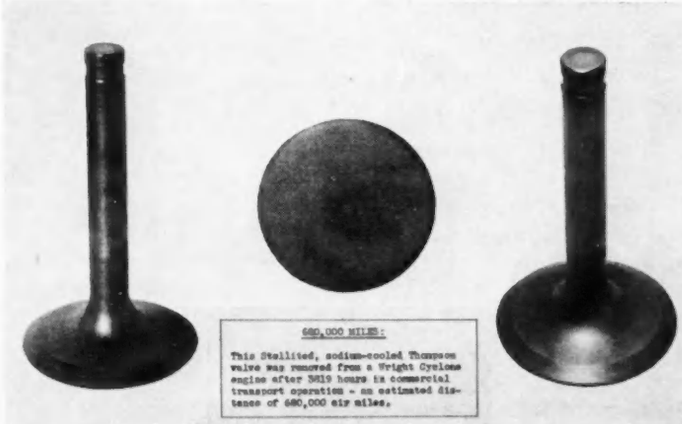


Fig. 2 – Stellited, Sodium-Cooled Valve after 3819 Hr., or 680,000 Air Miles, in Wright Cyclone Engine



Fig. 3 – Stellited, Sodium-Cooled Exhaust Valve after 4080 Hr., or 714,000 Air Miles, in Pratt and Whitney Twin-Row Engine

clearance on the valve stem, the stem wear increased approximately 0.002 in.

To our knowledge these are the first definite automotive data from a test of this type, and indicate the desirability of low initial clearance from the wear standpoint. (See Fig. 5.) It is also desirable in promoting heat transfer, preventing block pounding, valve breakage, and guide wear, and in giving perfect valve seating. It also prevents oil- and air-pumping in the intake guide. Tests run with abnormal clearance give very bad results.

Means of getting close initial clearance:

1. By good surface finish on the valve stem – smooth-grind or honing.

2. Good guide finish: the broached and sized guide is best, providing the hole is not bowed in this operation. Reaming is not a good finish. Going to extremes, a ground or honed guide would be best.

3. Perfect alignment of the guide and block seat.

4. Good bearing of the guide in the block with proper cooling. Adequately cooling the guide allows cooler stem operation and freedom from scuffing and seizure. This point on design is important. The total guide length is not the controlling factor, but only the guide length which bears in the block. Sweeping passages with bosses high in the passage are best.

5. Copper-plating the valve stem 0.0002 in. The copper acts as a lubricant during initial running, fills the depressions, reduces initial wear and clearance. In one instance, clearance was reduced from 0.0045 to 0.0025 in., with improved wear. The copper is gone from the stem in a comparatively short time, but aids initial running. Good guide cooling allows the same close fit.

6. Chrome-plating the stem definitely reduces stem wear,

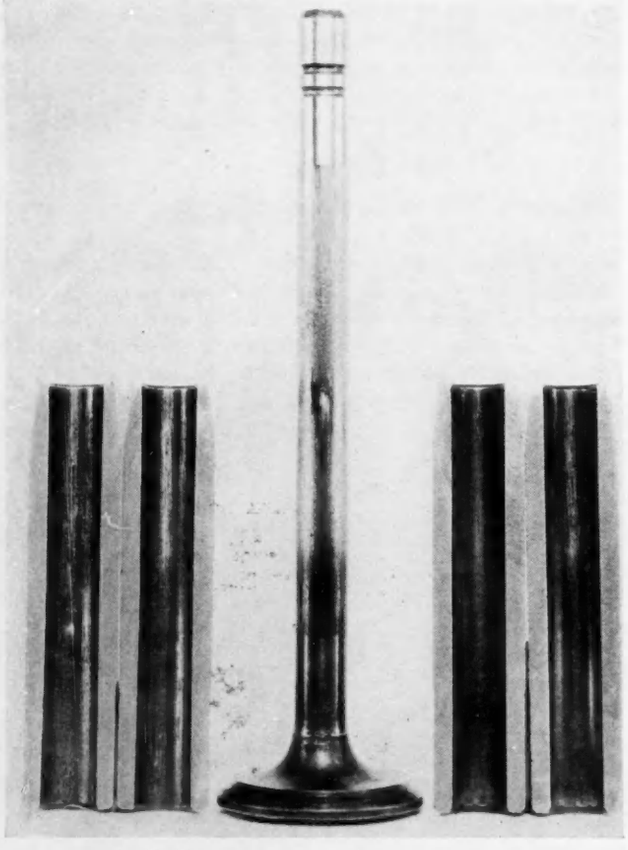


Fig. 5 – Example of Stem Scuffing in Rough-Finish Guide during Initial Run-In – Corrected by Good Guide Finish, Cutting Guide at Boss, and Reducing Stem Clearance

Table 1 – Honing-Machine Data

Machine	Honing Stone Used Make	Grit	Life	Machine Capacity Per Hour	Micro-Inch Reading
Piston	Norton	500	1,500	517	4 to 6
Drum	Norton	180-R 320-F	8,000 10,000	125 per spindle per man	17 to 20
Flywheel	Norton	150-R 220-F	1,700	90 per spindle	13 to 15
Tappets	Bay State	400	8,000	1100	2
Cam Main Bearing	Norton	500	750	85	2 to 4
Crankshaft	Bay State	500	200	40	5 to 7
Block Main Bearing	Bay State	150			
Valve Stem	Norton or Bay State	500	300,000	1000	2 to 4

but we have very little data on guide wear with a chrome-plated stem. We do, however, have the following, which is given us by Lieut.-Com. Rico Botta, of the Navy:

On a test with intake-valve stems plated with 0.0025 in. chromium, after 167 hr. operation at high brake mean effective pressure of 160 lb. per sq. in., the stem wear was only 0.0007 in. This value is considerably less than wear obtained on unplated stems in the same engine.

The U. S. Navy particularly has fostered chrome plating and reports excellent results. The chrome must not extend above the guide into the flame area as it checks and cracks. On our dynamometer we have operated chrome-plated valve stems with double clearance – the upper one-quarter of the stem was not plated and had 0.006 in. clearance, the lower three-quarters was plated and had 0.001 in. clearance. The results were excellent in the dynamometer test.

7. Proper lubrication – This is a flighty variable. Each engine is individual in this respect, and tests with metered lubrication are the only means of knowing the proper amount. Both too little and too much lubrication of the stems cause troubles, although of different types. Too little causes wear and hot-stem operation; too much may cause heavy stewing at the top of the guide, eventually holding the valve open.

8. Cut exhaust-valve guides off at the boss. This design helps in nearly all engines, as the iron permanently grows in the flame, may tilt the valve, and feeds heat to the stem.

9. Nitrided stems are effective but expensive. They are particularly good where there is too little lubrication.

10. We would recommend that service bulletins advise replacing valves and guides when a given amount of total clearance has developed; 2½ or 3½ times the initial clearance might be a good value. Also, if the guides have bell-mouthed at either end, they should be replaced.

Surface Treatments

This is a new and lively subject at the present time. Some time ago the Bureau of Standards reported a wear test of significance. This test showed that, when two steel surfaces coated with oxide film, were rubbed together, there was only a slight distortion around the edges of the oxide film. When the two surfaces were rubbed together, free from film, they became rough and distorted. Considerable work is being done along this line with piston-rings and other parts. The phosphate and oxide coatings have shown good results in initial tests.

Jack Gordon of Cadillac Motor Co. has run tests on a clutch face which chattered. The part was Bonderized and the chattered condition disappeared. He also treated a camshaft and got much better appearing tappet faces after a hard run. Attempts are being made to treat tappet faces.

L. T. Girdler of Standard Automotive Parts is conducting extensive tests on valve guides with surface treatment. This development is also too new to have definite data or recommendations. However, initial laboratory tests indicate less tendency to seize when dry valve material is rubbed on the coated surface under pressure, than with plated or plain combinations. Interestingly, better results were obtained with a finish showing tool marks than with a lapped finish. This result may mean that a reamed valve guide can be used, thus reducing cost.

Material Combinations

In an engine there are some peculiar material combinations which wear well. Hardness is not the only criterion – structure and corrosion resistance are important. For instance, a chilled-iron tappet face on a steel camshaft is an exceptionally good combination; yet the chilled iron is not as hard as other materials which do not wear as well, and it does not have free graphite. As camshafts are being changed to chilled-iron faces, the tappets are changed to ball-bearing faces.

Table 2 – Wear of Hard and Soft XB (Silcrome) Valve Stems in Combination with Hard or Soft Guides

Series II – Group 1
Test Specimens
XB Steel Stems – 0.3425 in. ± 0.00025 × 3 in.
Production Finish — Soft 25 Rockwell "C"
Hard 47.5 Rockwell "C"
Cast-Iron Guides – 0.3435 in. Diameter × ⅝ in. thick.
Smooth Ground Finish — Soft 90 Rockwell "B"
Hard 33.5 Rockwell "C"
Test Machine – Standard Automotive Parts Co.
Speed – 263 cycles per min.
Load on specimen 1.27 lb.
Total cycles each test 1,000,000
Test Conditions – Run without lubrication.

Specimen Number		Material Hard or Soft		Wear of Stem,	Wear of Guide,	Total Wear,
Stem	Guide	Stem	Guide	gm.	gm.	gm.
4	3	Soft	Hard	0.0165	0.0109	0.0274
6	5	Soft	Hard	0.0146	0.0121	0.0267
			Average	0.0155	0.0115	0.0271
3	4	Hard	Soft	0.0137	0.0123	0.0260
5	6	Hard	Soft	0.0137	0.0133	0.0270
			Average	0.0137	0.0128	0.0265
7	7	Hard	Hard	0.0132	0.0081	0.0213
9	9	Hard	Hard	0.0139	0.00635	0.02025
			Average	0.0135	0.00722	0.02077
8	8	Soft	Soft	0.0087	0.0205	0.0292
10	10	Soft	Soft	0.0108	0.0209	0.0317
			Average	0.00975	0.0207	0.03045

Again, hardened cobalt-chrome valves operating in hardened steel valve guides in aircraft engines around salt water gave terrific trouble. Carbuncles formed on the valve stem due to electrolysis when the engine was idle, and rapidly wore the guides. Or, over a longer time, the valves stuck in the guides so that it was necessary to pound them out with a hammer.

Also, hardened exhaust-valve stems in aircraft motors wore much faster than the soft austenitic stems. We attribute this result to the particular hardened structure and also to the fact that austenitic steel work hardens in operation.

We have run a great many valve-and-guide combinations on a wear machine. Table No. 2 shows results. In general, the tests show that a hard valve stem wears least, but it wears the guide more; and a hard guide wears least, but wears the stem more. A hard-stem-hard-guide combination seems best of all.

There is one exception to the hard-guide-soft-stem wear, and that is the nitricastiron guide. This guide is made from iron with a chrome-aluminum addition. It will harden from



Fig. 6 – Stellite Aerotype Valves after 110,000 Miles in a Greyhound Bus – Stem Wear, 0.004 In.

700 to 900 Brinell by nitriding. The total nitriding depth will be 0.008 to 0.010 in., and the high hardness the first 0.003 in. The material has shown remarkable properties as a cylinder-sleeve and valve-guide material. This fact we attribute mostly to the hardness structure, some to the corrosion resistance. The surface structure is hard iron and chrome nitride needles interspersed with graphite. The hard needles take the wear and the graphite feeds lubrication. Tests on cylinder sleeves will be given in another paper. Field tests with these guides show six to ten times the life of a gray-iron guide. The guide bore is honed, and clearance can be set close. This is an expensive guide to make, but it is a real solution where severe guide wear is a problem. It is being used in a limited way in production. This guide must not extend above the boss, that is, into the flame.

In this connection, practically all high-output aircraft engines here and abroad are using nitrided steel barrels with excellent results. Again we attribute this condition mostly to hardness, some to corrosion resistance.

Valve-stem wear varies in different automotive and tractor engines and with operating conditions. We have seen it as little as 0.002 in. in 100,000 miles, with a perfect installation (see Fig. 6, 110,000 miles), and as much as 0.008 in. in 20,000 miles. The points just enumerated under close clearance will reduce valve-stem and guide wear. Better guide materials can

Table 3 – Wear of NI.-CR.-MO. Guide – Duration of Test 20 Hr. – Load 2.74 Lb. – Silcrome Valve Stems

ANALYSIS OF SAMPLES

	Ni.-Cr.-Mo. Iron				Ni-Cr. Iron			
T.C.	3.44	Mn.	0.76	T.C.	3.44	Mn.	0.76
Si.	2.58	Ni.	1.01	Si.	2.58	Ni.	0.64
S.	0.084	Cr.	0.31	S.	0.084	Cr.	0.11
P.	0.18	Mo.	0.53	P.	0.18	Rockwell B	97
		Rockwell B	101				

WEIGHT LOST BY SPECIMENS

Ni.-Cr.-Mo. guide	0.0165 gm.
Ni.-Cr. guide	0.0406 gm.
Stems with Ni.-Cr.-Mo. guide	0.0169 gm.
Stems with Ni.-Cr. guide	0.0140 gm.
Total Wear	
Ni.-Cr.-Mo. combination	0.0334 gm.
Ni.-Cr.	0.0546 gm.

be used to reduce guide wear further, in addition to finish and clearance. The molybdenum irons with additions of chrome and nickel definitely wear better. The gray-iron guide has a tendency to bell-mouth, and also to grow permanently when exposed to the exhaust flame. Alloy irons are better in this respect. Laboratory tests indicate a wear superiority of ni.-cr.-mo., as shown by the data in Table 3. Definitely less guide wear can be engineered by the use of better guide materials.

General

Wear in other parts of the valve train can be minimized by the following expedients:

1. Valve-seat wear, pounding, and "pick-up" can be corrected entirely by the use of a good seat insert, thus increasing valve life. Hard seat inserts predominate and are highly satisfactory – they are low in cost, but must be ground; soft seat inserts are more expensive, can be reamed, are satisfactory, need a wider valve seat to reduce unit impact.

2. Valve-tip wear is eliminated by hardened tips on ferritic steel or welded tips on austenitic steel. In this connection, wear of the entire valve train is reduced by the use of an automatic valve-clearance regulator. Such a device is beneficial to the block seat, valve tip, stem and guide wear, and to timing. This device will find wide use in the future.

3. The distribution of rocker-arm travel is important in the valve-stem wear of overhead engines. A good proportion is 40 per cent travel above center and 60 per cent below, producing minimum thrust. Also, a good finish on the valve contacting surface is important; a roller or ball would be a great improvement, but these designs are expensive. Contact near the center of the valve tip is important.

4. Wear or pounding of the valve seat itself is corrected by the use of some of the newer steels, which are hardenable for operation, or the use of Stellite puddled on the seat. The latter method is general exhaust-valve aircraft practice, and for heavy-duty automotive valves.

5. Worn rocker arms which are noisy have been quieted and wear reduced by tin-plating the part.

6. Worn retainer necks on valves can be prevented by proper fit of the lock. The snug-fitting beaded lock is becoming used universally for this purpose.

7. Worn intake-valve seats indicate that the air cleaner is not functioning; also the spring may be defective, allowing the valve to bounce and spin as it reseats.

8. Turning valves usually show a better seat condition, particularly on the exhaust, but can cause accelerated valve-seat and block wear, particularly on the intake side. The valves should not spin, but it is often desirable to have them turn slowly.

9. Valve springs are important in connection with wear. If the ends are not flat, side thrust is developed, inducing rapid wear, seat pounding, and breakage. Also, the springs should be balanced. Weak springs allow the valves to flutter, causing seat wear and breakage; springs too heavy cause seat wear and stretched valves. Valve springs are much more important than the amount of attention that they sometimes receive would indicate.

10. Water-pump shafts have given considerable trouble due to wear. With a packing gland, rust and dirt imbed in the packing and wear the shaft away. This condition is eliminated entirely by the use of the packless pump or mechanical seal, to which there is a distinct automotive trend. Water does not reach the shaft with such a seal, and thus the type of wear just mentioned does not occur. The ordinary bearing wear is reduced further by the growing trend toward ball bearings, with greatly reduced bushing wear.

For pumps with packing, many things have been done to reduce wear and corrosion. Stainless material, nitrided shafts, and chrome-plated shafts are used. J. B. Fisher reports that an "Ihrigized" or siliconized shaft, developed by Dr. H. K. Ihrig, of the Globe Steel Tubes Co., wears exceptionally well, despite the low hardness. The surface has the ability to absorb oil, and is exceptionally corrosion-resisting in low-temperature work, which quality no doubt explains the fine results obtained. It is very difficult to machine. The use of oilless or sintered bushings is of advantage when it is difficult to get lubrication. They have the property of transmitting oil by capillary attraction, and are of great use in many cases.

11. Tappets – Tappet wear is not a serious problem. Lubrication is good and faces stand well. Most of the barrels are hardened and close clearance is used. A honed face or surface treatment may help some. Alignment is important, however, as often L-head tappets strike the valve off center, producing thrust.

One service manager who I know has a stock answer for a returned worn part: "Its designed life has expired." Our problem is to extend the normal expectancy of this designed life. Definite progress is being made but, at this moment, it is not likely that production will be curtailed due to infinite engine life.

Supplement

Regarding the surface treatment of cylinder liners, E. A. Halloran, of the Standard Oil Co. of Calif., submits the following information:

The metal-treating process developed by the Standard Oil Co. of Calif. for ferrous frictional surfaces comprises a chemical sulphidizing treatment of the surfaces of the metal. This process involves essentially the immersion of parts in the treating solution of moderate temperature, and the treatment produces a surface which breaks in rapidly and is more susceptible to wetting with oil than are untreated surfaces.

Scuffing and wear tests in a testing device showed an increase in film strength of 5:1, and a reduction in wear of 20:1, as a result of the treatment. The process is particularly applicable to treatment of cylinder liners and piston-rings. Tests in several hundred Diesel engines of a well-known make showed a marked decrease in break-in time, and the scuffing during break-in was reduced to at least one-tenth that obtained with untreated parts.

Application of the process to cylinder liners and piston-rings allows full engine load to be applied very early in the break-in cycle, practically eliminating scuffing during break-in, and reduces wear in subsequent operations.

Acknowledgments

The writer wishes to thank those in the industry who have contributed data for this paper:

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J. B. Fisher, Waukesha Motor Co.

Thompson Products Engineering Staff.

E. A. Halloran, Standard Oil Co. of Calif.

Quantitative Analysis of Heat Transfer in Engines

By R. N. Janeway

Head, Dynamics Research Dept., Chrysler Corp.

IN the first part of the paper, which he presented at the Detroit Section Meeting of the Society, April 11, 1938, Mr. Janeway reviews definitions and fundamental laws of conduction, convection and radiation.

The second part, which follows, is devoted to a quantitative analysis of heat transfer in engines. It takes up, in turn, the heat transfer from gases to the cylinder walls; the heat transfer from the cylinder surface to the water; and piston-head temperature. Three significant conclusions are discussed.

SEVERAL years ago we investigated the limiting efficiency of the four-cycle engine based on the actual characteristics of the working gases in contrast to the theoretical air cycle which is obviously fallacious. The temperature-energy diagrams used for the products of combustion and combustible mixture are shown, respectively in Figs. 1 and 2. These were originally given in a paper on "Interpretation of the Indicator Card" which we presented before the S.A.E. in 1929.[1] Since then the combustible-mixture diagram has been corrected to include the effect of progressive vaporization of the portion of the fuel which is liquid at the start of compression.

When it first was decided to tackle this problem, extensive calculations were made using what were thought to be representative values for exponents of compression and expansion. Efficiency and output were calculated, using the temperature-energy diagrams, over the range of compression ratio and volumetric efficiency. Comparison of the results with actual engine performance showed that, although at low speeds actual efficiency fell short of our so-called ideal, at high speeds some engines not only equalled but actually surpassed our figures for limiting efficiency.

This experience taught two lessons: first, that exponents could not be assumed but must be derived analytically from the laws of heat transfer; second, that engine speed is of paramount importance as an inherent factor in limiting efficiency.

I. Heat Transfer from Gases to Cylinder Walls

A. Analytical Method

In attacking the problem of determining rationally heat transfer during each stroke of the cycle, we were faced by a confusion of variables. Each factor known to influence heat transfer from a gas to surrounding surface is subject to continual change throughout the cycle; gas temperature and cylinder surface change from instant to instant; and the coefficient of heat transfer, sensitive as it is to turbulence, was entirely obscure both as to mean value and possible variation during the cycle.

The rate of heat transfer from a gas to a surrounding surface is, in general, proportional to the product of surface, temperature difference, and a coefficient expressed in terms of B.t.u. per unit area per degree temperature difference. It is obvious then that, for a given coefficient, the effect of known variations of surface and temperature difference can be determined by integrating their product with respect to time. This operation can be done most conveniently for the various strokes of the cycle by the graphical method of plotting the product of simultaneous values of surface and temperature difference against crank angle which is proportional to time. The cylinder-surface variation is, of course, obtained readily from the piston positions at various crank angles for any given engine. The gas-temperature variation during compression and expansion also can be computed for given initial temperature and exponent at the same piston positions. Fig. 3 shows a typical set of curves giving the variation of gas temperature with crank angle for three different exponents; Fig. 4 shows the variation in exposed surface per cylinder with crank angle, both for a 3¼ in. by 4¼ in. L-head engine of 5.4:1 compression ratio.

Since the temperature difference involves wall temperature as well as gas temperature, the next question to be decided was how to handle the wall temperature. Although the surface temperature at any point is certainly constant for all practical purposes under an equilibrium operating condition, we do know that different portions of the surface, such as combustion-chamber, piston, valves, and cylinder wall vary widely in temperatures. This condition means that, at each piston position, the mean temperature of the exposed surface is different due to the inclusion of a varying proportion of relatively cool cylinder-wall area. The lack of reliable data on these internal temperatures necessarily makes impossible an exact determination of overall mean wall temperature during the stroke. Fortunately, however, the results are not unduly sensitive to wall temperature due to the high gas temperature during most of the cycle. For instance, taking a representative mean value of temperature on expansion of 2700 deg. fahr. with a 300 deg. fahr. mean wall temperature, gives a mean temperature difference of 2400 deg. fahr. An error in assumed mean wall temperature of ±50 deg. fahr. would therefore involve a discrepancy of only ±2 per cent in the calculated heat transfer. This amount is obviously negligible.

[1] See S.A.E. TRANSACTIONS, Vol. 24, 1929, pp. 92-101, 114: "Interpretation of the Indicator Card," by R. N. Janeway.

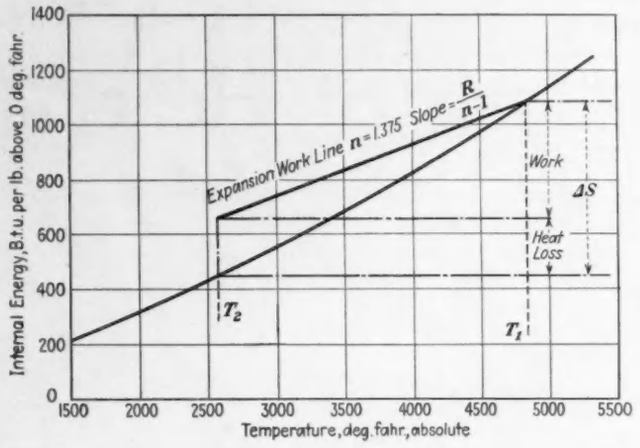


Fig. 1 – Temperature-Energy Chart for Combustion Products during Expansion

Based on estimated temperatures of various portions of the internal surface, the integrated mean temperature over 180 deg. of piston travel was found to be approximately 300 deg. fahr.

In the absence of definite data on change in internal surface temperature with speed, it was decided to use 300 deg. fahr. as the mean wall temperature at full load over the speed range. This assumption is felt to be justified, first by the insensitivity of the results to moderate changes in temperature; second, by the fact that increased speed is accompanied by increased rate of water circulation in the jackets and consequently greater rate of heat transfer to water. This factor should overcome the tendency for the temperature to build up with increasing speed, at least for the directly cooled surfaces.

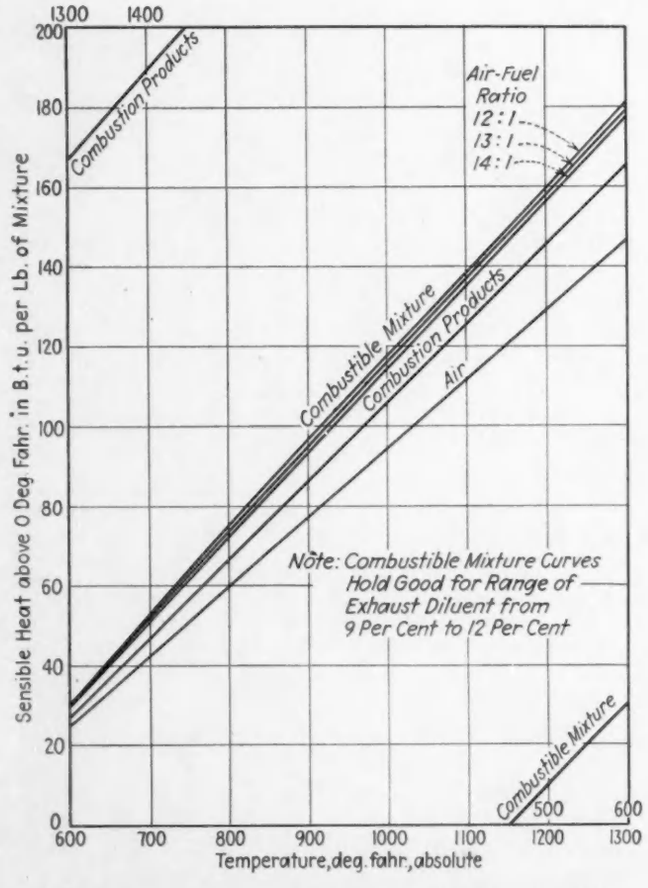


Fig. 2 – Temperature-Energy Chart for Combustible Mixture, Air, and Combustion Products

Having our surface variation and wall temperature, all required information was now available for integrating the surface times temperature difference for any given gas temperature curve as a basis for determining the total heat transfer on each stroke of the cycle.

Figs. 5 and 6 show typical integration areas for compression and expansion strokes. Referring to Fig. 5, it will be noted that, at the beginning of compression, the heat interchange is negative, that is, the gas is cooler than the surface and picks up heat instead of losing it. The net area is positive, of course, corresponding to a net loss of heat over the compression stroke.

Fig. 6 shows that the increase in surface is in almost the same proportion as the drop in temperature difference during expansion.

This condition would tend to equalize the rate of heat loss

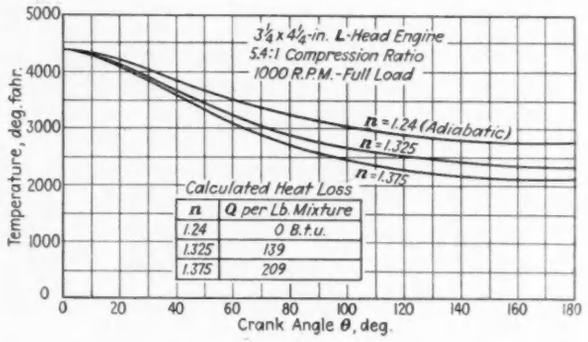


Fig. 3 – Variation in Gas Temperature with Crank Angle for the Expansion Stroke

during the stroke except for unknown variations in coefficient of heat transfer. Since the rate of expansion depends on both piston position and velocity, it is subject to considerable change in magnitude during the stroke. In order for a uniform exponent to hold from point to point during expansion, the rate of heat loss would have to change in a particular relation to the rate of expansion. Considering all the independent variables operating, it would be straining the laws of probability to expect a uniform exponent to obtain from point to point. However, this statement does not mean that the assumption of uniform exponent introduces any appreciable error in the analysis. From an efficiency point of view, we are concerned only with the overall energy balance which depends only on the initial and final temperatures. Variations in the point-to-point temperature path from that corresponding to a uniform exponent might affect theoretically the integrated heat loss area. The possible degree of error, however, is microscopic considering the high gas temperature level, and the fact that the product of surface and temperature difference varies so little over the stroke.

It will be noted in Figs. 5 and 6 that the heat loss in B.t.u. per min. represented by the particular area is expressed as a constant times the area and the coefficient, K, and is independent of engine speed. Although the absolute time equivalent of the crank-angle period varies inversely with speed, the number of cycles per minute increases directly, so that speed cancels out when the heat loss is put on the basis of unit time. This condition indicates that the total heat rejection would tend to be independent of speed if the coefficient remained constant. *The principal influence of speed on the limiting cycle efficiency thus clearly lies in its effect on the coefficient of heat transfer.*

This point brings us to a consideration of the most elusive factor in the problem, namely, the coefficient of heat transfer.

It is well recognized that the rate of heat transfer from a gas to surrounding walls is vitally dependent on the gas velocity or turbulence relative to the surface. However, in order to correlate properly the coefficients of heat transfer during the various phases of the cycle, it was necessary to dig deeper into the fundamentals involved.

The greatest advance in the subject of heat transfer dates back to 1874 and the classical paper of Prof. Osborne Reynolds on "The Extent and Action of the Heating Surface for Steam Boilers."[2] Reynolds previously had investigated and formulated the laws of resistance to fluid flow. According to his theory, resistance to turbulent flow is caused by the loss of energy of particles in eddying from layer to layer of the flowing fluid. The magnitude of the resistance depends upon the number of eddying particles per unit of time which is a function of the weight of fluid flowing through unit area per unit of time, or of velocity times fluid density. Reynolds, being a great philosopher, reasoned from this point that the rate of heat transfer between the fluid and surface must follow a similar law since, the more particles which could eddy from the body of fluid and contact with the surface, the greater would be the direct transfer of heat, independent of conduction through the fluid. This theory was substantiated experimentally both by Reynolds and others, who found that rate of heat transfer, as in the case of resistance to flow, depends upon weight of fluid flowing through unit area per unit of time. In other words the coefficient of heat transfer depends not merely upon velocity of flow but on the product of velocity and density. In addition, the coefficient is found to be substantially proportional to the specific heat of the fluid at constant pressure.

The importance of density to heat transfer in the engine is obvious when it is considered that the density is variable from

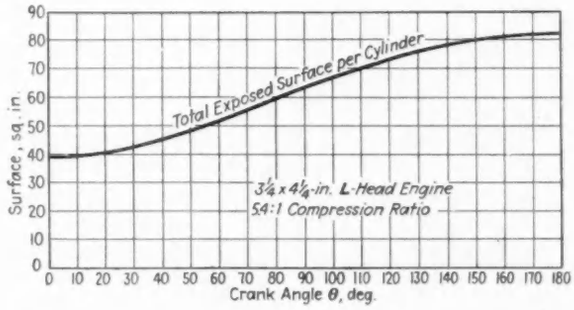


Fig. 4 – Variation in Exposed Surface with Crank Angle

point to point throughout the cycle, as well as being basically dependent on volumetric efficiency.

In analyzing the variation in density, one fact of great importance stands out, namely, that the density at a given piston position is the same both on compression and expansion. This fact eliminates density as a variable in considering the relative mean coefficients of heat transfer on compression and expansion. However, the density is obviously a maximum during combustion and several times greater than the mean density during compression and expansion, depending on the compression ratio. This point leads to an expectation of a considerably greater heat-transfer coefficient during combustion as compared with compression or expansion. Likewise, on the exhaust stroke, after the bulk of the spent charge has been released, the density is a fraction of the mean density during compression and expansion and the expected coefficient should be correspondingly lower.

[2] See Proceedings of the Literary and Philosophical Society of Manchester, Vol. 14, 1875, pp. 7-12: "The Extent and Action of the Heating Surfaces for Steam Boilers," by Osborne Reynolds.

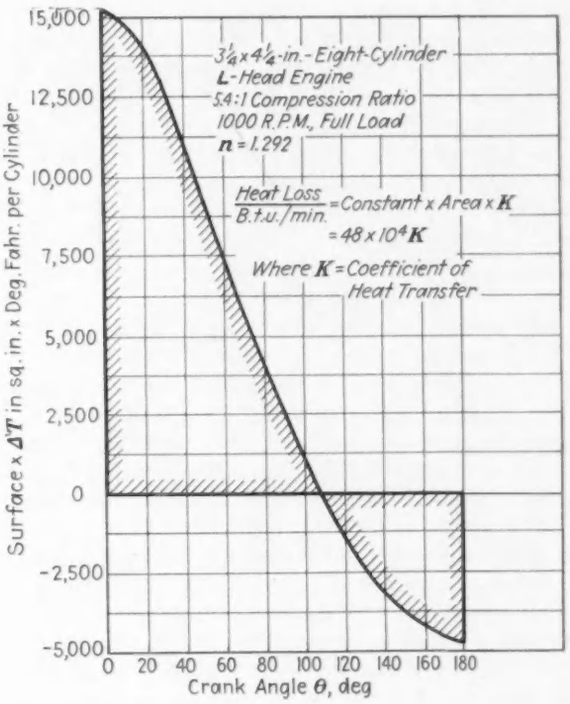


Fig. 5 – Graphical Integration of Heat Loss on Compression

Since the specific heat of the fluid varies widely with temperature as shown, for example, by the increasing slope of the temperature-energy curve of Fig. 1, the coefficient of heat trans-

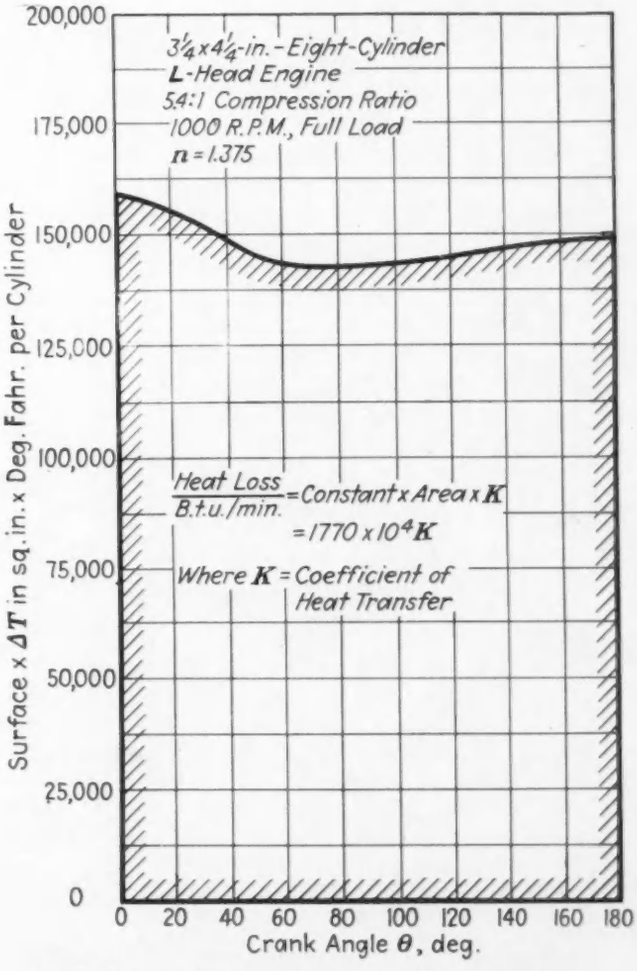


Fig 6 – Graphical Integration of Heat Loss on Expansion

fer must be influenced correspondingly throughout the cycle. This condition means that, other things being equal, the coefficient will be appreciably higher during expansion than during compression and will be increased still further during combustion.

The velocity or turbulence effect in the engine is infinitely more complex than the case of gas flowing in a pipe. Instead of unidirectional flow at constant velocity, we have first the high entrance velocity of the fresh charge through the valve, setting up an unknown degree of agitation, then the rapid piston movement in the opposite direction. Comparing the probable turbulence on compression and expansion, we know definitely only that the effect of piston velocity must be the same. During compression we have, in addition, an unknown degree of turbulence residual from the intake stroke. During expansion, although this additional turbulence is probably lacking, the heat transfer by convection is augmented by a certain amount of radiation which is a function of temperature and nature of medium, and independent of the convection effect. Although radiation is a much smaller factor than convection in heat transfer from hot turbulent gases, it probably becomes appreciable at least during the first part of expansion. Unfortunately there are no available data on which to base an estimate of the relative losses by convection and radiation in the engine.

It is felt that consideration of all factors involved, particularly the fact that density and piston velocity are the same, makes reasonable the assumption that the coefficients of heat transfer for compression and expansion are in the ratio of the mean specific heats of the gases during the respective strokes. The value of this relation is practically constant over the speed range, the mean specific heat during compression being about 80 per cent of that during expansion.

It follows that the heat losses on the two strokes must be in the ratio of the respective products of heat-transfer integration area and constant-pressure specific heat.

It has been shown that any particular exponent for compression or expansion under given initial conditions corresponds to a definite quantity of heat loss or addition as well as to a definite temperature vs. crank-angle relation which, together with the fixed surface variation of a given engine, determines the integration area. Therefore, for any compression exponent there can be only one expansion exponent which will meet the required relation between the calculated heat losses for the two strokes.

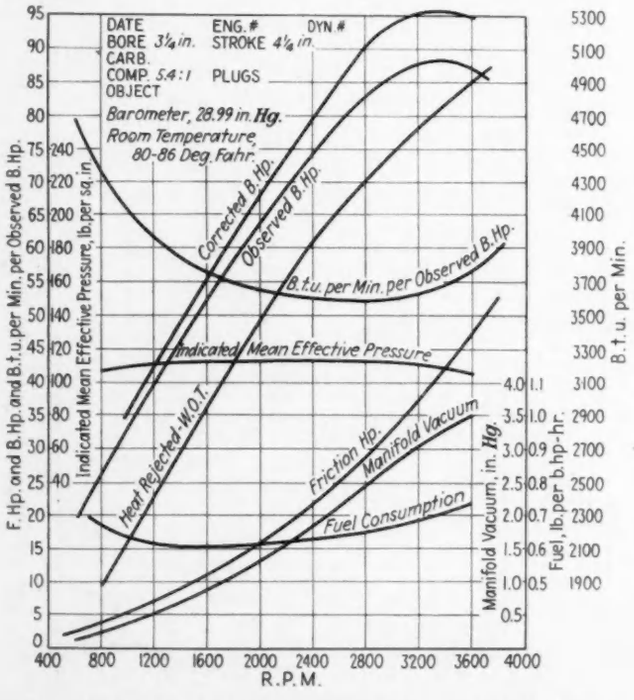


Fig. 7 –Curves of all Test Data Taken and Calculated I.M.E.P.

The application of this method to the evaluation of probable values of heat-transfer coefficients obviously requires an experimental basis. By correlating actual engine performance data as to volumetric efficiency and power output with the analytically determined variations in the heat transfer during the several phases of the cycle, the actual exponents of compression and expansion can be determined together with the corresponding heat-transfer coefficients.

For this purpose a representative engine was chosen, namely a 3¼ in. by 4¼ in. eight-cylinder L-head type of 5.4:1 compression ratio for which considerable data were available, including heat-dissipation measurements at full load over a speed range of 800 to 3600 r.p.m. The carburetor had standard fixed jets, and no attempt was made to obtain the best adjustment at each speed. It also should be noted that the runs were necessarily of fairly long duration in order to permit accurate heat-dissipation measurements under equilibrium conditions. For this reason also, the power output was somewhat lower than would be obtained on the usual power run. Fig. 7 shows curves of all test data taken and the indicated mean effective pressure calculated from the observed brake torque and friction. The variation in surface with crank angle, shown in Fig. 4, was based upon the combustion-chamber and cylinder dimensions of this engine.

With the air and fuel consumption and the indicated output known, the net work per pound of mixture is obtained readily. It has been shown that the method of relating compression and expansion exponents by means of heat-loss integration areas leads to a definite expansion exponent for any assumed compression exponent. The addition of a net-work requirement means that there will be but one pair of exponents which will satisfy the condition for balance between calculated heat losses and heat-transfer integration areas.

A further requirement in evaluating the actual cycle is the allowance for heat loss during combustion. This item is almost entirely a matter for speculation, first, because of the unknown degree of loss by radiation; second, because of the uncertainty as to the degree of turbulence which exists during combustion relative to that during compression and expansion.

It is reasonable to expect a considerably higher coefficient to obtain during combustion than during expansion or compression if only because the density is much greater. The turbulence effect must also be great to make possible the rapid combustion rate which we know exists. However, although we know the turbulence effect on combustion rate increases with speed, we have no assurance that the coefficient of heat transfer is affected similarly. In fact, there is evidence that the coefficient of heat transfer does not increase continuously with increase in turbulence but approaches a condition of saturation beyond which it tends to remain constant.

The radiation characteristic will, of course, be independent of speed and the total radiant loss will change only with temperature and time of combustion. The combustion time interval can be approximated fairly closely by considering the end point to occur at, say 20 deg. after top dead-center at all speeds, and adding to this value the observed spark advance in degrees of crank angle.

Considering all the uncertainties surrounding this subject, it was decided to lump the radiation and convection loss together by using a single coefficient, assumed to be constant

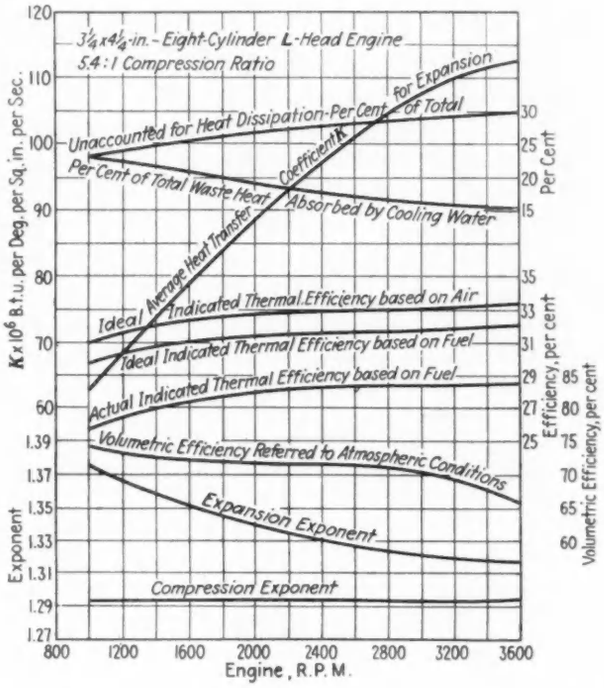


Fig. 8 – Comparison of Analytical Results with Actual Performance

over the speed range, and to evaluate the heat loss from this coefficient, mean combustion-chamber surface, temperature difference, and combustion time. The coefficient used was obtained by assuming the heat loss during combustion at 1000 r.p.m. to be 5 per cent of the heat input. Calculation for the conditions at 1000 r.p.m. showed the corresponding value of coefficient to be more than twice as great as the estimated mean value for compression and expansion at the same speed. Due to the fact that the combustion crank angle period increases at a much lower rate than the speed, the heat loss on the assumed basis was found to drop to 2.5 per cent of the heat input at 3600 r.p.m.

It is obvious that the assumptions made as to combustion heat loss, at the worst, cannot affect the cycle efficiency more than a few per cent.

It should be noted that the heat addition during combustion was determined on the basis of complete combustion of air, which is the condition automatically obtained with sufficient excess fuel for maximum power. Since the mixture ratio in the tests was that corresponding to production full-load carburetor setting, this assumption is believed to be accurate. The heat value of the mixture was, therefore, taken as equal to a weight of gasoline one-fifteenth the weight of air, which is in the chemical combining proportions.

The only loss taken into account, other than the heat losses, is that due to the time required for combustion and escape of gases at the opening of the exhaust valve. The combined effect of these factors is to reduce the area of the theoretical indicator card about 5 per cent at practically all speeds. Accordingly, the actual net work was taken as 95 per cent of the theoretical in all calculations.

B. Results of Analysis

1. Coefficient of Heat Transfer – The estimated mean coefficients of heat transfer derived for expansion are shown in the curve of Fig. 8 over the speed range. Although the trend of the curve is seen to be typical of observed variations in heat-transfer coefficient with velocity, the characteristic can be defined better by plotting the points on log-log coordinates. In transferring the points to the log-log sheet of Fig. 9, the product of r.p.m. and corresponding volumetric efficiency was used as abscissa instead of r.p.m. alone. This method brings the basic charge density into the picture.

Plotted on this basis the four points determined in the analysis plainly lie on a straight line of slope = 0.526, defining the law of variation to be $K = a(NV)^{0.526}$, where N = r.p.m. and V = volumetric efficiency. The value of a corresponding to the absolute values is 2.33×10^{-6}.

It is significant to compare the line for the derived engine values with those obtained by plotting experimental data obtained by R. Royds[3] in tests on compressed air flow in pipes, as shown in Fig. 9. The three lines represent results obtained at different constant air pressures, respectively, 15, 35, and 55 lb. per sq. in. absolute. It will be observed that these results, plotted against the product of velocity and density, or lb. per sec. per sq. ft. passage area, show close agreement for the three air pressures, and a slope varying from 0.578 to 0.615.

Although the values of coefficient for the engine are necessarily plotted against a purely relative factor of speed times density which cannot be compared directly with the absolute basis of Royds' data, the characteristic variation in coefficient follows a very similar law, and the absolute values of coefficient start where Royds' values leave off, indicating a greater order of magnitude of turbulence in the engine than in the air-flow experiments.

It is of interest to note that the value of coefficient assumed for calculating the heat loss during combustion is 167×10^{-6}.

2. Exponents of Compression and Expansion – The practically constant compression exponent arrived at in the analysis and shown in Fig. 9 is accounted for in the previous discussion by its closeness to the adiabatic value. The estimated value is practically constant over the speed range at 1.293. The mean adiabatic exponent, as defined by the temperature-energy curve over the same range of temperature, is 1.307. Thus, we appear to be much closer to the true adiabatic condition on compression than we would expect ordinarily.

The expansion exponent, on the other hand, is shown to be quite sensitive to speed, falling from 1.375 at 1000 r.p.m. to 1.317 at 3600 r.p.m. The corresponding mean adiabatic exponent for the temperature range of expansion is 1.24 to 1.26.

The falling off in the exponent with increase in speed is a direct result of the failure of the coefficient of heat transfer to increase in direct proportion to speed. Since the time duration of the stroke decreases in proportion to the increase in speed, the net heat loss per stroke is reduced, so that expan-

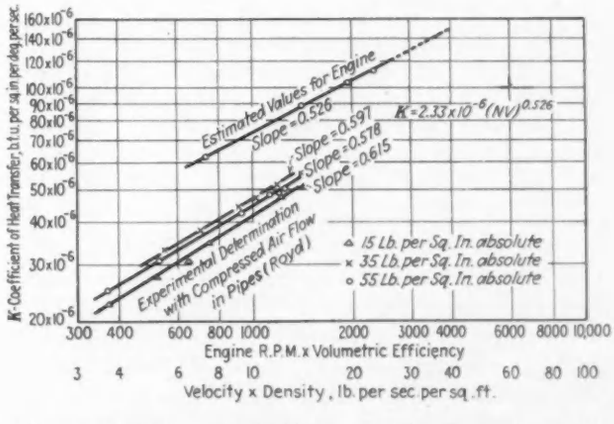


Fig. 9 – Variation of Coefficient of Heat Transfer with Speed and Density

[3] See p. 188: "Heat Transmission by Radiation, Conduction, and Convection," by R. Royds, Constable, London, 1921.

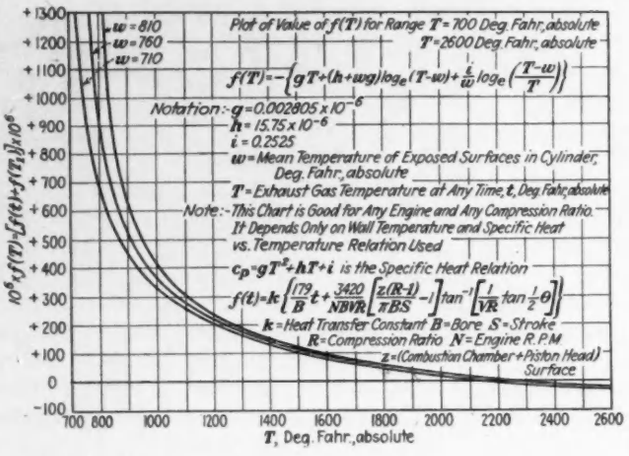


Fig. 10 – Chart for Use in Determining the Exhaust-Stroke Temperature Variation

sion approaches closer to the adiabatic condition. This tendency is also responsible for the increase in thermal efficiency as the speed goes up, namely, from 29.7 per cent at 1000 r.p.m. to 32.1 per cent at 3600 r.p.m., a rise of 8 per cent.

3. Heat Balance.

a. Heat Loss During Exhaust Stroke – In order to complete the picture of heat transfer from gases to cylinder walls it is important to estimate also the heat rejection from the burnt gases during the exhaust stroke. Although this loss, of course, has no bearing on cycle efficiency, it is an appreciable item in accounting for the total measured heat dissipation to the water jackets. It is essential, also, in determining with reasonable accuracy the temperature of the residual exhaust gas in the combustion-chamber.

As brought out earlier in the discussion, density is a basic factor in determining the heat-transfer coefficient. This point means that the coefficient during exhaust, after the bulk of the burned charge has escaped, is necessarily much lower than its mean value during compression and expansion.

It is apparent that correction of the coefficient for the effect of density could not be made before the mean value obtaining on compression and expansion had been estimated. However, the residual exhaust-gas temperature at the end of exhaust stroke must be known in order to calculate the mixture composition and resultant temperature for use in the analysis.

Fortunately, the mixture composition and temperature are not particularly sensitive to the residual exhaust-gas temperature so that it was possible to use a rough approximation for the correction of the coefficient on exhaust without affecting the accuracy of the analytical results.

Even though the coefficient of heat transfer is given, the determination of heat loss on the exhaust stroke is complicated by the fact that both the temperature and the weight of gas vary during the stroke and are related to each other and to the varying rate of heat loss by a complex differential equation. Determination of the heat loss was impossible without a solution to this equation.

The basic equation was set up by equating the heat loss as determined by surface, temperature difference, coefficient of heat transfer, and time, to the loss as obtained from the product of gas weight, specific heat, and temperature change.

It is apparent that each equation contains both temperature and time as variables. In order to obtain a solution, the time and temperature variables were separated, and a function of time, t, obtained which was equal to a function of temperature, T. Although this solution did not give the heat loss directly, it made possible the determination of temperature variation during the stroke from which the heat loss could be obtained by integration.

The curve of Fig. 10 shows the value of the function of temperature over the temperature range from 700 to 2600 deg. fahr. absolute for three different mean wall temperatures. In using this chart to determine the exhaust-gas temperature at any crank angle, the function of time is calculated for the given coefficient and crank angle. Since the change in the value of the two functions is the same, the value of $f(t)$ added to the value of $F(T)$ at the known initial temperature at the beginning of the stroke, gives the value of $f(T)$ at the unknown temperature. The temperature is then determined by reading off the curve at the corresponding value of $F(T)$.

For instance, if the initial gas temperature is 1900 deg. fahr. absolute and the wall temperature 300 deg. fahr., the curve gives the initial value of $F(T)$ to be 40. If the temperature is to be found at 180-deg. crank angle, the $f(t)$ is evaluated from the known constants for $\theta = 180$ deg., and is found to be, say, 240. The value of $F(T)$ at the unknown temperature is, thus, $F(T) = 240 + 40 = 280$. Referring to the curve it is found that the value of 280 corresponds to a temperature of 1100 deg. fahr. absolute. This would then be the gas temperature at the end of the stroke.

The calculation of the coefficient of heat transfer on exhaust stroke is shown in the appendix. It is necessary to cut and try in order to get the mean gas density, on which the coefficient depends, to check with the mean gas temperature during the stroke.

It should be noted that this chart (Fig. 10) is applicable to any engine and for any compression ratio.

b. Total Heat Loss – Comparison of the total calculated direct heat loss from the gas to the walls with the measured heat dissipation is brought out in the curve of "Unaccounted-for Heat Dissipation" of Fig. 8 which shows the difference between the two quantities in per cent of the measured loss. It will be observed that the percentage unaccounted-for by calculation increases with speed. This increase is to be expected since the per cent of total output going into friction, which must be dissipated as heat, increases rapidly with speed and the quantity which is absorbed by cooling water must increase likewise.

On the other hand, if the unaccounted for heat dissipation is considered in relation to the total waste heat as represented by the sum of net heat rejection in exhaust gases and friction, we find that the proportion of total waste heat absorbed by cooling water decreases with increasing speed, as shown in the corresponding curve. It will be observed on Fig. 8 that

R.P.M.	1000			2000			2800			3600		
Input (Ideal)	5330			10340			14350			18890		
Useful Work	1652	(417)	Ind.Eff.	3260	(668)	Ind.Eff.	4760	(1013)	Ind.Eff.	5650	(1645)	Ind.Eff.
Friction		(235)	31%		(695)	38.5%		(1287)	33%		(1985)	30.4%
All Other Losses		3677			6980			9610			11360	
Total Losses			3912			7680			10897			13345
TOTAL		5330			10340			14350			18890	
Itemized Losses												
Heat Loss to Cooling												
Measured		2150			3470			4350			4900	
Heat Loss Calculated												
Compression	30			63			85			106		
Combustion	266			366			407			409		
Expansion	1109			1707			2087			2288		
Exhaust Stroke	246			410			516			630		
Total Calculated	775	1651	1651	1546	2546	2546	3091	3091	3091	3433	3433	3433
Unaccounted for in Cooling Water	235	499		924			1259			1467		
Friction (HP)		826			675			1287			1985	
Net Exhaust Heat												
Rejectio from Cyl.		2086			4434			6519			7885	
T.W.L. Waste Heat			2262			5109			7746			9818
	Total Losses		3912			7655			10837			13345

3½ x 4½ 8 cylinder L-Head Engine.
5.6 Compression Ratio.

Fig. 11 – Itemized Heat Balance in B.T.U. per Min.

the actual percentage of total waste heat picked up by cooling water declined from 22 per cent at 1000 r.p.m. to 15 per cent at 3600 r.p.m., with the percentage tending to approach a constant value at the higher speeds. Although a considerable portion of this waste-heat absorption is due to unavoidable friction, the absorption from the exhaust gases in the ports and that by radiation from the exhaust manifold are important items. The additional load so imposed upon the cooling system is serious enough to justify close attention to detail design of the exhaust ports and manifold with the object of minimizing heat transfer into the water jackets.

It is of interest to examine the distribution of energy as represented in the heat balance of Fig. 11. It will be seen that at 1000 r.p.m. the unaccounted-for heat loss is equal to the sum of the total friction and 13 per cent of the net heat rejected in exhaust. This certainly seems to be an ample allowance for waste-heat absorption and indicates that the assumptions made in the analysis do not exaggerate the rate of heat transfer from gases to cylinder walls.

II. Heat Transfer from Cylinder Surface to Water

The calculated data of heat loss from the gases to the cylinder surface, provided by the foregoing analysis, makes possible an estimate of the temperature distribution over the surface, together with the relative importance of metal conductivity and convection from walls to water in dissipating the heat.

A. Mean Temperature Drop Through Cylinder Walls

The assumed mean internal surface temperature of 300 deg. fahr. is based on the following calculated temperature distribution:

	Total Area, sq. in.	Mean Area During Cycle, sq. in.	Mean Temperature, deg. fahr.
1. Cylinder walls	43.25	21.60	248
2. Combustion-chamber	31.00	31.00	300
3. Piston head	8.25	8.25	436
Total surface	82.50	60.85	300

The total heat flow through the cylinder walls can be calculated by taking the proportion of the total accounted-for heat loss corresponding to the percentage of total surface represented by the mean cylinder-wall surface plus the piston-head area. This method assumes that all heat loss to the piston head is ultimately transferred to the cylinder wall but does not include the heat generated by piston friction which cannot be isolated by calculation.

Thus, if we take the data for 1000 r.p.m.

$$\text{Total heat loss} = \frac{1651}{8} = 206 \text{ B.t.u. per min. per cylinder.}$$

Mean cylinder-wall surface plus piston surface

$$= 21.6 + 8.25 = 29.85 \text{ sq. in.}$$

$$\text{Total heat flow to cylinder walls} = \frac{29.85}{60.85}(206) = 101 \text{ B.t.u. per min.}$$

Mean heat density on cylinder walls

$$= \frac{101}{43.25} = 2.34 \text{ B.t u./sq. in./min.} = 5.6 \text{ B.t.u./sq. ft./sec.}$$

Taking the wall thickness as 5/16 in. and the heat conductivity of cast iron,

$$K = 0.0075 \text{ B.t.u./cu. ft./sec./deg. fahr.}$$

$$\Delta T = \frac{h \times t}{K} = \frac{5.6 \times 5 \times 1}{0.0075 \times 16 \times 12}$$

$$= 19.5 \text{ deg. fahr.}$$

At 3600 r.p.m. the total heat loss is approximately twice as great so that the temperature drop is about 39 deg. fahr.

Since the combustion-chamber surface is exposed to the gases throughout the cycle, the heat density is equal to the total heat loss divided by total mean surface or

Heat density in combustion-chamber

$$= \frac{206}{60.85} = 3.39 \text{ B.t.u./sq. in./min.}$$

or 8.1 B.t.u./sq. ft./sec.

The temperature drop through the wall 5/16 in. thick,

$$\Delta T = \frac{8.1}{0.0075} \times \frac{5}{16} \times \frac{1}{12}$$

$$= 28 \text{ deg. fahr.}$$

At 3600 r.p.m., with twice the heat flow, the drop becomes 56 deg. fahr.

It is evident that comparatively little is to be gained by the use of high-conductivity materials in cylinder-heads, since even an increase of 3:1 in conductivity can reduce the mean internal temperature only about 35 deg. fahr. under high-speed conditions. High-conductivity material does have the advantage of relieving hot-spots and tending to produce greater temperature uniformity by heat conduction along the walls.

B. Heat Transfer from Walls to Water

On the basis of the aforementioned temperature distribution of internal surface and water inlet and outlet temperatures respectively of 145 deg. fahr. and 160 deg. fahr., the heat-transfer coefficient from the external surface to the water can be calculated readily.

Neglecting the relatively small difference between external and internal wall surface and taking the mean heat density for the cylinder walls = 5.6 B.t.u./sq. ft./sec. as before, the coefficient,

$$K = \frac{5.6}{\Delta T}.$$

Since approximately half the total heat flow takes place to the cylinder walls, the water-temperature rise will also be about half the total, giving a mean water temperature in contact with cylinder walls of

$$\frac{145 + 152}{2} = 148.5 \text{ deg. fahr.}$$

Starting with the mean internal surface temperature of 248 deg. fahr. and allowing for the temperature drop through the walls of 19.5 deg. fahr.,

$\Delta T = 248 - 19.5 - 148.5 = 80$ deg. fahr. from jacket surface to water.

$$\text{Then } K = \frac{5.6}{80} = 0.070 \text{ B.t.u./sq. ft./sec./deg. fahr.}$$

$$= 0.000486 \text{ B.t.u./sq. in./sec./deg. fahr.}$$

Applying this heat-transfer coefficient to the combustion-chamber, we get a temperature drop from jacket to water which is higher in proportion to the increased heat density, or

$$\Delta T = \frac{8.1}{0.07} = 116 \text{ deg. fahr.}$$

Checking back to the internal surface temperature of the combustion-chamber, we get

$$\text{Mean water temperature} = \frac{152 + 160}{2} = 156 \text{ deg. fahr.}$$

Internal surface temperature of combustion-chamber = $156 + 116 + 28 = 300$ deg. fahr.

It will be noted that this value is approximately eight times as great as the coefficient from gas to wall during expansion at the same engine speed.

Nevertheless, it is a very low value for heat transfer from

metal to water, particularly as compared with that obtained in tubes. For such cases, typical values are given by the formula[4]

$$K = \frac{0.0004\ (JV)^{0.8}}{D^{0.2}} \text{ B.t.u./sq. in./sec./deg. fahr.}$$

where J = reciprocal of viscosity in centipoises at mean temperature between wall and water.

= 3.52 at 212 deg. fahr.; 2.88 at 180 deg. fahr.; 2.13 at 140 deg. fahr.

V = Water velocity, ft. per sec.

D = Inside diameter of pipe, in.

Comparison of the calculated coefficient of 4.86×10^{-4} with values obtained from the preceding formula shows that the degree of turbulence in the water jacket must be of a very low order. For instance, if we consider the case of a pipe 6 in. in diameter, and $J = 2.88$ corresponding to 180 deg. fahr., substitution of the coefficient value 4.86×10^{-4}, gives a velocity of 0.7 ft. per sec.

C. Overall Temperature Drop

It should be observed from the formula given for heat-transfer coefficient from a tube to water that the coefficient varies as the 0.8 power of velocity. This relation means that the increase in water velocity generated by the pump will tend to compensate for the greater heat density at higher engine speeds. The internal temperatures of directly water-cooled surfaces should, therefore, be subject only to the slight increase due to increased temperature drop through the walls.

The resistances to heat transfer by conduction through the wall and by convection to the water can be combined into a single coefficient according to the relation

$$U = \frac{1}{R_c + R_w}$$

$$R_c = \text{resistance of wall} = \frac{L}{K}$$

$$= \frac{5 \times 1}{16 \times 12 \times 0.0075} = 3.47$$

$$R_w = \text{convection resistance} = \frac{1}{K}$$

$$\text{(at 1000 r.p.m.)} = \frac{1}{4.86 \times 10^{-4} \times 144} = 14.3$$

$$U = \frac{1}{3.47 + 14.3} = 0.056 \text{ B.t.u./sq.ft./sec./deg. fahr. at 1000 r.p.m.}$$

Assuming the convection coefficient to vary as the 0.8 power of water velocity, as in the general formula for heat flow in tubes, and that velocity is proportional to engine speed, a general expression can be written for the combined coefficient

$$U = \frac{1}{3.47 + 14.3 \left(\frac{N}{1000}\right)^{0.8}} \text{ B.t.u./sq. ft./sec./deg. fahr.}$$

III. Piston-Head Temperature

The piston head provides an interesting problem in heat conduction because both the quantity of heat flow and the area of heat path vary continuously from the center to the periphery. It can be assumed safely that practically all heat dissipation from the piston takes place between the rings and the cylinder wall; also, that the heat flow to the piston head is uniform over its surface.

In this case, the equation for heat conductivity must be written in differential form

$$Q = KS \frac{dT}{dL} \text{ or } dT = \frac{Q}{KS} dL$$

[4] See *Industrial and Engineering Chemistry*, Vol. 14, 1922, pp. 1101-1105: "Heat Transfer by Conduction and Convection," by W. H. McAdams and T. H. Frost.

Considering the conditions at a given radius, r, over an increment of distance, dr,

$Q = h \pi r^2$ where h = heat flow density, B.t.u./sq. in./sec.

$S = 2 \pi rt$ where t = head thickness

and $dT = \frac{h}{2tK} rdr$

Integration from $r = 0$ to $r = R$, the outer piston radius, gives, for uniform head thickness, t:

$$T_o - T_R = \frac{hR^2}{4tK}$$

It is evident from the foregoing development that a uniform head thickness represents inefficient use of material, since the heat density increases directly with radial distance from the piston center. For this reason, the inner surface of the head is usually made conical with thickness increasing from a minimum at the center to a maximum at the periphery.

The variation in thickness can, therefore, be written,

$t = a + br$ where a = thickness at center.

b = increase in thickness per inch of radius.

Substituting this expression in the basic equation and integrating, gives

$$T_o - T_R = \frac{h}{2K} \left[\frac{R}{b} - \frac{a}{b^2} \log_e \frac{a + bR}{a} \right]$$

In applying these equations to the engine in question, we can assume the rate of heat flow to the piston to be the same as to the combustion-chamber walls without serious error because, although the mean piston temperature is appreciably higher than that of the chamber walls, the difference is not great relative to the mean temperature difference between the gases and the surface. The basic data then become

h = 8.1 B.t.u./sq. ft./sec. at 1000 r.p.m.

R = 1⅝ in.

K = 0.0075 for cast iron, = 0.025 for aluminum } B.t.u. per cu. ft./sec./deg. fahr.

a = ⅜ in. thickness at center

$b = \frac{⅜}{1⅝} = 0.115$ corresponding to ⅜ in. thickness at periphery.

Substituting in the equation, we get for iron, at 1000 r.p.m.

$$T_o - T_R = \frac{8.1}{24 \times 0.0075} \left[\frac{1.625}{0.115} - \frac{0.187}{0.115 \times 0.115} \log_e \frac{0.375}{0.187} \right]$$

$$= 193 \text{ deg. fahr.}$$

If the temperature, T_R = 340 deg. fahr., T_o = 533 deg. fahr. } $T_m = 436$

At 3600 r.p.m., $(T_0 - T_R)$ becomes twice as great or 386 deg. fahr., corresponding to a temperature of 726 deg. fahr. at the piston center.

By using aluminum, having about three times the conductivity of iron, the temperature drop through the head is cut to one-third. Therefore, at 1000 r.p.m., the temperature drop becomes $\frac{193}{3} = 64$ deg. fahr. and, at 3600 r.p.m., 128 deg. fahr. The corresponding piston-center temperatures would be, respectively, 404 deg. fahr. and 468 deg. fahr. The great advantage of aluminum in this application is apparent, in addition to the weight reduction which it makes possible.

Conclusion

1. The method for evaluating coefficients of heat transfer from the gases to the internal engine surface is presented in the belief that it yields values that are reasonably close to the truth.

2. The actual values derived for the test engine are probably representative, but should not be accepted as generally

applicable because of the great many possible variables among different engines.

3. The most significant specific conclusions arrived at from this analysis may be summarized as follows:

a. The fact that the coefficient of heat transfer from the gases increases at a lower rate than piston velocity and charge density means greater thermal efficiency the higher the speed and the volumetric efficiency. As a corollary, it follows that thermal efficiency tends to be reduced at part loads, since the heat losses do not decrease in the same proportion as the weight of charge handled.

b. The cylinder-surface-to-volume ratio is a primary factor limiting engine thermal efficiency because it determines the percentage of heat rejection from the working gases to the cooling water.

It follows that, from the efficiency standpoint, fewer and larger cylinders are preferable to more and smaller cylinders for a given total displacement. On the other hand, limitations of design dictate a trend to greater multiplicity of cylinders in striving for greater volumetric efficiency and higher speeds. Since these objectives in themselves tend to increase thermal efficiency, a net gain may result in spite of higher surface-to-volume ratio. However, multiplication of cylinders without corresponding improvement in specific output is definitely detrimental to efficiency.

c. The use of high conductivity materials for the most part does not pay dividends commensurate with cost where the surfaces are directly water cooled. However, they offer great improvement where only indirect cooling is available.

Acknowledgment is made to Lee Baker of the Chrysler Research Division for his invaluable assistance in making the great volume of detail calculations required in this analysis; also for his ingenious mathematical solution of the problem of determining heat loss during the exhaust stroke.

Appendix

Outline of Cycle Analysis for Evaluation of Heat-Transfer Coefficients

I. Required Test Data

1. Brake Hp. = B.Hp.
2. Friction Hp. = F.Hp.
3. Fuel Consumption = F.
4. Air Consumption, cu. ft. per min. = V.
5. Carburetor Air Temperature, deg. fahr. absolute = T_a
6. Barometric Pressure $\times$ 0.491 = P_a, lb. per sq. in.
7. Manifold Vacuum.

II. Procedure

I.Hp. = B.Hp. + F.Hp.

(1) $\text{I.M.E.P.} = \dfrac{792{,}000\ \text{I.Hp.}}{DN}$ D = Piston Displacement, cu. in. N = R.P.M.

(2) Volumetric Efficiency Referred to Carburetor Air Conditions

$$\text{V.E.} = \frac{3456V}{DN}$$

(3) To Find Fresh Charge Temperature (T_o) and Pressure (P_o) Assumed to Exist in Cylinder Before Mixing with Exhaust Diluent.

$$V.E. = \frac{T_a}{P_a} \times \frac{P_o}{T_o} \quad \text{hence} \quad \frac{P_o}{T_o} = \frac{P_a}{T_a} \times (V.E.)$$

Choose P_o in Reasonable Agreement with Manifold Vacuum.

(4) Mixture Composition

(a) Weight of Air per Min., $A = V \times 0.0764 \times \dfrac{520}{T_a} \times \dfrac{P_a}{14.7} = \dfrac{2.7 P_a V}{T_a}$

(b) Air-Fuel Ratio $r = \dfrac{A}{F}$

(c) Exhaust Weight per Pound of Air.

$$w_e = \frac{P_a}{P_o} \times \frac{T_o}{T_e} \times \frac{R_a}{R_e} \times \frac{1}{CR-1}$$

Assume an Exhaust Diluent Temperature of T_e

Assume Exhaust Pressure = P_a

CR = Compression Ratio

R_a = 0.0686 = Gas Constant for Air

R_e = 0.0709 = Gas Constant for Exhaust Gases

(d) Fuel Weight per Pound of Air $f = \dfrac{1}{r}$

(e) Composition per Pound of Mixture:

Air, lb., $W_a = \dfrac{1}{1 + w_e + f}$

Fuel, lb., $W_f = \dfrac{f}{1 + w_e + f}$

Exhaust, lb., $W_e = \dfrac{w_e}{1 + w_e + f}$

Fresh Charge, lb., $W_{fc} = W_a + W_f$

$$R_m = 1.985 \left[\frac{W_a}{28.85} + \frac{W_f}{112} + \frac{W_e}{28} \right]$$

(5) Pounds of Mixture per Minute, $W_m = \dfrac{A}{W_a}$

(6) Displaced Volume per Pound of Mixture

$$V = \frac{W_a}{0.0764} \times \frac{14.7}{520} \times \frac{T_o}{P_o}$$

(7) Required Net Work per Pound of Mixture

$$E_n = \frac{\text{I.M.E.P.} \times V \times 144}{778} = \frac{V \times \text{I.M.E.P.}}{5.4}$$

(8) Determination of Temperature of Mixture (T_1) at Beginning of Compression From the Relation

$$(S_{T_1} - S_{T_o})\, W_{fc} = (s_{T_e} - s_{T_1})\, W_e$$

where S = Sensible Heat of Fresh Charge

s = Sensible Heat of Combustion Products.

Subscripts Indicate the Temperatures at which the Heat Contents Are Taken.

(9) Pressure at Beginning of Compression

$$P_1 = \frac{wRT}{V} = \frac{W_m \times R_m \times 778 \times T_1}{\dfrac{D}{1728} \times \dfrac{N}{2} \left(\dfrac{CR}{CR\text{-}1} \right) \times 144} = \frac{1.87 \times 10^4\, W_m\, R_{fc}\, T_1\, (CR\text{-}1)}{DN\, (CR)}$$

(10) Compression Relations:

Assume Compression Exponent n_c

$$P_2 = P_1\, (CR)^{n_c} \qquad T_2 = T_1\, (CR)^{n_c - 1} \qquad E_{1.2} = \frac{R_{fc}}{n_c\text{-}1} \Delta T_{1\text{-}2}$$

$$\frac{T_2}{-T_1} \qquad \frac{S_2}{-S_1} \qquad \frac{E_{1.2}}{\Delta S_{1.2}} \qquad \frac{}{\Delta Q_{1.2}} \qquad E = \text{Work}$$

(11) Combustion: Assume Heat Transfer Constant

$K = 167 \times 10^{-6}$ B.t.u./sq. in./deg./sec.

Assume Temp. T_3 and Mean Wall Temp.

T_w = 300 deg. fahr.

$$\Delta Q_{2.3} = 0.000167 \times \frac{Z}{2} (T_3 - T_w) \times \frac{(\Theta_a + 20)}{360} \times \frac{N}{2} \times \frac{60}{N}$$

$$= 0.000167 \frac{Z}{2} (T_3 - T_w) \frac{(\Theta_a + 20)}{12}$$

where Θ_a = Deg. of Spark Advance

Z = Combustion-Chamber and Piston-Head Surface.

Fuel Burned $W_{fb} = \frac{W_a}{15}$

Energy Released $= \Delta H_{2.3} = \frac{W_a}{15} \times 18{,}500$

Change in Sensible Heat $\Delta S_{2.3} = \Delta H_{2.3} - \Delta Q_{2.3}$

Energy Content after Combustion $s_3 = s_2 + \Delta S_{2.3}$

(12) Expansion: Assume n

T_3 Corresponds to s_3 as read from the T-S Chart for Combustion Products (Check back to assumed value)

$$P_3 = P_2 \frac{T_3}{T_2} \times \frac{R_e}{R_{fc}}$$

$$T_4 = T_3 (CR)^{1-n_e}$$

$$P_4 = P_3 (CR)^{-n_e}$$

$$E_{3.4} = \frac{R_e}{n_e - 1} \Delta T_{3.4} \qquad \frac{T_3}{-T_4} \qquad \frac{s_3}{-s_4} \qquad \frac{\Delta S_{3.4}}{-E_{3.4}}$$

$$\frac{}{\Delta T_{3.4}} \qquad \frac{}{\Delta S_{3.4}} \qquad \frac{}{\Delta Q_{3.4}}$$

(13) Exhaust:

Assume Adiabatic Expansion of Exhaust from P_4 to P_a, then Constant-Pressure Cooling through Exhaust Stroke. Temperature after Adiabatic Expansion (T_5), get Mean Adiabatic Exponent for the Range from the T-S Chart using

$$\gamma - 1 = R_e \frac{\Delta T}{\Delta S}$$

(*Continue reading at top of opposite column*)

$$T_5 = T_4 \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

Method of Obtaining K_e

Ratio of Mean Densities on Exhaust and Expansion

$$\text{Density Ratio} = \frac{\frac{1}{2}(CR-1)+1}{CR} \times \frac{2T_1}{T_5+T_e} \times \frac{28}{30.5} \times \frac{P_a}{P_1}$$

where $\frac{28}{30.5}$ = Molecular Weight Ratio of Exhaust to Combustible Mixture.

Multiply the Speed by the Density Ratio and the Volumetric Efficiency To Obtain a Corrected Speed × Volumetric Efficiency Value.

Read K'_e from Curve of Fig. 10.

Correct K'_e for mean constant pressure specific heat:

$$K_e = K'_e \frac{[C_{pm}]\frac{T_5}{T_e}}{[C_{pm}]\frac{T_4}{T_3}}$$

At full load $\frac{K_e}{K'_e}$ is practically constant at about 0.8 over the speed range.

Evaluate $f(t)$ at desired values of t during stroke.

Evaluate $f(T_5)$ from curve sheet (Fig. 10).

Then $f(T) = f(t) + f(T_5)$

and temperatures during exhaust may be read from the curve sheet knowing $f(T)$ for each point.

Check final exhaust temperature against original assumption of T_e.

Then heat loss is obtained by integration method as follows. Surface A is obtained from Fig. 4.

Θ	A = Exposed Surface	t = time, sec.	$f(t)$	$f(t) + f(T_1)$	T	$\Delta T = T - T_w$	$A \times \Delta T$
0							
20							
↓							

Θ = Degrees Crank Angle

Plot $A\Delta t$ and Integrate with Planimeter.

If Integration Area = B sq. in.

and 1 In. Vertical = M deg. sq. in.

1 In. Horizontal = Q sec.

Then:

1 sq. n. = MQ deg. sq. in. sec.

(*Continue reading on opposite column*)

$$\Delta Q_e = BMQ \times K_e \times \frac{N}{2} \text{ B.t.u. per min.}$$

(14) Check Output: 5 Per Cent Indicator-Card Loss due to Rounding.

Net Work $E_n = (E_{3.4} - E_{1.4})\ 0.95$

Must check item (7).

(15) Graphical Integration Areas for Compression and Expansion.

Θ	Surface A	Relative Volume	T_c	$\Delta T_c = T_c - T_w$	$A\Delta T_c$	T_e	$\Delta T_e = T_e - T_w$	$A\Delta T_e$
0								
20								
40								
↓								

Plot $A\Delta T$ Values and Integrate with a Planimeter

As before, for Area B sq. in.

Scale Constant MQ $\frac{\text{deg. sq. in. sec.}}{\text{sq. in.}}$

Heat Loss $\Delta Q = K \frac{N}{2} (BMQ)$

If Compression Area is B_c and Expansion Area B_e.

then $\frac{\Delta Q_{1.2}}{\Delta Q_{3.4}} = \frac{B_c \times 0.80}{B_e}$ must be true for Balance.

(*Continue reading on opposite column*)

(16) Calculated Heat Losses, B.t.u. per min. and Heat-Transfer Constant

$\Delta Q_c = \Delta Q_{1.2} \times W_m$ $\qquad$ $\Delta Q_e = \Delta Q_{3.4} \times W_m$

$$K_c = \frac{2\Delta Q_c}{N(B_c MQ)} = 0.8K_e = \frac{1.6\Delta Q_e}{N(B_e MQ)}$$

(17) Efficiency of Cycle

Based on Air $e_a = \frac{E_n}{\Delta H_{2.3}}$

Based on Fuel $e_f = \frac{E_n}{W_f \times 18{,}500}$

March of Progress in the Development of Transportation

By Pierre Schon

General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co.

IN the first part of this paper, read at the March 3, 1938, meeting of the Baltimore Section of the Society, the author discussed the advantages of cab-over-engine trucks as compared with the engine-under-the-hood type, covering much of the same ground included in his earlier paper: "Cab-Over-Engine Trucks – Their Place in Transportation," published on pages 421-427 of the September, 1937, S.A.E. TRANSACTIONS.

At Baltimore Mr. Schon also traced the evolution of transportation by depicting examples of various forms of transportation at different stages in their development; portrayed the progress in allied industries; evaluated the uncontrollable expense increases from greater regulation, higher taxes, and increased pay-rolls; and touched upon dangerous practices and maintenance methods.

In the remainder of the Baltimore Section paper, which follows in full, Mr. Schon interprets the trend of commercial-vehicle regulation in the various States, aided by tables showing 1937 changes. He then states the case of four million truck owners in opposition to proposed drastic grade-climbing-ability regulations, showing what their effect would be on the size of engines and other torque-transmitting parts, and upon truck prices, operating costs, and freight rates. Safety features claimed to result from such legislation are scrutinized critically.

THE natural evolution of the "March of Progress in the Development of Transportation" is retarded occasionally by artificial obstacles. One hundred years ago stage-coach and canal-boat operators, motivated by the instinct of self-preservation and in order to protect their business, attempted to arouse public opinion against the development of the railroads. Similarly a most vigorous campaign of propaganda has been directed against highway transport, resulting in regulatory acts such as the notorious Texas weight law where a truck is limited to a payload of 7000 lb. if it passes more than one railroad station, but the law allows 14,000 lb. payload if the truck is used to bring freight to the railroad depot.

The anti-truck campaign reached its peak during 1930 to 1936, when amendments to existing State laws imposed various restrictive regulations on commercial vehicles. That the pendulum is swinging in the other direction is evidenced by the changes made in 1937 when 43 States were in regular session:

Legal Vehicle Size Changes in 1937

Height – Code 12 ft. 6 in.		
South Dakota,	from	11 ft. 6 in. to 13 ft. +
Kansas,	from	12 ft. to 12 ft. 6 in. +
Pennsylvania,	from	14 ft. 6 in. to 12 ft. 6 in. —
Washington,	from	No restriction to 12 ft. 6 in. —
Length of Single Vehicle – Code 35 ft.		
South Dakota,	from	28 ft. to 30 ft. +
Maine,	from	36 ft. to 40 ft. +
New Hampshire,	from	30 ft. to 33 ft. +
Iowa,	from	30 ft. to 33 ft. +

In the height regulations, only two reductions were made and, in both cases, the new laws are in line with the Uniform Code recommendation of 12½ ft. In the single vehicle lengths, four States increased previous limitations, however, the code recommendation of 35 ft. was disregarded entirely in South Dakota and Maine.

Legal Vehicle Size Changes in 1937 – Combinations

Length for Tractor Semi-Trailer – Code 35 ft.		
Maine,	from	36 ft. to 40 ft. +
Kansas,	from	50 ft. to 35 ft. —
Pennsylvania,	from	70 ft. to 45 ft. —
South Dakota,	from	38 ft. to 40 ft. +
Length for Combinations – Code 45 ft.		
Kansas,	from	50 ft. to 45 ft. —
Pennsylvania,	from	70 ft. to 50 ft. —
South Dakota,	from	38 ft. to 40 ft. +
Washington,	from	85 ft. to 60 ft. —

The tractor semi-trailer length limit of 35 ft. is a controversial issue, as the Uniform Code classifies it as a single vehicle although it actually is a combination of two entirely different units. Kansas adopted the Code recommendation of 35 ft., whereas Maine and South Dakota changed to 40 ft. and Pennsylvania to 45 ft.

For other combinations, Kansas followed the Code recommendation, whereas the other three States disregarded the Code.

1937 Changes in Vehicle Weight Laws

Axle Weights – Code 18,000 lb.				
Idaho,	from	16,000 lb.	to 18,000 lb.	+
Iowa,	from	16,000 lb.	to 18,000 lb.	+
Maryland,	from	No restriction	to 18,000 lb.	—
New Hampshire,	from	15,000 lb.	to 16,000 lb.	+
Washington,	from	18,500 lb.	to 18,000 lb.	—
Gross Weight Four-Wheel Truck				
Idaho – Axles less than 13 ft.,	from	24,000 lb.	to 22,000 lb.	—
Axles 13 ft. or more,	from	24,000 lb.	to 28,000 lb.	+
Maryland – Solid Tires,	from	25,000 lb.	to 22,000 lb.	—
Pneumatic Tires,	from	25,000 lb.	to 26,000 lb.	+
New Hampshire,	from	23,000 lb.	to 26,000 lb.	+
California,	from	22,000 lb.	to 26,000 lb.	+

The Uniform Code recommends a maximum axle weight of 18,000 lb. for low-pressure pneumatic tires and a limitation of 16,000 lb. for other tires. Laws were changed in five States with a most noticeable trend toward adopting the Uniform Code recommendations. Note four-wheel truck weight changes in four States.

1937 Changes in Vehicle Weight Laws – Combinations

Gross Weight Six-Wheel Truck				
Idaho,	from	50,000 lb.	to 42,000 lb.	—
Maryland,	from	40,000 lb.	to 38,000 lb.	—
New Hampshire,	from	23,000 lb.	to 38,000 lb.	+
Gross Weight Tractor Semi – Three Axles				
Idaho,	from	*L* plus 40	to 42,000 lb.	+
New Hampshire,	from	35,000 lb.	to 38,000 lb.	+
Florida,	from	22,000 lb.	to 34,000 lb.	+

Gross weight reductions for the six-wheel truck were enacted in two States, while a substantial increase was allowed in New Hampshire.

The three-axle tractor semi-trailer was given an increase in three States, and the same trend applies to changes in the weight laws affecting other combinations and maximum gross weights. It is interesting to note that Idaho abandoned the *L* plus 40 formula as recommended in the Uniform Code.

Gross Weight – Maximum for Combinations

Idaho, from *L* plus 40 to 68,000 lb.
Kansas, from 62,000 lb. to 63,000 lb.
Maine, from 36,000 lb. to 40,000 lb.

The results of 1937 legislative activities constitute a most complimentary tribute to the well organized Truck Owners' Associations in the various States, and they also indicate that the trucking industry is capable of meeting and defeating any and all attempts to hamstring the future development of highway transportation. With only 9 States convening for regular session during 1938, no drastic anti-truck bills are pending on the State legislative calendars up to now, and there seems to be little cause to worry about the legislative situation for 1938. As a matter of fact, in several Southern States, bills are pending for more liberal weight laws, that is, Kentucky, Alabama, and Mississippi.

Legal Performance Requirements

There has been considerable agitation for a new type of regulation, however, which, if enacted on the basis as proposed by certain anti-truck propagandists, would positively wreck the entire trucking industry. I am referring to the "Legal Performance Requirement," also known as the "Grade Ability" proposal, which would require a loaded motor truck to climb a certain grade at a speed of 20 m.p.h. There is no difference of opinion on the 20 m.p.h. speed, but the joker is in the per cent of grade, ranging from 3 to 6 per cent.

In order to get a better understanding of the situation, perhaps a chronology of events will bring us up to date on this subject.

It must be admitted that, during the depression years when money was scarce, investments in motor trucks were held down to the lowest figure by many operators. This policy resulted in placing on the highways a substantial number of vehicles with insufficient engine power to climb the grades at a reasonable speed, and with not enough braking ability to hold safely the loaded vehicles going down the hills. However, these vehicles are less numerous today, for the operators found out that high operating expenses resulted from this practice.

The danger of abnormally slow vehicles on the highways also was realized by the National Conference on Street and Highway Safety which convened in Washington during the early part of 1934 and, as a result of committee deliberations, the conference recommended a 3 per cent grade ability at 20 m.p.h. for commercial vehicles. However, when this recommendation was submitted for vote in the final meeting of the conference on May 24, 1934, the anti-truck faction succeeded in packing the delegation and the committee recommendation for a 3 per cent grade ability regulation was amended by majority of "Viva Voce" vote, resulting in the Uniform Code, Act 5, Page 38, reading as follows:

Uniform Code, Act 5, Page 38

"A suggested method of determining whether the motive power is adequate to propel the vehicle and load at a reasonable speed is to require that the vehicle be powered so that it can ascend a grade of — per cent at a speed of not less than 20 m.p.h. A speed of 20 m.p.h. for commercial vehicles is considered reasonable on an ascending grade. The percentage gradient to be inserted should be representative of conditions within each particular State. In a State of flat terrain a 3 per cent grade should be adequate. In hilly and mountainous territory the grade should be 5 or 6 per cent."

One Eastern State attempted to impose this regulation in 1935, but the enforcement was abandoned when it was found that no tractor semi-trailers and but few of the heavier trucks, when fully loaded, had sufficient grade ability at a speed of 20 m.p.h. to comply with this regulation.

The subject was studied by the U. S. Bureau of Public Roads, resulting in a report to the American Association of State Highway Officials in 1936, reading as follows:

Section 8 – Climbing Ability

"The addition of the following recommendation is suggested:

"(a) On and after ——— all motor vehicles and combinations of vehicles shall be capable of climbing a concrete pavement incline of 5.0 per cent with a sustained speed of not less than 20 m.p.h. when loaded with maximum legal load."

This recommendation resulted in closer investigation of the subject by manufacturers and operators in order to find out just what a 5 per cent grade ability regulation would do to present-day gross-weight and payload ratings; how it would affect the four million motor trucks in the hands of private and commercial carriers, and also the future design of new vehicles.

It is evident from the table of figures on the opposite page that much larger engines would be required as the per cent of grade is increased. A bigger engine means heavier chassis weight as all torque-transmitting parts and other component chassis units must be larger and heavier to stand the strains of the greater engine power.

The price of vehicles also increases whenever larger engine power is specified and, in the history of transportation, high speed invariably means a higher price for the service. A trip on a fast ocean liner costs considerably more than on a tramp

Engine Size and Approximate Gross Weight Limitation for Vehicles Traveling 20 M.P.H. on Grades of

Engine Displacement, cu. in.	3 Per Cent	4 Per Cent	5 Per Cent	6 Per Cent
200	21,000	17,000	14,200	12,300
225	24,000	19,000	16,700	14,400
250	26,000	21,000	18,100	15,600
275	28,000	23,000	19,000	16,200
300	30,000	24,000	20,000	17,700
325	32,000	25,300	21,200	18,200
350	34,000	26,500	22,200	19,300
375	36,500	27,000	23,500	20,500
400	38,600	28,000	24,500	21,700
425	39,500	29,000	25,500	22,500
450	41,000	31,500	26,300	23,600
475	42,200	33,000	27,900	24,500
500	43,500	34,200	29,500	25,400
525	44,500	36,000	30,000	26,200
550	46,000	37,800	31,500	27,000
575	47,500	38,600	32,000	27,900
600	49,000	39,900	33,800	28,700
625	51,500	40,600	34,200	29,600
650	53,000	41,200	35,000	30,500
675	54,100	43,000	36,000	31,200
700	55,000	44,300	37,100	32,000

steamer and the same applies to the 20th Century Limited with its excess fare between New York and Chicago; to the main liners in aviation, down to the difference in the price between a sixteen-cylinder Cadillac and a Chevrolet.

It is a foregone conclusion that larger engines and higher performance ability will increase greatly the operating cost of a motor truck and, unless trucking rates can be increased in relative proportion, any regulatory measure of this type would spell ruination to the entire industry.

Taking the Maryland 26,000-lb. weight law as an example, a four-wheel truck on pneumatic tires traveling at 20 m.p.h. on various grades would be able to carry a payload at a cost per ton-mile as indicated in the following table:

Maryland 26,000-Lb. Gross Weight

Grades	3 Per Cent	4 Per Cent	5 Per Cent	6 Per Cent
Engine Displacement, cu. in.	250	350	450	525
Truck Investment	$2,500	$3,500	$5,000	$7,000
Chassis Weight, lb.	6,000	8,000	10,000	12,000
Body and Payload, lb.	20,000	18,000	16,000	14,000
Body Weight, lb.	2,000	2,000	2,000	2,000
Net Payload, lb.	18,000	16,000	14,000	12,000
Net Payload, tons	9	8	7	6
Operating cost per mile at 150 miles per day	$0.1349	$0.1494	$0.1707	$0.2046
Cost Per Ton	$2.25	$2.80	$3.65	$5.11
Cost Per Ton-Mile	$0.0160	$0.0186	$0.0243	$0.0340

The preceding figures indicate the need for careful study of this proposed regulation, especially considering the fact that the trucking industry, at this time, is not financially able to discard present vehicles in operation, nor is the commercial carrier in a position to stand such drastic increases in the operating cost at existing freight rates.

After the 5 per cent grade ability recommendation was submitted in 1936, vigorous protests were raised against this drastic proposal and the U. S. Bureau of Public Roads initiated a physical test study on U. S. Highway No. 40 near Baltimore, having a uniform grade of 6 per cent for approximately 2000 ft.[1]

This study revealed that approximately 95 per cent of the trucks tested were unable to climb this 6 per cent grade at a speed of 20 m.p.h. In other words, a 6 per cent grade ability performance regulation would condemn the majority of commercial vehicles now in use to the junk pile. Of course, there is an alternative: *"the payload can be reduced"* to give a better grade ability to the four million motor trucks in use today.

In reality, a 5 or 6 per cent grade ability regulation is nothing more or less than the notorious Texas weight law, disguised as a safety measure.

When we speak of the *necessity for regulation* from the standpoint of safety, why concentrate on the motor truck when other slow vehicles on the highway are just as objectionable, and perhaps much more dangerous? What about the slow passenger-car driver, the old car relics, the farm wagons, which together are far more numerous and therefore constitute much greater hazards on the highway?

In a recent study on this subject conducted by a large driveaway company employing about 300 drivers, replies to questionnaires revealed that these professional drivers consider highway hazards in the following sequence of importance:

Highway Hazards in the Eyes of the Truck Driver

1. Slow passenger cars.
2. Horse-drawn vehicles without lights.
3. Other slow trucks.
4. House trailers attached to small passenger cars.

An antique four-cylinder car with abnormal play in the steering-gear connections, ambling along at 20 m.p.h., invariably swerving to the center of the road at the sound of a horn and with the driver then stepping on the gas when you try to pass him, perhaps constitutes the greatest danger on our highways today.

There is another element which has been overlooked in this agitation against the motor truck, that is, will the passenger-car driver coming along at a speed of 50 or 60 m.p.h. curb his desire for passing a slow truck? Will he slow down to 20 m.p.h. or will he attempt to pass any vehicle traveling at a slower speed than his own car?

Supposing he decides to pass the slower vehicle, will he be safer if the truck is traveling at 5 m.p.h. or at 20 m.p.h.? Don't let us overlook the fact that you and I and 29 million other drivers will *not* crawl along behind a slower-moving vehicle whenever we feel that we can pass by *safely*, and by safely I mean what the individual thinks in his own mind is a safe time to step on the gas and get ahead. Now we all know that we can pass a slow vehicle in less time and within a shorter distance of travel than a vehicle which moves along at a faster speed. For instance, in seconds of time and feet of travel, we note quite a difference for a car traveling at 45 m.p.h. in passing slow vehicles traveling at speeds of 5, 10, 15, and 20 m.p.h. respectively.

Passing Time and Distance Required at 45 M.P.H.

Truck Speed, m.p.h.	Truck Travels, ft.	Truck Length Plus Travel	Time, sec.	Distance for Passenger Car
5	10.3	40.3	1.4	92.2
10	23.5	53.5	1.6	105.5
15	41.0	71.01	1.87	123.2
20	65.5	95.5	2.24	147.77

[1] See S.A.E. TRANSACTIONS, May, 1938, pp. 209-214: "The Outlook on Legal Performance Requirements," by J. Trueman Thompson.

In this comparison, the following dimensions were used:

Passenger-car length	17 ft.
Space of turn out	15 ft.
Truck length	30 ft.
Turning back in line	20 ft.
	82 ft.

Perhaps too much emphasis is placed on the motor truck from the safety angle, and insufficient attention has been given to the fact that highway planning and building has not kept step with the expansion of usage of the motor vehicle.

Only a very small percentage of our three-million-mile highway system can be classed as automobile roads, the great majority are still of the same design as laid out originally to accommodate the horse and wagon traffic – narrow winding trails. It is true that we have placed a harder topping on some of them; yet, from the safety standpoint, our road system is 25 years behind the progress which has been made by the automotive industry.

Type of Highways in U. S. A.	Miles (Approximate)
Total Highway Mileage	3,000,000
Unsurfaced mud roads, miles	2,000,000
Total surfaced, miles	1,000,000
Loose-gravel surfaced, miles	886,300
Hard-top surface, miles	113,700
Two-lane major highways, miles	105,700
Wider than two lanes, miles	8,000
Lighted highway, miles	100

Highway hazards created by slow-moving vehicles are minimized or eliminated entirely on multiple-lane highways. All slow-moving vehicles cannot be legislated off the highways by attempts to prescribe impractical grade-ability regulations for motor trucks, unless such regulations also apply to the much more numerous slow passenger cars and the thousands of horse-drawn vehicles in the rural areas.

Traffic congestion has reached the point in certain areas where it ceases to be a regulatory problem. Highway and traffic engineers know how to build roads that will solve our traffic problems. This layout with a dividing island down the middle separating opposing streams of traffic, is a typical example of what highway engineers could do in the interest of highway safety if sufficient funds were available.

If the 300 million dollars of the motorist tax moneys which are being diverted every year for fish hatcheries, propagation of oyster beds, eradication of the cattle tick, doles for relief, and dozens of other purposes not related in any way to highway construction and maintenance, could be used for the purpose intended, many of the dangerous bottlenecks in our present highway system could be widened out in accordance to the traffic needs.

Truck transportation, slow or fast, is an integral part of our social and industrial structure, and these four million trucks pay for the use of the highways at the tune of over 400 million dollars per year. A slow truck has just as much right on the highway, from the taxpayers' standpoint, as has the passenger car.

To make our highways safer, we need more and better highways; roads adequate for the 29 million cars we have placed upon them; a system of super-highways; and multiple-lane highways with a separation in the center to segregate the two streams of traffic rushing at 60 m.p.h. in two different directions.

Too little attention has been given by highway authorities to the removal of "death traps and bottlenecks" which sometimes can be eliminated at a reasonable cost. Extensive studies on this particular subject are being conducted now in Michigan by its Highway Commissioner, Pat Van Wagoner.

Traffic engineers claim that they could eliminate 90 per cent of the accidents and congestion. The approach to the George Washington Bridge, between New York and New Jersey, with its bypasses and scientific planning, relieves congestion and allows a steady flow of traffic in and out of New York City.

At the present time there is a bill pending in Congress calling for the construction of a network of super-highways. This plan, if added to the regular road-building programs, eventually will help to bring our highway system up to par, and adequate to meet the needs of the "March of Progress in the Development of Transportation."

Conclusion

In view of the far-reaching and destructive effects on the trucking industry through the enactment of impractical and drastic grade-climbing ability regulations, the owners of four million trucks are entitled to the following information:

Question No. 1 – What interests are back of this agitation demanding drastic reductions in truck gross weights?

Question No. 2 – Is this proposal intended to speed up the flow of traffic on the highways, if so, will such a regulation also affect slow vehicles other than trucks?

Question No. 3 – Are the truck operators aware of the fact that a 5 or 6 per cent grade ability regulation will reduce payloads 1/3 to ½ on a large number of present vehicles, and actually prohibit the use of all tractor semi-trailers and other combinations?

In fairness to an industry which renders an indispensable and valuable service to the general public, we believe that any proposal which threatens to double the cost of the service rendered – threatens to result in drastic increases in the cost of transportation – should be analyzed most carefully from all angles, other than partial and perhaps imaginary highway safety reasons.

Processing Engine Bearings

DIE-CASTING of bearing metals leaves very little to be desired due to the nature of the method. It is easy to control segregation and crystal structure, except in the case of parts like the connecting-rods that have light and heavy sections. In cases of this kind, the core must be water-cooled to accelerate cooling in that direction. Then, if too much metal is removed from the chilled surface, we are very apt to remove the best part of the bearing. Centrifugal casting presents other problems. If sufficient data are at hand, the control of segregation is not difficult. These data include speed of rotation and temperature of the metal in relation to the diameter of the tubing being cast. If either speed or temperature is too high, segregation cannot be prevented. The result is a bearing with no copper-tin crystals to support the shaft. Instead, we have a bearing of almost pure tin, unless we pour a heavy lining of babbitt into the tubing or rod being spun. The result here is equally as bad, because then we bore out all the tin and have a very hard mixture of tin and copper which will crack under heavy loads or pounding. The use of a cooling system applied at the proper time will create the desired crystal structure and also retard segregation in cylindrical bearings.

Excerpt from the paper: "Engine Bearing Processing and Its Effects," by R. A. Watson, Federal-Mogul Corp., presented at the Northern California Section Meeting of the Society, San Francisco, Calif., Jan. 11, 1938.

Automotive Bearing Materials and Their Application

By Arthur F. Underwood

Research Laboratories Division, General Motors Corp.

DEVELOPMENT of higher octane fuels, which will allow more power output through higher compression ratios with a given size engine, and the increasing use of Diesel engines for trucks and buses, will bring about a need for radically superior bearing materials, Mr. Underwood predicts in the early part of this paper.

Stating that a bearing material does not always require excellence in all characteristics, he lists the properties which make a good bearing alloy as: fatigue strength; ability to resist seizure to the steel shaft during periods when there is metal-to-metal contact; good bonding characteristics; mechanical compressive strength; conformability; embeddability; and corrosion resistance.

After describing tests for each characteristic, he treats the functional differences of the present available bearing materials, some of which are still in the final period of development.

Mr. Underwood shows that the numerous commercially available bearing materials have individual characteristics which must be considered in conjunction with the use to which each one is put. Bearings are like people, he says, in that they must be selected for the job they are to perform.

The author believes that future bearing materials will give greater fatigue life than the popular white metals and will have their anti-score characteristics. They will be non-corrosive and less susceptible to dirt and deflections, he adds.

Bearings, he predicts, will be designed for higher load factors as these materials are developed. This trend means narrower bearings, he says, and states that even now one design of connecting-rod has reached the limiting width necessary for the bolt.

IN the past, the demand for better bearing materials has been created by several factors of which the following are important examples: A greater endurance has been required to assure better reliability. Stiffening of the crankshaft to reduce vibration and deflection has been accomplished by narrowing the bearing which results in higher bearing pressures. Higher rotational speed, to utilize materials more efficiently, has intensified the problems enormously.

For the future, developments along the same lines will occur. In addition, there are at least two which will be accelerated and probably will bring about the need for radically superior bearing materials. Higher octane fuels will allow more power output through higher compression ratios with a given size engine. Besides increasing bearing loads, this development means a greater rate of pressure rise which aggravates bearing operation. The increasing use of Diesel engines for trucks and buses will further the need for improved alloys.

Bearing Properties – It is the object of this paper to treat the functional differences of the present available materials, of which several are still in the period of final development. Before considering these individual metals, it is well to establish the various "properties" which make a good bearing alloy but, as will be pointed out later, an alloy does not always require excellence in all characteristics.

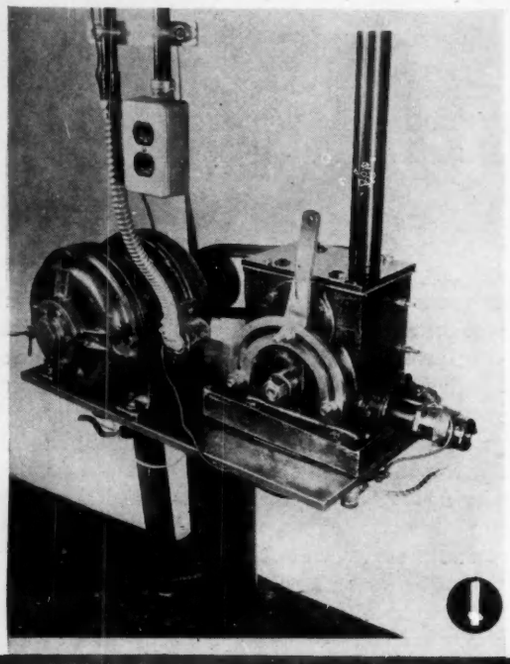
1. Fatigue strength usually is the most important property of a bearing material as it determines the allowable load and the life in the majority of cases. It is recognized generally that alternations of load will cause fatigue cracks which may surround and undermine areas. Loss of sufficient area so greatly reduces the load-carrying ability that failure occurs. An increase of load, an increase of temperature, and unfavorable design of the support are important factors in reducing fatigue life. It should be pointed out that bond fatigue is included.

Since most automotive bearings have been cast of babbitt which is particularly susceptible to fatigue, it is natural that several laboratory test machines have been developed. In a previous paper[1], one was described that produces an alternating load on a complete connecting-rod by the use of rotating inertia weights. Recently a new test machine has been developed. It operates on the principle of a very high-rate spring (upwards of 500,000 lb. per in.) to produce the load. A rotating shaft runs against the bearing to develop the typical oil film. This device, shown in Fig. 1, rates the

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 13, 1938.]

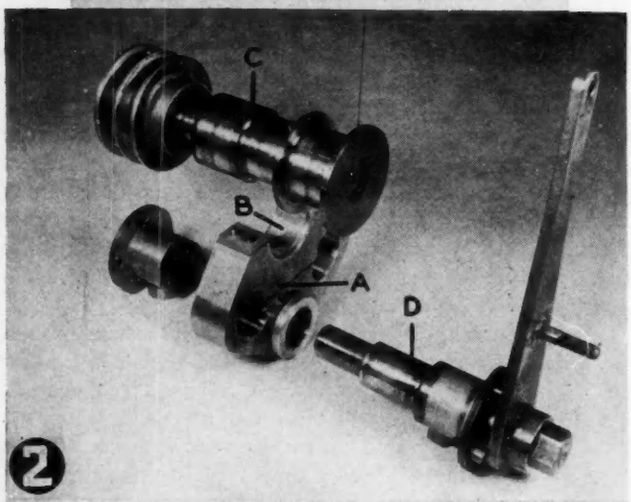
[1] See Symposium on Lubricants, Chicago Regional Meeting of the A.S.T.M., March 3, 1937, pp. 29-44: "Automotive Bearings – Effect of Design and Composition on Lubrication," by Arthur F. Underwood.

fatigue life of a specific bearing material without conditions imposed by rod design. An initial cost of a tenth of the original machine, and less attention and maintenance during test are to be noted. The view in Fig. 2 shows spring *A*, which deflects 0.020 in. under 10,000 lb. load, and which carries the test bearing *B*. Two bearings support the upper shaft which is revolved by an electric motor. At the center of this shaft is a journal *C* which is 0.018 in. eccentric and is 0.020 in. smaller in diameter than the bearing. At each revolution, the bearing is loaded by deflecting the spring unit. In order to start, the load must be removed by lowering the spring. The lower shaft is eccentric by 1/16 in. so that this can be done. After starting, the lower eccentric shaft is turned the proper amount to raise the spring against the upper rotating eccentric. An electric immersion heater regulates the oil temperature and a thermocouple is employed to register the bearing temperature. A babbitt bearing can be cracked up in 10 to 20 hr. of operation.



Although test machines have been developed to give quickly comparable results of various metals, an engine test is still used, and should be considered the final check. However, experience may be required to separate true fatigue failure from the results of other influences such as score properties or mechanical design.

It is nearly impossible to state the load under which a bearing material should operate because there are usually so many indeterminate factors. The temperature of operation is just as important as load and is dependent on several other items such as clearance, speed, grade and flow rate of oil, and mechanical design. As an example, Fig. 3 is given. These connecting-rod bearings are typical of those from two makes of engines. Both cars had been run 25,000 miles under a definite durability schedule. The load was exactly the same for both (860 lb. per sq. in.) and the *PV* was nearly the same. Yet the difference in operating life is apparent. We believe it is best to rate the fatigue life in a relative manner, such as a certain copper-lead ordinarily will give five times the life of tin-base babbitt.



2. Anti-friction or non-scoring property is the ability of the bearing material to resist seizure to the steel shaft during the periods when there is metal-to-metal contact. This property is most important when high surface velocity is combined with high load. We do not understand the mechanism of scoring, so the many tests which have been used are based on a set of conditions which are believed to represent operating conditions. In general, the tests may be divided into two groups: First, there are the scientific data which may be obtained by measuring the coefficient of friction through the use of inclined planes. These data are especially difficult to utilize. Second, more practical results are likely with a test which measures the score resistance by pressing two parts together while one or both are rotating. Speed, load, surface finish, lubrication, and so on, affect the result but, if the conditions are selected properly, pertinent data can be obtained. The variable results from the different test machines used during the earlier work on extreme-pressure lubricants are a good example of the effect of these factors.

Although it is necessary to rate bearing materials according to their non-scoring quality, there is no method for predicting the performance of, say, a certain copper-lead when used

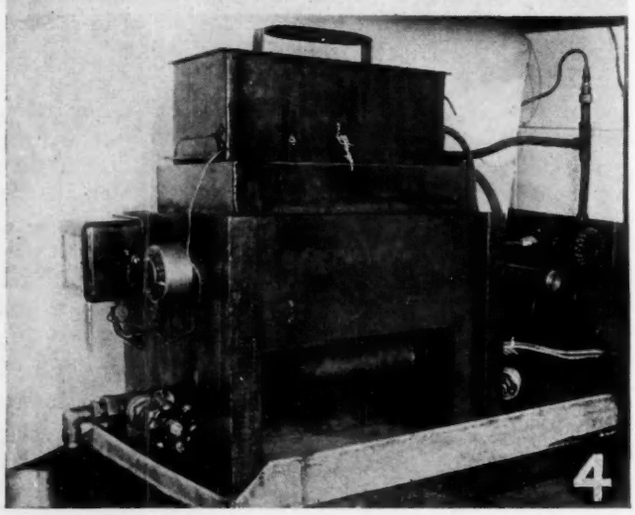


Fig. 1 – Bearing Fatigue Machine

Fig. 2 – Working Parts of Bearing Fatigue Machine

A – Spring bearing support
B – Bearing
C – Rotating eccentric shaft
D – Load-adjusting eccentric shaft

Fig. 4 – Bearing-Corrosion Test Machine

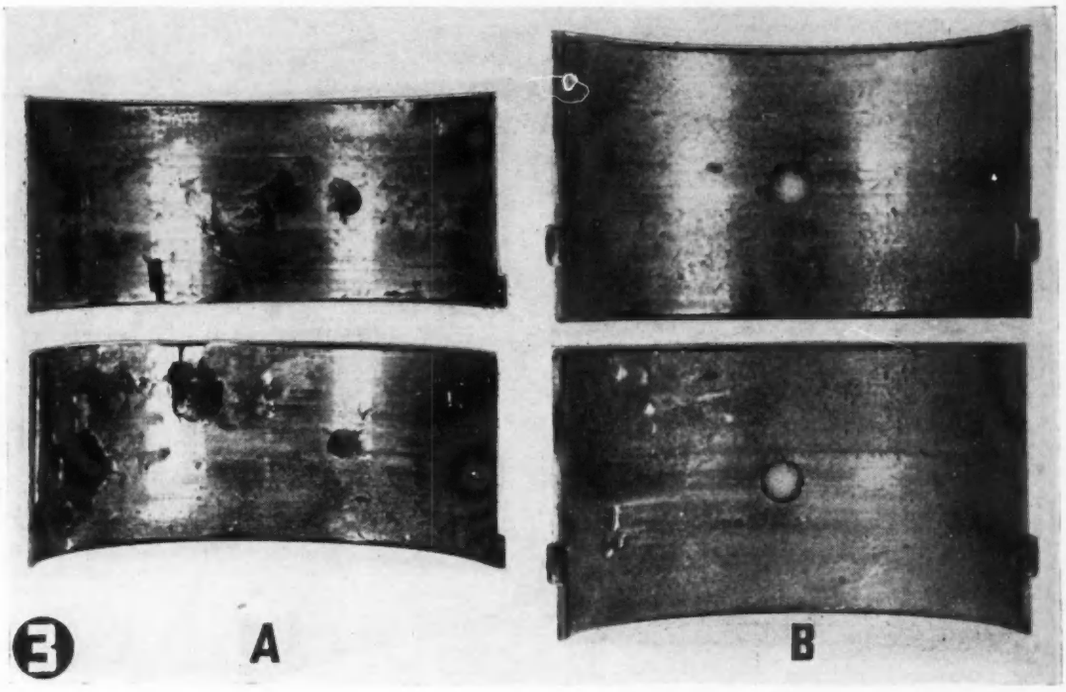


Fig. 3 – Effect of Operating Conditions on Bearing Life – Both Bearings Road Tested on Same Schedule for 25,000 Miles

Bearing	A	B
Material	Tin base	Tin base
Total load at 60 m.p.h.	1730 lb.	2160 lb.
Load at 60 m.p.h.	860 lb. per sq. in.	860 lb. per sq. in.
PV at 60 m.p.h.	23,600	24,500

in a connecting-rod bearing in a specific engine. This condition is because the deflections of the shaft and bearing support produce metal-to-metal rubbing of an indeterminate amount. One of the most active sources of deflections is the rate of pressure rise during the power cycle. Several years ago we were trying to make copper-lead bearings run in a certain line-eight engine, but at any speed of 3400 r.p.m. or above the connecting-rod bearings would always fail by scoring within 90 min. If the same test were carried out by turning the engine with a dynamometer, scoring would not occur. Several similar instances have shown that, as the rate of pressure rise is increased, the scoring quality of the alloy becomes increasingly important. It is therefore absolutely essential that an engine test be made under operating conditions.

3. In this paper, what is meant by bonding characteristics is the chemistry of adhesion, and not resistance to fatigue. We have found that, by heating to a temperature somewhat below the melting point, an excellent check can be had. Thus, if a steel-backed silver strip is heated to 1300 deg. fahr., badly bonded areas will blister. Similarly, if a lead-tin alloy is soldered to copper, it will continue to adhere after being subjected to 370 deg. fahr. But the addition of antimony to the lead-tin will destroy the bond on heating to the same temperature.

The adhesion of copper-lead is tested best by curling to the shape of a bearing from the flat. Because lead sweats out at relatively low temperatures, a heating test does not prove satisfactory. To the best of our knowledge, the shaping test is the only one which can be used on babbitt without destroying the bearing.

The time-honored methods of shearing off the metal, such as by a chisel, or making a tension test by pulling apart two pieces of metal which have been bonded together by the bonding alloy, are useful laboratory tests. But, of course, they are not applicable to continuous inspection of the product.

4. Mechanical compressive strength is naturally the resistance to continued extrusion by compression at the operating temperature. It is rare that failure occurs from this source because most alloys have hardening elements which raise the mechanical strength above the fatigue strength. When it does fail, there will be a lip of bearing material extruded into the chamfer space.

5. The conformability of a material is a measure of its readiness to adjust its dimensions to such influences as shaft deflection and misalignment. It is perhaps a combination of anti-score quality and mechanical strength as the bearing material must be worn away or extruded. And during this wear or displacement, there must be neither excessive heat generated nor seizure to the shaft.

Only in an engine under operating conditions can the suitability of a particular alloy with respect to this property be found.

6. When extraneous dirt is entering a bearing, embeddability is of great importance. Deep scratches which usually encircle the entire circumference of the bearing material are observed readily. The more easily the dirt (often steel chips) can be embedded into the bearing, the less will be the grooving of the steel crankpin. Most trouble from dirt is found in new engines through the lack of thorough cleaning of the parts before assembly. An outside circulating oil system used during run-in at the factory is very beneficial.

7. Corrosion resistance of bearing materials is the most recent of the properties to be given consideration. The problem obviously occurs only when there is a material subject to attack, and a corrosive lubricant. It is only within the past few years that corrodible materials have been used in the endeavor to produce an alloy having a higher fatigue life than babbitt. Cadmium and lead, and certain of their alloys, are subject to this deterioration.

The corrosiveness of the lubricant is brought about by the addition of an acid such as oleic in order to improve the oiliness of the oil, or by the oxidation of the lubricant itself to an acid during engine operation. Certain acids do not affect bearings. Thus an oxidized naphthenic oil may have an

acid number of 2 or 3 and not attack lead or cadmium, whereas a paraffin-base oil may produce corrosion at an acid number of 1, or even less. Crude oils contain a natural inhibitor which is removed by the more drastic types of refining, such as with solvent extraction. Oxidation, forming corrosive acids, can be controlled by the addition of the proper inhibitors to the lubricating oil. The use of inhibitors in the metal will be taken up later under the individual materials.

The apparatus shown in Fig. 4 has been quite useful in the general study of the corrosion resistance of bearing alloys and the problem of the oxidation of lubricants. Various modifications in the test method may be used to determine if an oil is corrosive in its original condition, if it will become corrosive as the result of heating at elevated temperatures, the effect of inhibitors, the effect of catalysts, and whether or not the bearing material is subject to corrosion.

The organic acids, normally present or developed by oxidation of the oil in the test, differ in their corrosive tendency toward bearing alloys. The per cent of acid or neutralization number is therefore not a quantitative measure of the bearing corrosion, consequently only the loss in weight of the bearing alloy should be used as an indication of the corrosive properties of lubricants or the corrosion-resistance characteristics of bearings in service.

The tendency of a lubricant to become corroded in use can be studied further by determining the rate of development of acidic compounds or the rate of bearing corrosion. The effect of inhibitors and catalysts on the oxidation of lubricants can be determined from the change in these rates.

In the test method used by our laboratory at the present time, the oil is impinged (Fig. 5) against a copper baffle 2 in. x 10 in., and one cadmium-silver bearing, both of which are abraded freshly to remove the burnished or cold-worked surfaces, for 5 hr. at a temperature of 325 deg. fahr. Our experience indicates that a lubricant that does not corrode the cadmium-silver bearing in 5 hr. in this test will not cause corrosion in service unless the driving conditions are unusually severe. The test may be continued for additional periods of 5 hr. each in order to differentiate between lubricants that do not corrode in the initial 5-hr. period.

This test has been shown to be so specific and sensitive that an engine test is not required to determine the corrosive tendencies of lubricants or the corrosion-resistance properties of bearing alloys.

Various Materials Discussed

Bearing Materials – With the general considerations of bearing properties covered, we now have a background on which to consider the individual materials. It has been said that bearing alloys are like people in that they must be selected for the job that they are to perform. An engineer may be able to set a fractured arm, but a doctor would be preferable.

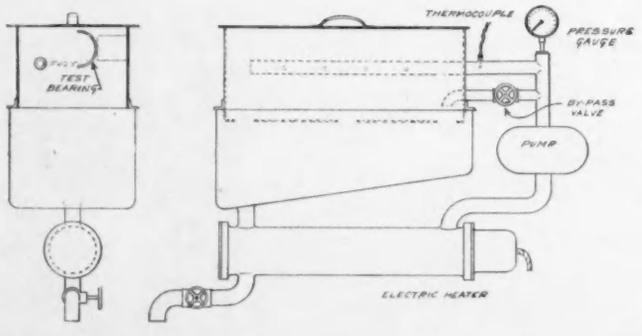


Fig. 5 – Diagram of Bearing-Corrosion Test Machine

Tin-Base Babbitt – There are many formulas for the composition of this material, and numerous papers and reviews are available that endeavor to show one or several to be superior to the others. To the best of our knowledge, there is very little worthwhile difference in fatigue life when all are applied properly. Probably more good has been done in recent years by decreasing the thickness of the babbitt, than by juggling percentages of tin, copper, antimony, and other elements said to offer longer life. Fig. 6 is a series of photographs showing the gradual failure of tin-base babbitt.

The foremost reason for babbitt to be so popular is its most excellent anti-score properties. It is the least sensitive of the alloys to oil-film failure and metal-to-metal contact from shaft distortion. This advantage, together with the fact that babbitt is bonded and cast quite readily, has made it the first choice for any bearing where the obtained fatigue life is sufficient. Similarly, it is excellent in its conformability characteristic, as it readily adjusts its dimensions to a runnable state. In past years it was quite common to burnish or burn-in babbitt bearings. Also considerable dirt can be embedded in the surface without undue scratching of the steel journal.

Another mark of superiority is its resistance to corrosion. It is not attacked by acids which ordinarily are added to improve the lubricating properties, or by acids formed from oxidation during running.

Briefly then, tin-base bearings are used whenever possible except in circumstances where a higher fatigue life or mechanical strength is needed.

Lead-Base Material

Lead-Base Babbitt – This class of alloy is similar to tin-base babbitt in all of its general characteristics. Fifteen years ago when lead-base babbitt was used about ⅛-in. thick, it was found to be inferior to tin-base alloy. Lead-base material is definitely softer and more plastic and, for a long time, was used in making camshaft bearings. Quite recently it has been tried again in a modern connecting-rod. When well supported and used very thin, its fatigue life is equal to, and sometimes greater than, tin-base babbitt. As with tin-base babbitt, there do not appear to be essential differences between the fatigue life of the two S.A.E. formulas for this alloy. One of our Divisions produced a little over 200,000 cars during 1937 having connecting-rods and main bearings lined with S.A.E. 14.

No fear of corrosion is to be entertained because research has shown that as little as 3 per cent tin will stop all attack. The usual S.A.E. compositions – S.A.E. 13 and S.A.E. 14 – have 5 per cent and 10 per cent tin, respectively.

A saving of about $25,000 per 100,000 cars can be made in material cost by changing from tin-base to lead-base alloy. Naturally, the actual saving depends on the market price of tin and lead. A further advantage is possible in time of war, as tin is imported whereas lead is mined in our country.

Briefly then, lead-base babbitt has the same general properties as tin-base babbitt and can be used to replace it if it can be handled properly from a design standpoint. Under some conditions there may be a saving in cost or an increase in life, or both.

Hardened Lead – This alloy differs from the lead babbitt in that it is hardened with 1 to 2 per cent tin, calcium, and small percentages of other alloys, instead of tin and antimony. Fatigue life of a bearing material is affected by design and operating conditions but, according to our comparative data (Fig. 7), the life of hardened-lead bearings is similar to that

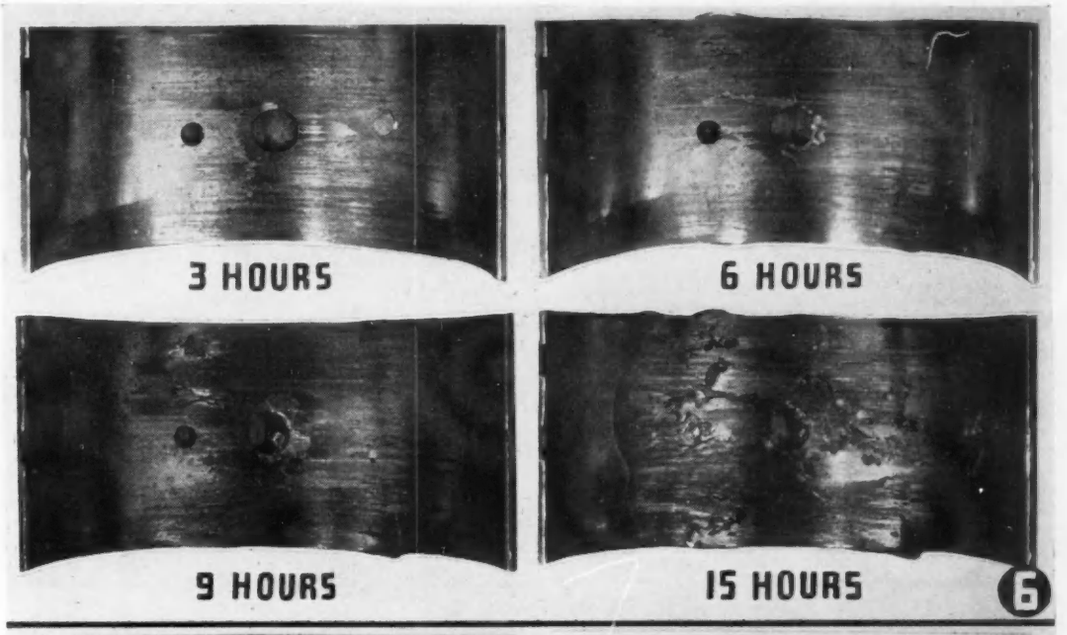


Fig. 6 – Progressive Fatigue of Tin-Base Babbitt Bearing During a 15-Hr. Test

Run at 4800 r.p.m., 5000-lb. total load, 350 deg. fahr. bearing temperature in the machine shown in Fig. 1.

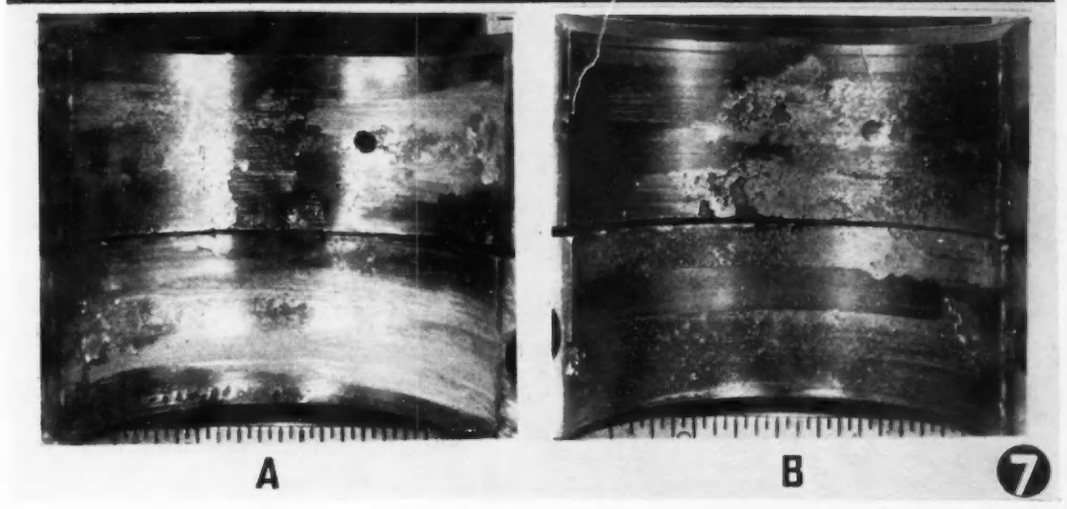


Fig. 7 – Fatigue Life of Babbitt and Hardened Lead

A – Tin-base babbitt after 40 hr. at 4500 r.p.m., 3400 lb. total load at 350 deg. fahr. bearing temperature

B – Hardened lead after 40 hr. with same conditions

of standard babbitt. In some instances, it has been reported to have somewhat longer endurance.

The one serious disadvantage is that it is subject to corrosive attack. The metal gradually dissolves in the oil so that the clearance opens up sufficiently to give a bearing knock or a lower oil pressure. A corroded bearing may have the original appearance or be slightly roughened. This fact limits its use to low oil-temperature operations or where inhibited oil can be assured.

With regard to the other bearing properties, this alloy has the same characteristics as tin-base and lead-base babbitts.

Briefly then, hardened lead can be used in place of tin-base or lead-base babbitts where there is no possibility of acid in the lubricant.

Cadmium – The principal improvement to be gained from using cadmium for a bearing material is that its alloys have a longer fatigue life than any of the three previously mentioned babbitts. In a paper by Smart[2], the subject of this new material was treated in full. Fig. 8 shows the relative fatigue life of a cadmium and a babbitt bearing. Cadmium has about three times the endurance in running time. It can be used in place of tin-base babbitt without changes in design or clearance.

Corrosive attack of cadmium is now well recognized. It is shown in Fig. 9. The bearing material is dissolved into the oil by acids, as in the case of lead. Under severe conditions, newly changed oil which is not properly resistant to oxidation, can become high enough in acid number after 500 miles to corrode away a very appreciable thickness of the alloy.

If the oil temperature is maintained below 250 deg. fahr., there is little likelihood of corrosion as the chemical activity rate is much retarded. One make of car has used this material successfully in a floating bushing for several years.

In a subsequent paper[3], Smart has demonstrated that the use of indium stops all attack. About 0.5 to 0.75 per cent of indium is plated on and diffused (at 340 deg. fahr.) into the finished bearing. Fig. 9 also has a treated bearing to provide contrast.

Thinner Lining Cuts Cost

The cost of cadmium and the inhibiting indium (when required) is considerably higher than tin. Up-to-date manufacturing processes have allowed the production of thin-backed shells with less than 0.010 in. of alloy. Only a few years ago the standard thickness was about 0.020 in., so the present thinner lining cuts the actual material cost.

Briefly then, cadmium alloys will give longer life when

[2] See Transactions of the American Society for Metals, Vol. 25, 1937, pp. 571-608: "Cadmium-Silver-Copper Alloys for Engine Bearings," by C. F. Smart.

[3] See Technical Publication No. 900, 1938, American Institute of Mining and Metallurgical Engineers: "Indium-Treated Bearing Materials," by C. F. Smart.

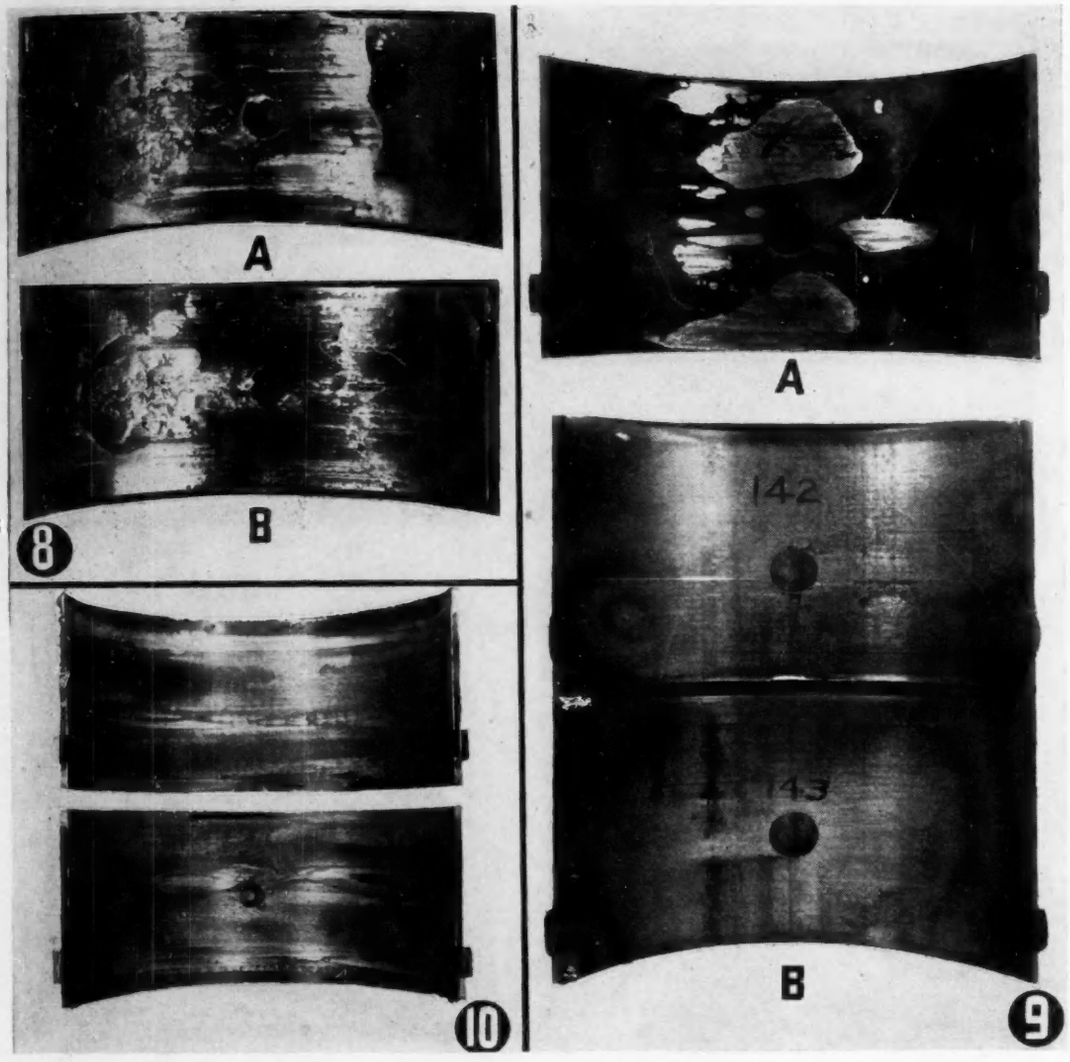


Fig. 8 (*upper left*) – Fatigue Life of Cadmium Alloy Against Tin-Base Babbitt

Both bearings run under the same test conditions; cadmium 23 hr., babbitt 7¾ hr.

A – Tin-base babbitt.
B – Cadmium - silver - copper

Fig. 9 – Effect of Indium on Corrosion of Cadmium-Alloy Bearings

A – Untreated cadmium-alloy bearing after 6¼ hr., 3700 lb., 4500 r.p.m., with 1 per cent oleic acid

B – Indium-treated cadmium - alloy bearing after 50 hr. under same conditions.

Fig. 10 (*lower left*) – Silver Bearings

Showing freedom from corrosion and fatigue after a total of 857 hr. with both corrosive and non-corrosive oil. Under the same conditions of test a copper-lead bearing has a maximum life of 100 hr. Crack is failure of steel backing

used in place of babbitt if there is no attack by corrosive lubricant. Indium can be alloyed with cadmium to stop corrosion with no loss in bearing properties.

Silver – Of the alloys which have been considered, all have followed the time-honored pattern set down for the requirement of a bearing material. It always has been reported that the alloy must consist of hard crystals in a soft matrix of, say, tin, lead, or cadmium. Hard crystals were to support individually their proportionate part of the load while the soft matrix supplied the conformability, embeddability, and other properties. Although the hard crystals did improve the mechanical strength and affect the endurance, this structure would allow no continued improvement over the base material.

It is our belief that this metallurgical idea is not wholly true. If a material can satisfy the necessary bearing properties required for the conditions under which it is to be used, then it is satisfactory. The solution may be a pure crystal, an alloy, or a mixture. Thus pure tin plating has been successful for a bearing material on pistons where the loading is low and the good non-scoring quality is needed. Recently we have found that absolutely pure silver is a possible bearing material. As usual, its properties determine the type of job it can do.

The fatigue strength of silver is very high when applied correctly. Fig. 10 shows a pure silver bearing after 857 hr. under operating conditions which no other bearing will survive for one-tenth the time. Failure occurred by cracking of the steel back and not of the silver lining. Continuing with its other good properties, this metal adheres strongly to steel although it is difficult to bond. Its bond can be checked by heating to 1300 deg. fahr. Corrosion will not occur.

Silver has a very erratic anti-friction characteristic which is shown up to worse advantage as the speed is increased. Quite often, the friction of one connecting-rod bearing will stop a machine or engine in very few seconds but scoring will not occur. A narrow streak of highly rubbed and polished silver will be observed on the bearing, but there is still no scoring in the sense that there is no welding to the steel journal. Scoring does occur under severe conditions. At the present time our data have shown no alloy which is an improvement. As with all materials having a moderate anti-score property, an increase in bearing clearance will give better lubrication conditions and decrease the probability of scoring.

Part of the sensitivity comes from the low conformability and embeddability of silver. Therefore it is a prerequisite that there be no undue shaft or support distortion. In fact, the bearing in Fig. 10 was used in a location which had severe bending and we found it advisable to bell-mouth. Clean oil especially free of the larger size particles is necessary to prevent grooving and scoring.

Cost of Silver Linings

Ordinarily it is remarked that silver is expensive, and it is by comparison with tin. The actual cost of the silver on an average-size connecting-rod bearing is 1 cent per each 0.001

in. thickness per half bearing (of approximately 4 sq. in. area).

Briefly then, silver is worth the increased cost where maximum fatigue life and compressive strength are essential but where operating conditions do not demand the anti-friction quality of babbitt.

Copper-Lead – Mixtures of these metals usually range from 30 to 45 per cent lead with the remainder copper. They are mixtures because lead will not alloy with copper above a fraction of 1 per cent. Small amounts of additional elements may be used by some manufacturers. These mixtures do not follow the orthodox metallurgical structure, because we have found that the best bearings have the copper in the continuous phase for a matrix to give high fatigue life. The lead is contained in the "pores" to give the anti-score quality.

In comparison with the babbitts, copper-lead should give perhaps four or five times the life under quite moderate conditions. Where very high loads must be sustained, especially at high temperatures where babbitt would crack in a short time, even better relative performance is obtainable. In Fig. 11 a direct comparison is made. The fatigue life can be improved by the addition of tin, but at a sacrifice in the anti-score properties.

All copper-lead compositions are well below babbitt with respect to anti-score quality although definite differences are made by manufacturing conditions and composition. If failure occurs because of lack of this quality, scoring can take place in a few seconds. It is believed that much of the wear attributed to copper-lead is incipient scoring. When tin is added to aid strength, the friction quality is impaired greatly. Tin-copper compounds are particularly undesirable for metal-to-metal contact at high rubbing velocities. In fact, we know of no alloying element to copper which definitely decreases scoring. An increase in clearance often will improve copper-lead operation in an engine. In these cases, it indicates a bearing which probably is on the "ragged edge."

Applications of Copper-Lead

These bearings are, therefore, found in locations where a substantial increase over the life of babbitt is demanded but where only moderate anti-friction properties are essential. Bus, truck, airplane, and Diesel engines are generally ideal. The high rubbing speeds and deflections encountered in some automobile engines are likely to induce scoring.

When properly cast and cooled, copper-lead is bonded firmly to the steel with copper only. Most bearings which

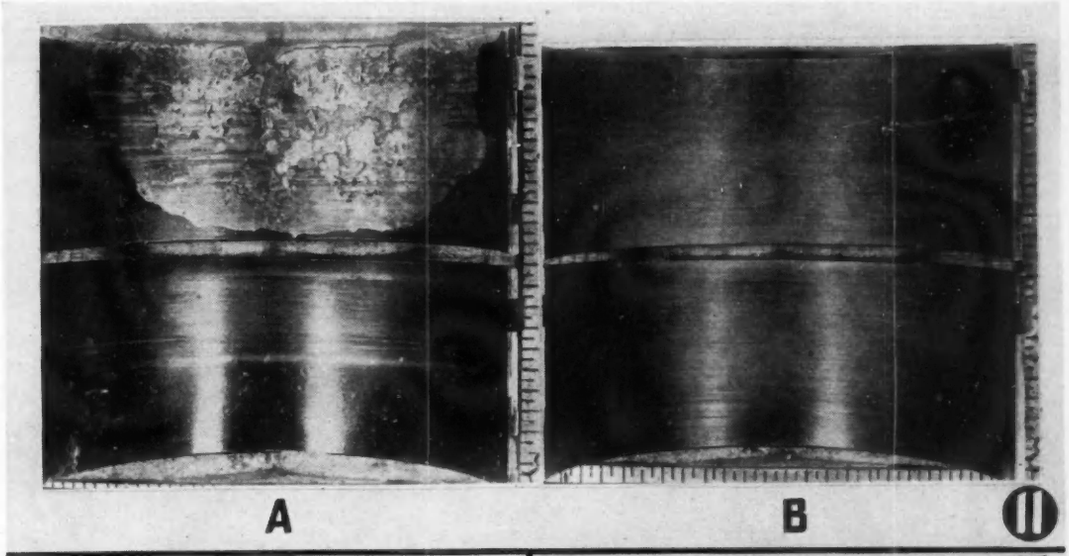


Fig. 11 – Fatigue Life of Copper-Lead Against Tin-Base Babbitt

Both bearings run under same conditions of 37 hr. at 4500 r.p.m., 4000 lb. total load, 350 deg. fahr. bearing temperature

A – Tin-base babbitt.
B – Copper-lead

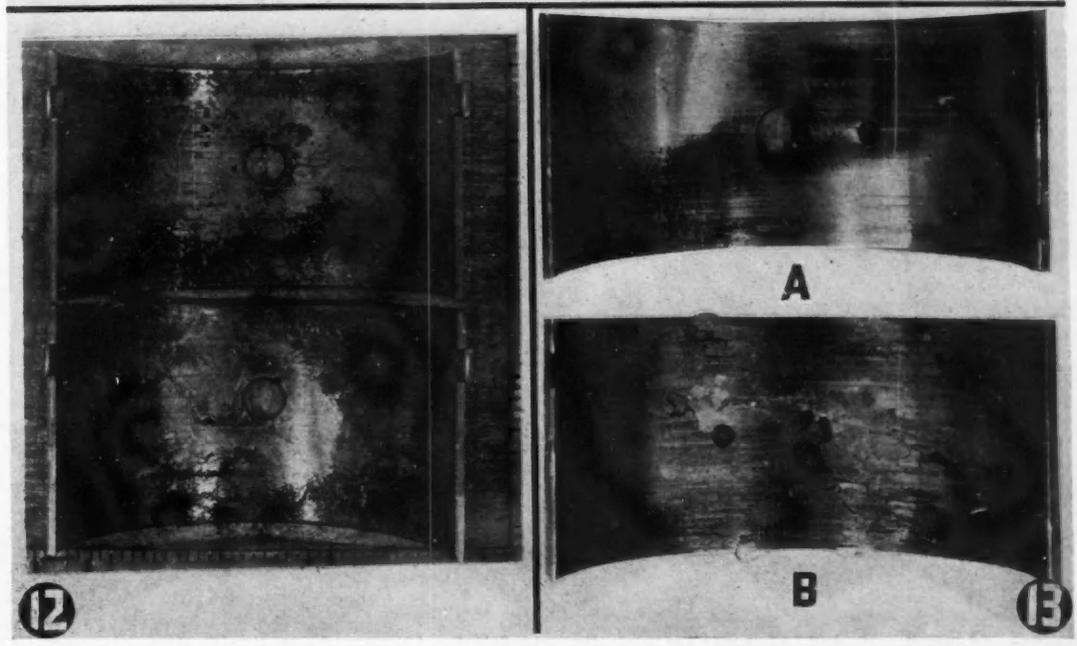


Fig. 12 – Copper-Lead Bearing – Corrosion Caused by Acidic Compounds Added to the Lubricant or Formed on Oxidation of the Lubricating Oil

Fig. 13 (*lower right*) – Effect of Deflection and Load on Fatigue Life

A – 55 hr. at 1680 lb. load and 0.020 in. deflection. No cracks after 55 hr. at 350 deg. fahr.

B – 9 hr. at 4800 lb. load and no deflection. Badly fatigued after 9 hr. at 350 deg. fahr.

have been run (especially if made by rolling from a flat strip) develop hair-line cracks over the entire surface, but these cracks do not readily undermine areas of metal by spreading parallel to the steel, as in the case of babbitt.

In common with most materials which are not highly resistant to scoring, it is not capable of conforming quickly to contortions of the shaft, or of embedding dirt. Where the load is mostly on one half of the bearing, such as in Diesel engines or moderate-speed bus and truck engines, it is possible to minimize the possible trouble. A babbitt lining is used for the low-load side, to embed the dirt and, to a certain extent, to improve conformability. There are definite instances in which the characteristics of such a composite bearing have been high enough to operate where scoring occurred on copper-lead only.

With the lead as essentially a pure element, it is subject to corrosion to the same extent as mentioned previously. When the lead dissolves in the lubricating oil, the remaining copper sponge compresses, thereby increasing bearing clearance. Also, with loss of the lead, the anti-friction quality is impaired. Fig. 12 is the result of running copper-lead in the presence of corrosive oil. Smart[3] has shown indium to be beneficial since it acts as a coating material to protect the underneath lead. Under severe operating conditions the indium may be worn off or penetrated by the acid oil so that corrosion occurs. Only by using non-corrosive oil can copper-lead be assured of giving the best results.

Briefly then, copper-lead as now employed commercially can be rated between babbitt and silver. It will give four or more times the fatigue life of babbitt but is liable to scoring because it is low in anti-score properties. In contrast with babbitt and silver, it is attacked by corrosive lubricating oil.

Conclusions – It has been shown that these numerous commercially available bearing materials have individual characteristics which must be considered in conjunction with the use to which each is to be put. We now realize that, although babbitts are lowest in fatigue strength, they are highly satisfactory in many cases where the life is long enough for the imposed conditions and the cost is low. Again, under corrosive conditions, any bearing containing corrodible lead should not be used. Another example might be silver, which is high in fatigue resistance but which requires careful design and consideration of cost to make it usable.

Properties of Ideal Material

What then might we idealize as a more perfect bearing material? One which could be utilized in practically all locations without regard to conditions imposed on the metal. It should have the fatigue resistance of copper-lead (or perhaps even silver), the anti-friction of, say, tin-base babbitt, the bond and mechanical strength of silver, the conformability, embeddability and corrosion resistance of a babbitt, and the low material cost of S.A.E. 14 which means copper and lead as against tin, cadmium, silver, or indium. Experimentally such bearings have been produced and tested at 4200 r.p.m. with 280 deg. fahr. oil in an engine which requires excellent anti-friction quality and which gives an expected babbitt life of 150 hr. After 150 hr., the experimental bearing was in perfect condition. Even at 375 hr. it remained in a remarkably fair, yet runnable, condition after the equivalent of 30,000 miles at 80 m.p.h.

Such bearings will be of great help in designing more compact and longer lived engines for the future. It has been the main purpose, here, to point out what characteristics a bearing material required and how the commercially available metals fulfilled them.

Bearing Design Factors – In a previous paper[1], lubrication factors were considered, so it is proposed that only the more mechanical items be dealt with. It must be recognized that, with such a complexity of factors in the bearing problem, each must be segregated by itself as much as possible.

Stiffness of the bearing support generally is presumed to be made as great as possible for best results. It generally is reasoned that bearings are cracked by deflections of the support. In our experiments, it has been shown that deflections are not the prime source of fatigue. We have operated the machine shown in Fig. 1 with a deflection of 0.020 in. and a moderate load of 1600 lb. at 350 deg. fahr. After 55 hr. the bearing (Fig. 13) remained as uncracked as at the start. Also in the same photograph is a bearing which has been operated at three times the load (4800 lb.) but no deflection. In 9 hr. failure has progressed to a critical point. It has been pointed out[1] that the part of the bearing under the rod column of a high-speed car engine is subject to earlier fatigue than the cap portion which sustains the heavier load. Deflection of the cap spreads the load over a larger area. Concentration of load on a smaller area overloads the bearing material. It is when deflections intensify this action that cracking occurs. If the deflections are uniform, longer life may be obtained.

Grooves in bearings now are recognized generally to interfere seriously with load-carrying ability. The hydraulic oil film which must support the load is ruptured completely. Of course, grooves in main bearings are required to carry the lubricant to the crankpin, and grooves in connecting-rod bearings are required to carry oil to the bearing surfaces in "oil-inertia" lubricating systems.

Causes of Wear

Wear is an important item in bearings which are designed for long life. It is caused by two factors. Dirt will groove the lining and the steel shaft. It also becomes embedded in the lining and laps away the shaft. Hardening the steel reduces the wear. The other factor is insufficient anti-score properties of the bearing that often will cause incipient scoring which shows up as wear. Hardened shafts are said to decrease the susceptibility to scoring. We are doubtful that it is the hardness as expressed by Brinell. Rather, it seems that a smoother and more "run-in" finish can be obtained on the harder steel.

Thickness of the bearing metal, especially babbitts and cadmium alloys, should be at the manufacturing minimum to give the longest fatigue life. The trend of the industry for a long time past has also been to reduce the thickness of the steel backing. It generally is held to a minimum consistent with ease in maintaining accuracy of shape, thus reducing weight and cost.

Bell-mouthing is a sign of shaft distortion. Main bearings are fatigued particularly by this situation. A higher percentage of counterweighting will be found beneficial. In some instances where bell-mouthing is uniform in degree and direction, we have pre-machined the lining to conform to the conical shape of a "run-in" bearing.

Trend in Bearings – Although it is difficult, if not foolhardy, to write for posterity, we believe that trends are evident. Future materials will give greater fatigue life than the popular white metals and will have their anti-score characteristics. They will be non-corrosive and less susceptible to dirt and deflections.

Bearings will be designed for higher load factors as these materials are developed. This trend means narrower bearings. Even now, one design of connecting-rod has reached the limiting width necessary for the bolt.

[1] See Symposium on Lubricants, Chicago Regional Meeting of the A.S.T.M., March 3, 1937, pp. 29-44: "Automotive Bearings – Effect of Design and Composition on Lubrication," by Arthur F. Underwood.

Safe Viscosity for a Motor-Car Engine Lubricant

By Stanwood W. Sparrow
The Studebaker Corp.

MR. SPARROW'S paper emphasizes the importance of the question: "What is the minimum safe viscosity for an engine oil?"

Although he does not attempt to solve the problem, he presents material "accumulated as a by-product of routine engine tests and development," which, he says, indicates that rather low viscosities may be safe for bearings if and when we can be sure that the amount of lubricant which reaches the bearings will be adequate. He adds that it also indicates the extent to which safe lubrication of the cylinder bores depends upon the ability to produce and maintain smooth surfaces on pistons, piston-rings, and cylinder walls.

He illustrates how a low viscosity is effective in increasing cranking speed and in reducing friction – thereby producing a gain in horsepower and fuel economy. He also cites examples to show the extent to which low viscosity is detrimental as regards oil consumption, blowby, and the protection which the oil film affords to the rubbing surfaces.

Finally, Mr. Sparrow discusses in some detail the effect of changes in viscosity upon the lubrication of pistons, piston-rings, cylinder walls, and main and connecting-rod bearings.

WE are told that "In the bright lexicon of youth, there is no such word as 'fail'," and we occasionally are told that in the dingy dictionary of engine lubrication we should mark the word "viscosity" as obsolete in so far as it applies to the protection afforded by an oil film. Nevertheless, such phrases as "scored pistons," "scuffed piston-rings," and "burned bearings" are still current in automotive circles. And, by a strange coincidence, the engineer's interest in the wear of cylinder walls seems to have increased since the driving public has become conscious of the merits of low viscosity in a motor oil. Hence, it is not unseemly to ask the question: "What is the minimum safe viscosity for an engine oil?" And, in fairness to the reader, it must be admitted that the comments which follow tend to show the importance of the question rather than to answer it.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 17, 1938.]

Since the selection of a lubricant is usually a compromise between desirable and undesirable characteristics, it is in order to review briefly the blessings of low viscosity before discussing the dangers.

Cranking Speed – Fig. 1 is a reminder that, the lower the viscosity, the higher will be the cranking speed. No one needs to be reminded that a high cranking speed is a prerequisite of easy starting.

Flow at Low Temperatures – The ability to reach its destination quickly is as important to a lubricant as to a race horse and the extent to which this ability is affected by viscosity is illustrated by Fig. 2. This comparison, however, is hardly fair to the more viscous lubricant since the delivery from a gear type of pump depends primarily upon speed and the effect of a higher viscosity should be to increase the pressure rather than to decrease the flow. A more significant comparison is offered in Fig. 3 which shows the time required for excess oil to appear at the top of the pistons when the engine is rotating at 700 r.p.m. In nearly all cylinders this time was nearly twice as long for the No. 50 oil as for the No. 10 oil. Before leaving this figure it should be pointed out that the relatively long time which elapses before excess oil appears is influenced by the effectiveness of the oil-control rings and it should not be inferred that the cylinder walls receive no lubrication during this period. With the piston-rings removed, a test run with No. 10 oil showed oil on all cylinder walls within 10 sec. from the time of starting.

Friction and Power – A major element of engine friction is the work of shearing oil films, and a decrease in viscosity means a decrease in this work. One example of such a decrease and the accompanying increase in brake torque is furnished by Fig. 4.

Friction and Fuel Economy – The effect of the change in friction, shown in Fig. 4, upon the specific fuel consumption at full load appears in Fig. 5. Of greater interest is the fuel economy at part load. Here the effect of viscosity becomes

CRANKING SPEED AT 0 DEG. FAHR MODEL "6-226" ENGINE
PRODUCTION BATTERY CHECKED AT 1300 GRAVITY
AND RECHARGED BETWEEN EACH RUN

Oil Number	Viscosity, Saybolt Universal Seconds			Cranking Speed, r.p.m
	at 210 Deg. Fahr	at 130 Deg. Fahr	at 0 Deg. Fahr.	
S.A.E. 10	42.8	77.5	3,500	59.7
S.A.E. 20W	52.5	144	21,000	31.3
S.A.E. 20	50.5	150	48,000	16.8
S.A.E. 30	63.0	264	220,000	10.5

Fig. 1 – A Low Viscosity Helps Cold-Weather Starting by Permitting a High Cranking Speed

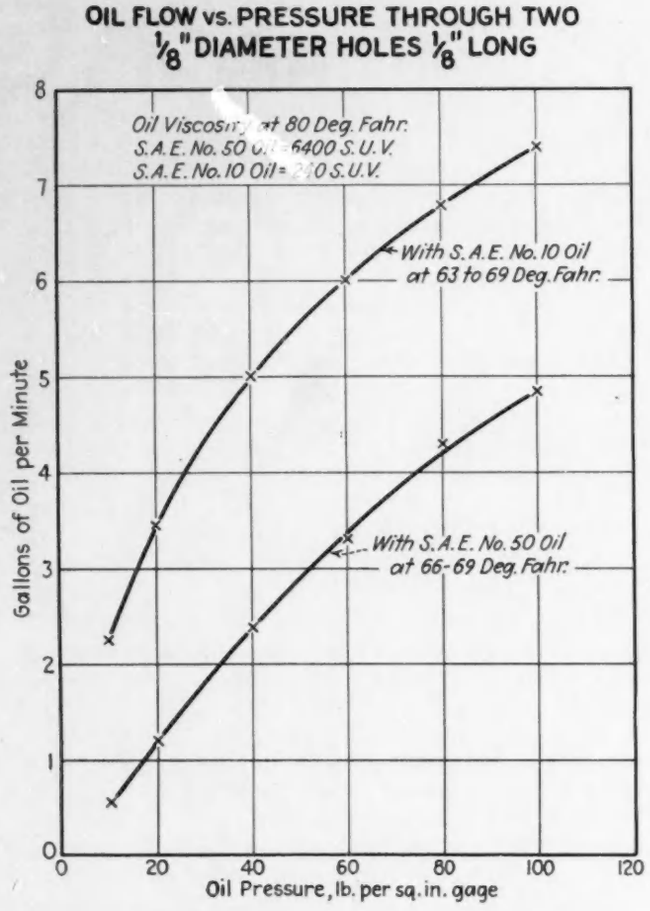


Fig. 2 – A Reduction in Viscosity Increases the Rate of Flow at Constant Pressure

marked since, at low speeds, as much power may be expended in overcoming engine friction as in propelling the car. The lower curve in Fig. 6 gives the power required to drive the car at various speeds. Directly above are curves of indicated horsepower, namely, the power to drive the car plus the power to drive the engine. Curves of gasoline mileage at the top of the figure were calculated upon the assumption that thermal efficiencies were not affected by the change in oil and hence that, at a given speed, the miles per gallon would vary inversely with indicated horsepower.

Oils differing widely in viscosity have been chosen for the foregoing comparisons in order to magnify the gains which

TIME REQUIRED FOR EXCESS OIL TO APPEAR AT TOP OF PISTONS WITH

Cylinder No.	S.A.E. 50 Oil	S.A.E. 10 Oil
1	10 Minutes 27 Seconds	3 Minutes 9 Seconds
2	29 " 5 "	9 " 0 "
3	more than 30 " — "	12 " 14 "
4	10 " 10 "	5 " 43 "
5	3 " 25 "	2 " 55 "
6	17 " 20 "	9 " 0 "

Engine Motored at 700 R.P.M. with Cylinder Head Removed.
All Bores Wiped Clean and Dry Prior To Test.

Oil Viscosity at 80 Deg. Fahr
S.A.E. No. 50 Oil = 6400 S.U.V.
S.A.E. No 10 Oil = 240 S.U.V.

Fig. 3 – Low Viscosity Reduces the Delay in Getting Oil to the Cylinders in Cold-Weather Starting

may be expected from a reduction in viscosity. It is now in order to consider what may be lost by such a change.

Oil Consumption – To one who can read and who retains some degree of confidence in the honesty of advertising, it would appear that the proper selection of piston-rings, combined with reasonable care in the manufacture of the engine, should provide everything desired in the matter of oil control. Unfortunately the public's mania for oil economy has now reached the stage where the average car owner is less perturbed by a grease spot on his vest than by one on the cylinder wall. Hence it is not possible to ignore the fact that an increase in viscosity tends to decrease oil consumption. To cite a single example: in one engine an oil consumption of 794 miles per gal. was obtained at 50 m.p.h. with S.A.E. 10 oil in contrast to 1675 miles per gal. with S.A.E. 50 oil.

Blowby – It would be silly to suggest that the choice of viscosity should be dictated by its effect on blowby. Nevertheless, it is but fair to record that an increase in viscosity frequently does reduce blowby. In one engine the substitution of No.

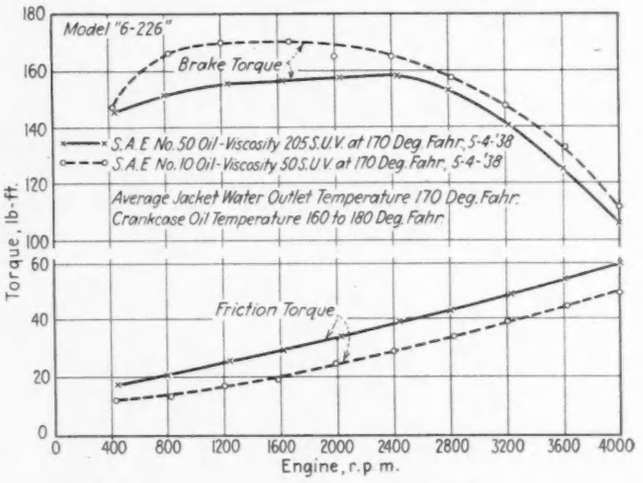


Fig. 4 – High Viscosity Causes High Friction and Low Power

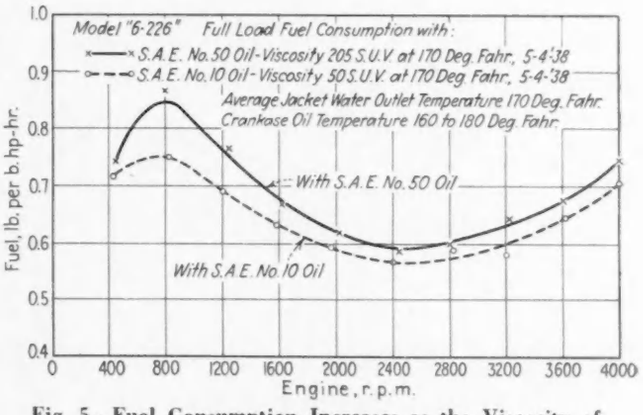


Fig. 5 – Fuel Consumption Increases as the Viscosity of the Lubricant Is Increased

40 oil for No. 20 oil reduced the blowby from 1.7 cu. ft. per min. to 0.5 cu. ft. per min. Measurements of blowby in another engine follow:

Speed	Oil	Blowby
4500 r.p.m.	S.A.E. 40	3.2 cu. ft. per min.
4500 r.p.m.	S.A.E. 30	3.9 cu. ft. per min.
4500 r.p.m.	S.A.E. 20	4.4 cu. ft. per min.
4500 r.p.m.	S.A.E. 10	6.6 cu. ft. per min.

What happens when a ring combination is very ineffective is shown in Fig. 7, where it will be seen that a change from No.

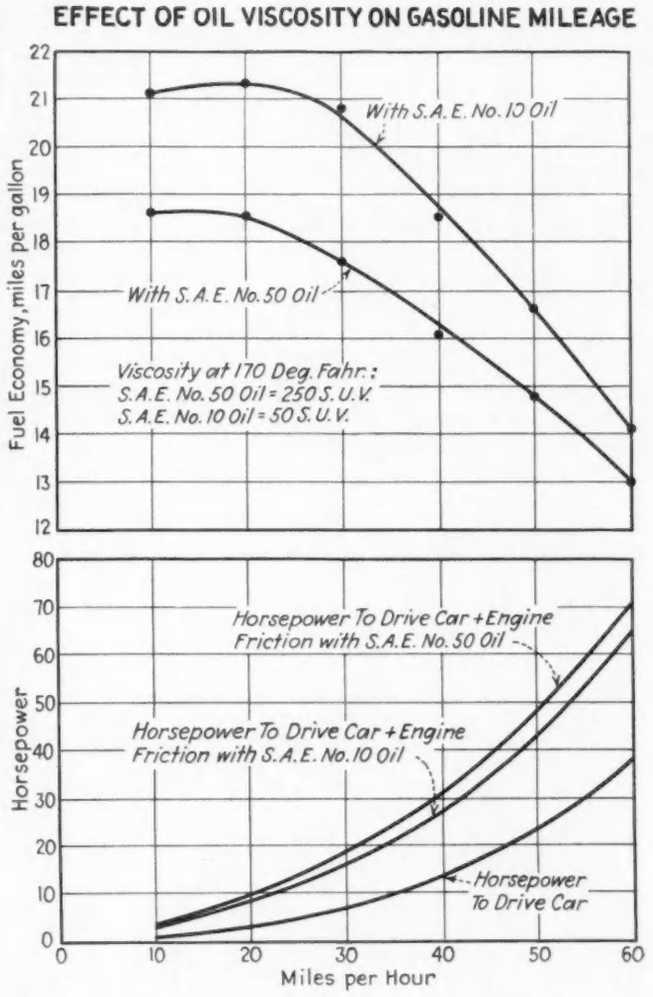


Fig. 6 – At Low Speeds It May Require as Much Power to Overcome the Friction of the Engine as to Propel the Car – A Change in Viscosity which Reduces Engine Friction Will Increase Gasoline Mileage

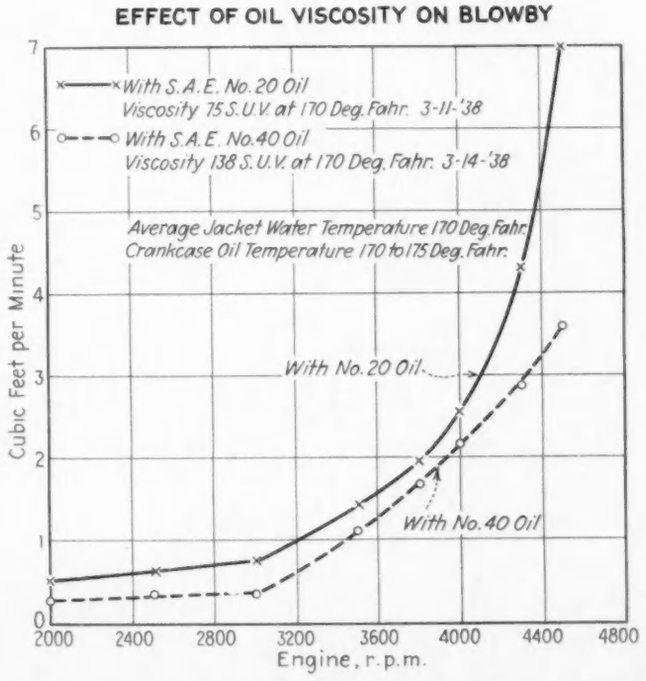


Fig. 7 – An Oil of Too Low Viscosity May Make It More Difficult to Control Blowby

40 to No. 20 oil was responsible for an increase of more than 3 cu. ft. per min. in the blowby at 4500 r.p.m.

Protection of Rubbing Surfaces – Perhaps the comments made thus far will serve as a smoke screen to conceal the dearth of data on the subject with which this paper is supposed to deal – the relation of the viscosity of an oil to the life of engine parts. Of such parts, those most likely to be affected are the main bearings, connecting-rod bearings, piston-rings, pistons, and cylinder walls.

Main Bearings – Prior to the preparation of this paper the author was convinced that he could offer convincing evidence that, in one engine at least, it was possible to prolong the life of the main bearings by the use of No. 50 oil instead of No. 20 oil. It was known that, when this engine was operated at 4500 r.p.m. with No. 20 oil, connecting-rod bearings would fail ordinarily in 15 hr. if the crankcase temperature were allowed to exceed 270 deg. fahr. There was no difficulty, however, in obtaining 50 hr. if: 1. No. 50 oil were used, or if 2. main bearings were replaced before the drop in oil pressure became excessive. Likewise the rod bearings would be in good condition after 50 hr. of operation with No. 20 oil provided the crankcase temperature did not exceed 200 deg. fahr.

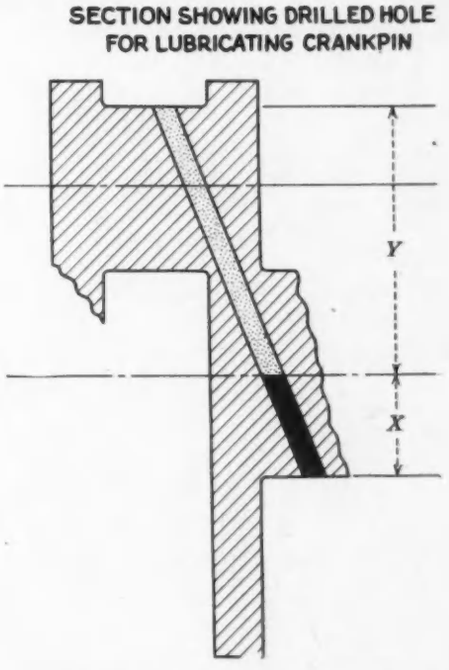


Fig. 8 – At High Speeds, the Flow of Oil to the Connecting-Rod Bearings Is Governed Very Largely by the Centrifugal Force Developed by the Column of Oil in the Crankshaft

The sequence of events is quite apparent; first, wear or partial destruction of the main bearings; second, a drop in oil pressure; and finally, failure of the rod bearings because of insufficient oil. Hence it appeared logical to conclude that the No. 50 oil had prolonged the life of the rod bearings by decreasing the wear of the main bearings, thereby preventing a large drop in the oil pressure and in the flow to the rods. After examining the records of these and subsequent tests, however, the validity of the above conclusion seems open to question. As far as the main bearings themselves are concerned, the records fail to show that the more viscous oil made any significant improvement in life. Hence it is possible that the benefit derived from the No. 50 oil came from the fact that, for a given amount of main-bearing wear, it gave a greater flow to the rods than did the No. 20 oil. The reason for the greater flow will be discussed in the section devoted to connecting-rod

Connecting-Rod Bearings After Running at 4000 R.P.M., Full Load for	1 Hr.,	Oil Viscosity at 130 Deg. Fahr.	76 S.U.V.
	1 Hr.,		46 S.U.V.
	1 Hr.,		40 S.U.V.
	1 Hr.,		35.5 S.U.V.
	1 Hr.,		33.3 S.U.V.
	4 Min.,		33.0 S.U.V.

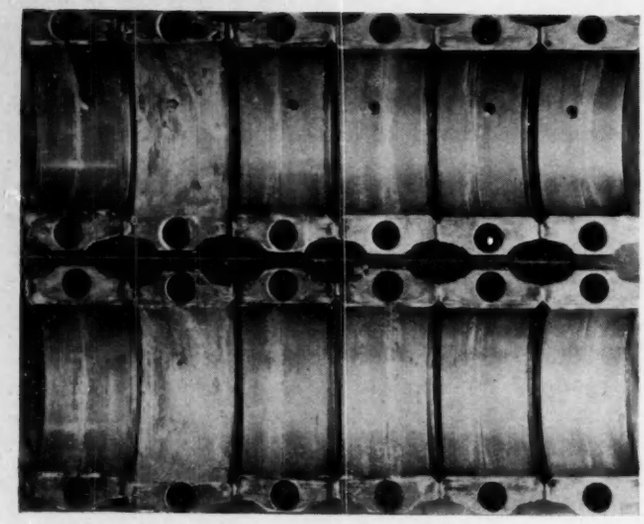


Fig. 9 – The Failure of This Connecting-Rod Bearing Probably Was Due to Insufficient Oil as the Other Bearings Show No Evidence of Breaking Down of the Oil Film Despite the Low Viscosity of the Lubricant

bearings. From the data at hand, therefore, there is not sufficient justification to warrant the statement that a No. 20 oil does not provide a safe viscosity for main bearings, even with a crankcase oil temperature between 275 and 300 deg. fahr.

In another design of engine, main and rod bearings have been found in good condition after 50 hr. of full-load operation at 4300 r.p.m. with No. 10 oil and a crankcase temperature of 200 deg. fahr. As this speed and temperature are rarely exceeded by this engine, the No. 10 oil can be considered

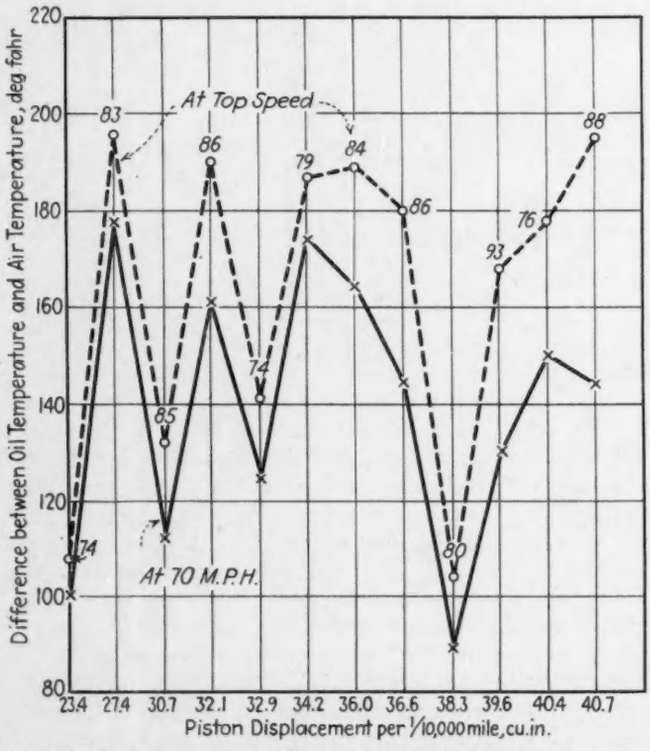


Fig. 10 – Temperatures of the Crankcase Oil for Twelve 1938 Cars

to have a safe viscosity for main and connecting-rod bearings as long as the bearing clearances are not excessive.

Connecting-rod Bearings – In the preceding section, an example was given of one engine in which the life of the connecting-rod bearings could be extended from 15 to 50 hr. by the use of No. 50 oil instead of No. 20 oil. That this increase was not due to an increase in the film strength of the oil was proved by the fact that equally good results could be obtained by replacing the main bearings before the drop in oil pressure became excessive. Obviously, the rod bearings failed because they did not receive enough oil, and it is pertinent, therefore, to inquire why a decrease in viscosity should decrease the flow to a connecting-rod bearing. In its essentials, the conventional lubrication system consists of a gear pump which delivers the oil to a channel through which it is conducted to various outlets. If the pump delivers oil at a constant rate then one might expect that a decrease in viscosity would decrease the pressure necessary to maintain the same rate of flow rather than that it would effect any major change in the rate of flow. Actually, the viscosity does affect the capacity of the pump. Under the conditions stated in Fig. 2, the flow, at a pump speed of 1000 r.p.m., was 5.3 gal. per min. with No. 50 oil and 4.7 gal. per min. with No. 10 oil. It is probable, however, that this change is of minor importance and that the key to the problem may

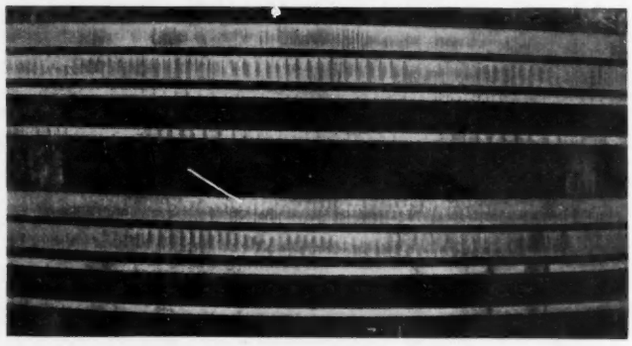


Fig. 11 – A Typical Example of the Scuffing Which Occurs at High Speeds and Loads

be found in a consideration of the factors governing the flow from the main bearing to the connecting-rod bearing. A glance at Fig. 8 calls to mind the fact that, at high speeds, the flow of oil to the connecting-rod bearing is governed very largely by the centrifugal force developed by the column of oil in the crankshaft. The significance of this point will be appreciated when it is realized that the difference between the weight of a column of No. 10 oil and a column of No. 50 oil is less than 2 per cent. To force oil into the crank, assuming the oil hole to be empty, it is necessary to build up sufficient pressure to overcome the centrifugal force produced by a column of oil extending from the centerline of the main journal to its surface, as indicated by the solid black area in Fig. 8. A reduction in viscosity reduces the pressure at the main bearing and hence decreases the force tending to produce flow into the main journal, but it does not reduce materially the centrifugal force resisting the flow. By the same line of reasoning, if the capacity of the pump is sufficient to keep the hole in the crankshaft full of oil, then the effect of a reduction in viscosity will be a marked increase in the rate at which oil is discharged from the crankpin. It is outside the province of this paper to consider all the intermediate stages between the full oil hole and the empty one. It will suffice to say that the overall effect of a reduction in viscosity is to make it harder for the oil to get into the crankshaft, and easier for it to get out.

Fig. 9 furnishes another example of a rod bearing failure which probably was due to insufficient oil since the other

1 Hr. at 4000 R.P.M., Full Load
With 10-W Oil,
Test No. 2, Sept. 29, 1936.

1 Hr. at 4000 R.P.M., Full Load
With S.A.E. 40 Oil,
Test No. 3, Sept. 30, 1936.

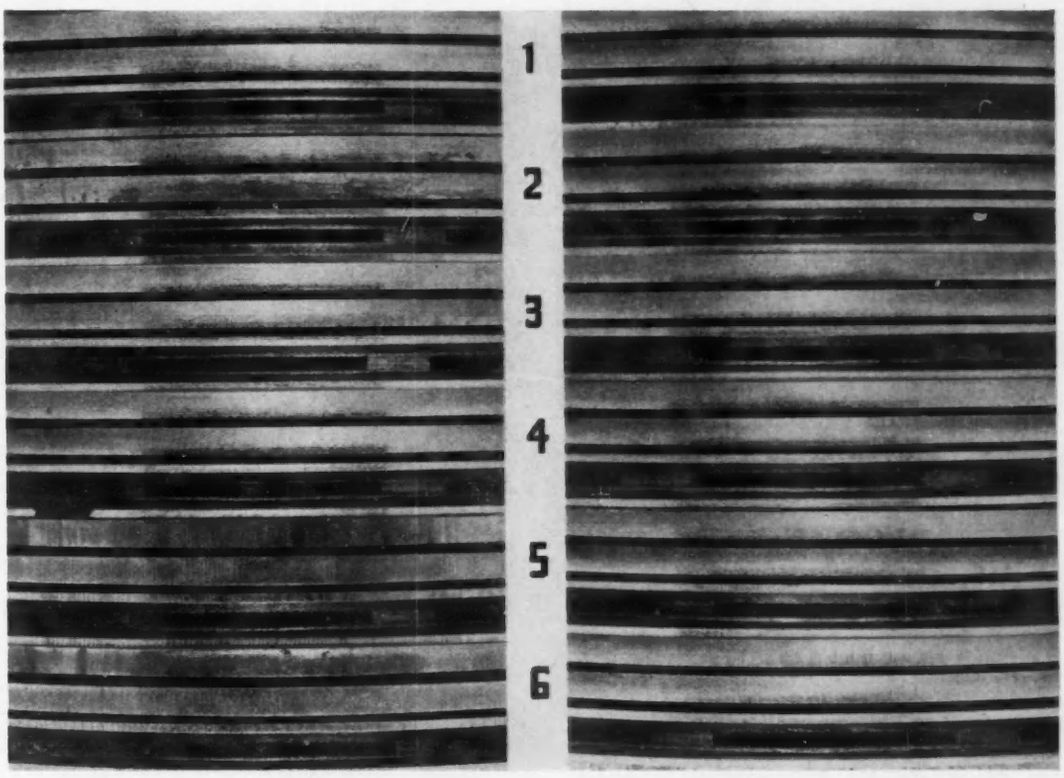


Fig. 12 – One Case Where Viscosity of a 10-W Oil Was Not Safe

Viscosity of 10-W Oil at 170 deg. fahr. = 50 S.U.V.
Viscosity of 40 Oil at 170 deg. fahr. = 138 S.U.V.

1 Hr. at 4000 R.P.M., Full Load
With S.A.E. 30 Oil,
Test No. 4, Oct. 1, 1936.

1 Hr. at 4000 R.P.M., Full Load
With 10-W Oil,
Test No. 5, Oct. 1, 1936.

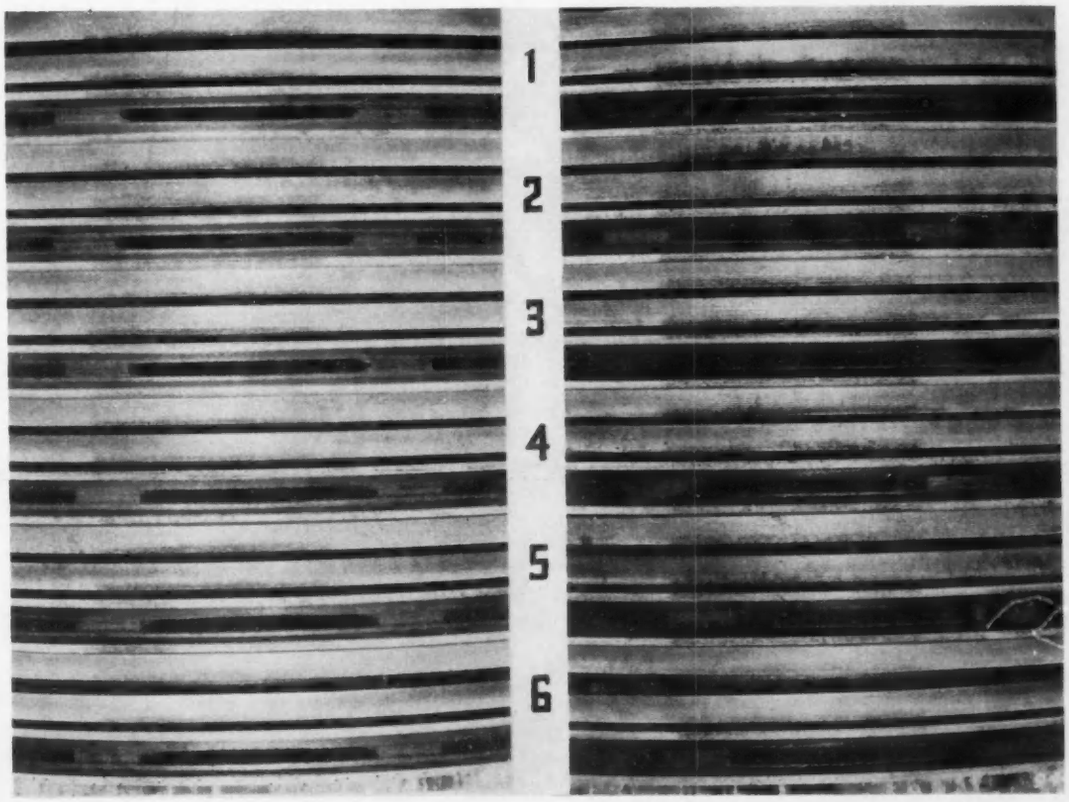


Fig. 13 – Under the Conditions of This Test the 10-W Oil Provided Safe Viscosity

Viscosity of 10-W Oil at 170 deg. fahr. = 50 S.U.V.
Viscosity of 30 Oil at 170 deg. fahr. = 108 S.U.V.

1 Hr. at 4000 R.P.M., Full Load
With 20 Per Cent Kerosene +
10-W Untreated Oil,
Test No. 10, Oct. 7, 1936.

1 Hr. at 4000 R.P.M., Full Load
With 20 Per Cent Kerosene +
10-W Treated Oil,
Test No. 14, Oct. 9, 1936.

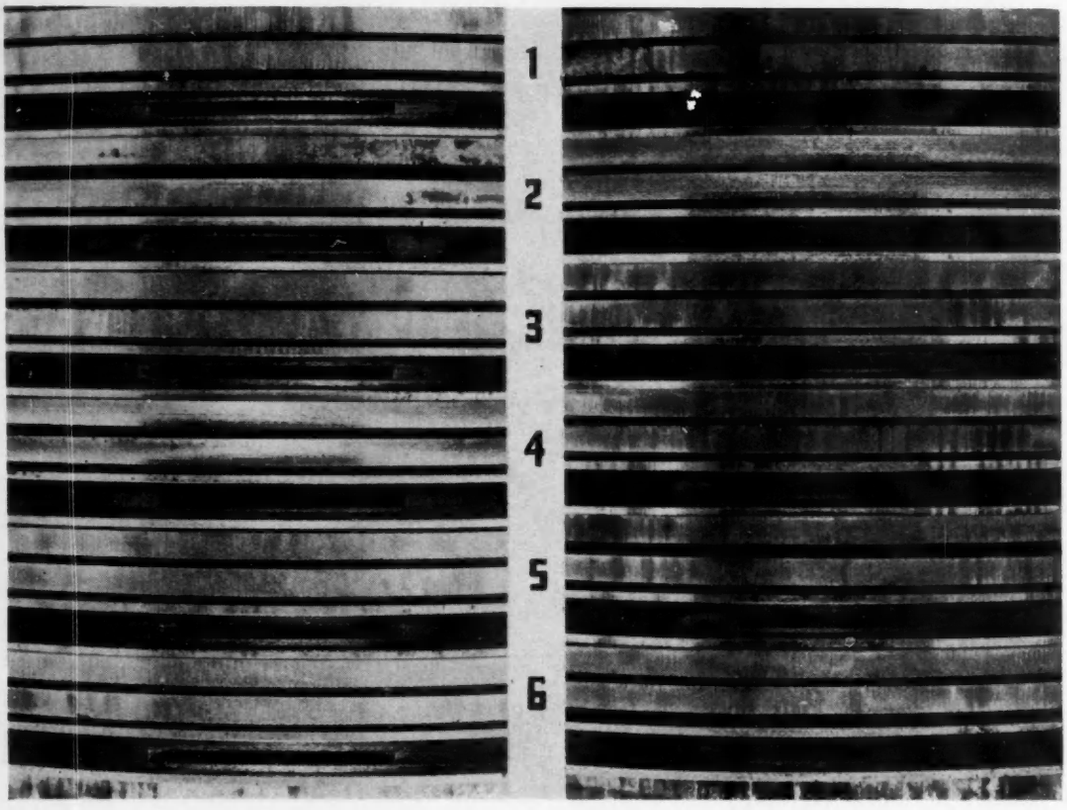


Fig. 14 – The Viscosity of This Lubricant Was Definitely Unsafe

Viscosity of this lubricant was 40 S.U.V. at 130 deg. fahr.

1 Hr. at 4000 R.P.M., Full Load
With No. 30 Oil,
Test No. 11, Oct. 8, 1936.

1 Hr. at 4000 R.P.M., Full Load
With No. 30 Oil,
Test No. 12, Oct. 8, 1936.

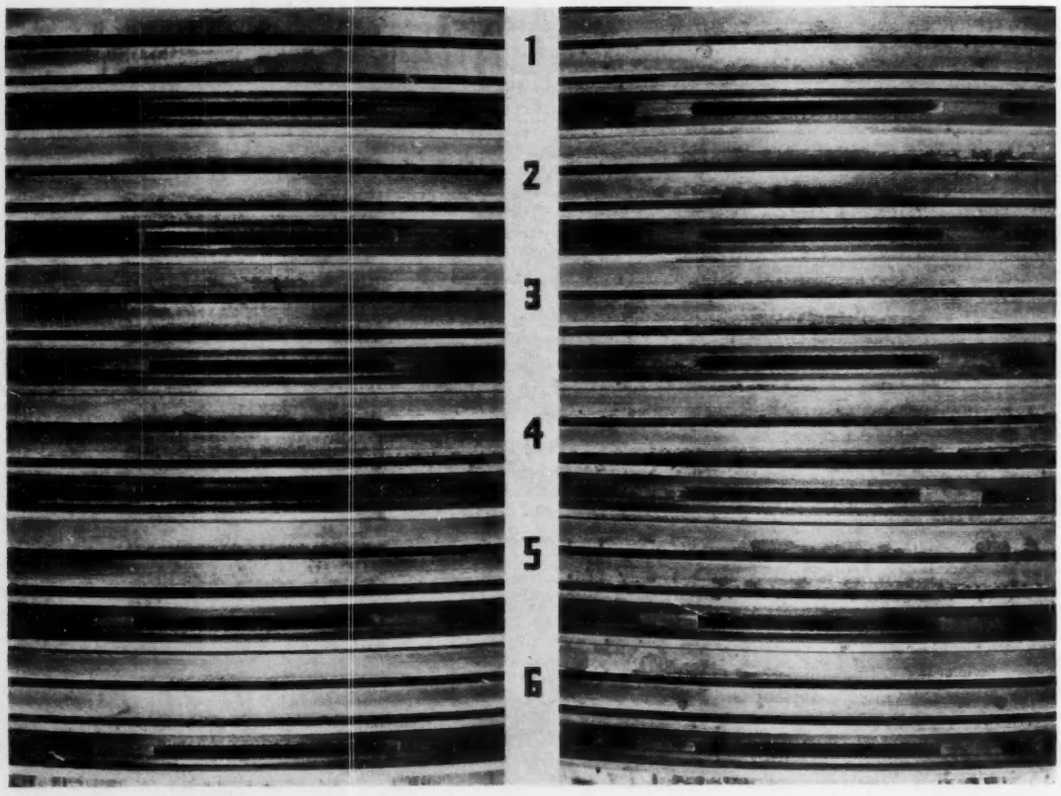


Fig. 15 – In Test 11 the Rings in No. 1 Cylinder Were Worn Badly Because of Previous Roughening of the Bores by the Use of a Lubricant of Unsafe Viscosity – This Run "Healed" the Bores Sufficiently to Prevent Excessive Wear in Test No. 12

Viscosity of 30 Oil at 170 deg. fahr. = 108 S.U.V.

bearings are in good condition despite the low viscosity of the lubricant with which they had been operated. In this connection it might be mentioned that on several occasions with straight kerosene as a lubricant we have operated at full load at 1000 r.p.m. for 50 hr. without any evidence of serious damage to either main or connecting-rods.

Bigger oil pumps and better lubrication systems eventually may change this picture. For the present, however, adequate flow appears to be the criterion by which the safe viscosity for connecting-rod bearings must be determined.

The choice of an oil to give safe viscosity depends upon the temperature at which it is to operate, and in this connection Fig. 10 is of interest. This figure shows temperatures of the crankcase oil for a dozen of the 1938 cars. At 70 m.p.h., the temperatures range from 89 to 178 deg. fahr. above the temperature of the atmosphere, whereas, at top speed, this range is from 105 to 195 deg. fahr.

Piston-Rings – The scuffing which occurs at low speeds is more likely to be due to a lack of oil than to insufficient viscosity. Under conditions of high speed and load which produce scuffing of the type shown in Fig. 11, viscosity is important. This point is illustrated by Fig. 12 which shows piston-rings after 1 hr. of full-load operation at 4000 r.p.m. With the 10-W oil (left), this one hour of operation was sufficient to wear away completely the tool marks on all of the rings in Cylinders 5 and 6, and the oil rings in all cylinders gave evidence of excessive wear. A similar run with No. 40 oil (right) produced comparatively little wear. Under these conditions, the viscosity of the 10-W oil hardly could be considered safe. A glance at Fig. 13 reveals a different picture. Here, rings, operated with the same oil, in the same engine, and under the same conditions, show no ill effects. The improvement is due solely to the "healing" of the bores by less than 3 hr. of operation with an oil of safe viscosity.

A similar illustration is furnished by a comparison of Figs. 14 and 15. The horrible condition of the rings in Fig. 14 leaves no doubt that the viscosity of a mixture of 80 per cent of 10-W oil and 20 per cent of kerosene is unsafe. Fig. 15 is of interest because it gives the results obtained in two tests made immediately following one of the runs with the 20 per cent kerosene mixture. In the first of these tests the roughness of the bores which had resulted from the previous run, caused the rings in No. 1 cylinder to wear excessively despite the fact that No. 30 oil was employed. The test, however, "healed" the bores sufficiently to prevent abnormal wear in the run that followed.

Surface Treatment of Piston-Rings – The characteristics of the material used in main and connecting-rod bearings are such as to reduce to a minimum the damage caused by momentary failure of the oil film. Recent developments in the surface treatment of piston-rings constitute a step in the same direction. Such treatments definitely increase the factor of safety against scuffing when the engine is operating near the borderline of safe lubrication. One example of the effectiveness of such treatments is furnished by Fig. 16. Obviously, the viscosity of the lubricant was safe only for the treated rings.

Piston – Scuffing of pistons usually is the result of a previous scuffing of piston-rings, but scoring occasionally occurs when the rings are in good condition. An increase in viscosity often increases the factor of safety against scoring and this condition is true particularly during the running-in period. In running-in one engine, a period of part-load operation was followed by 2 hr. at full load at speeds of 2500, 3000, 3500, and 4000 r.p.m. The running-in schedule was completed without trouble when No. 40 oil was used. When it was attempted to repeat the schedule with No. 20 oil, a piston scored at 2500 r.p.m.

In another case a rapid running-in schedule was followed which consisted of 15 min. of full-load operation at speeds of 2000, 2500, 3000, and 3500 r.p.m., followed by a full hour of full-load operation at 4000 r.p.m. This time No. 20 oil was used but with changes in jacket-water temperature to alter the viscosity of the oil upon the cylinder walls. Nothing happened during the first run in which the outlet temperature of the jacket water was not allowed to exceed 115 deg. fahr. The schedule was repeated with an outlet temperature of 140 deg. fahr. and No. 6 piston scored after ½ hr. of operation at 4000 r.p.m. No further trouble was encountered during the runs with a water temperature of 180 deg. fahr., but No. 4 piston scored at 3500 r.p.m. when the temperature was increased to 205 deg. fahr.

To be sure, the foregoing runs were made before the piston

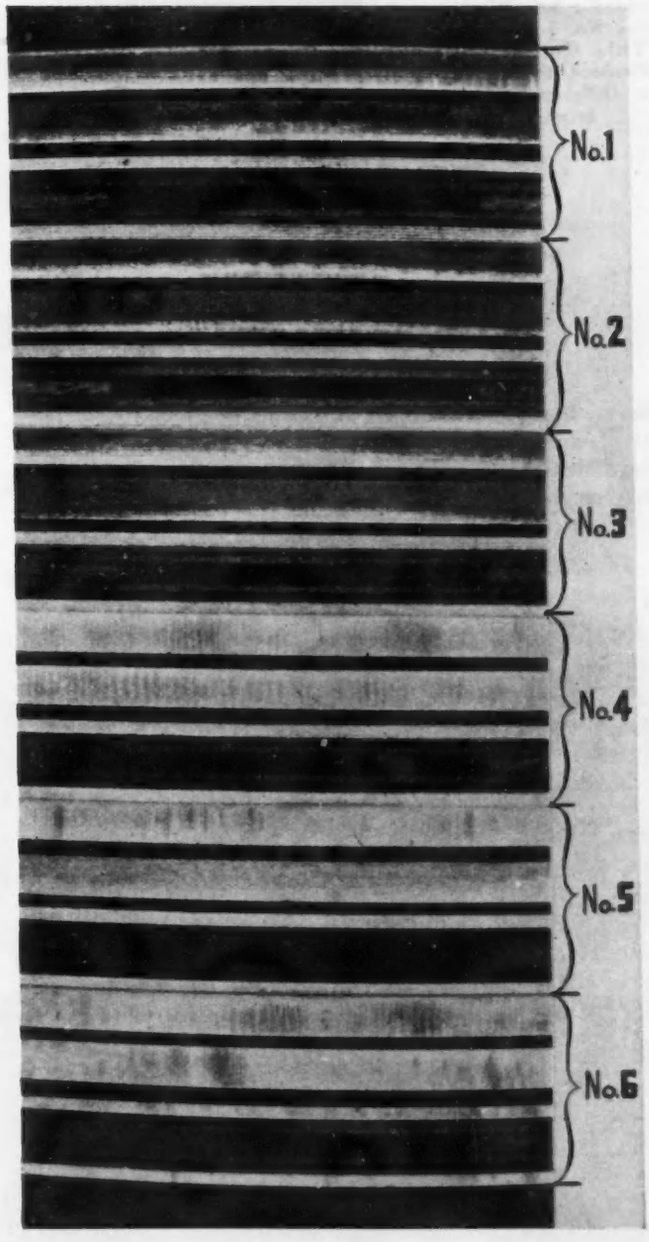


Fig. 16 – An Example of the Effectiveness of a Protective Surface Treatment of Piston-Rings

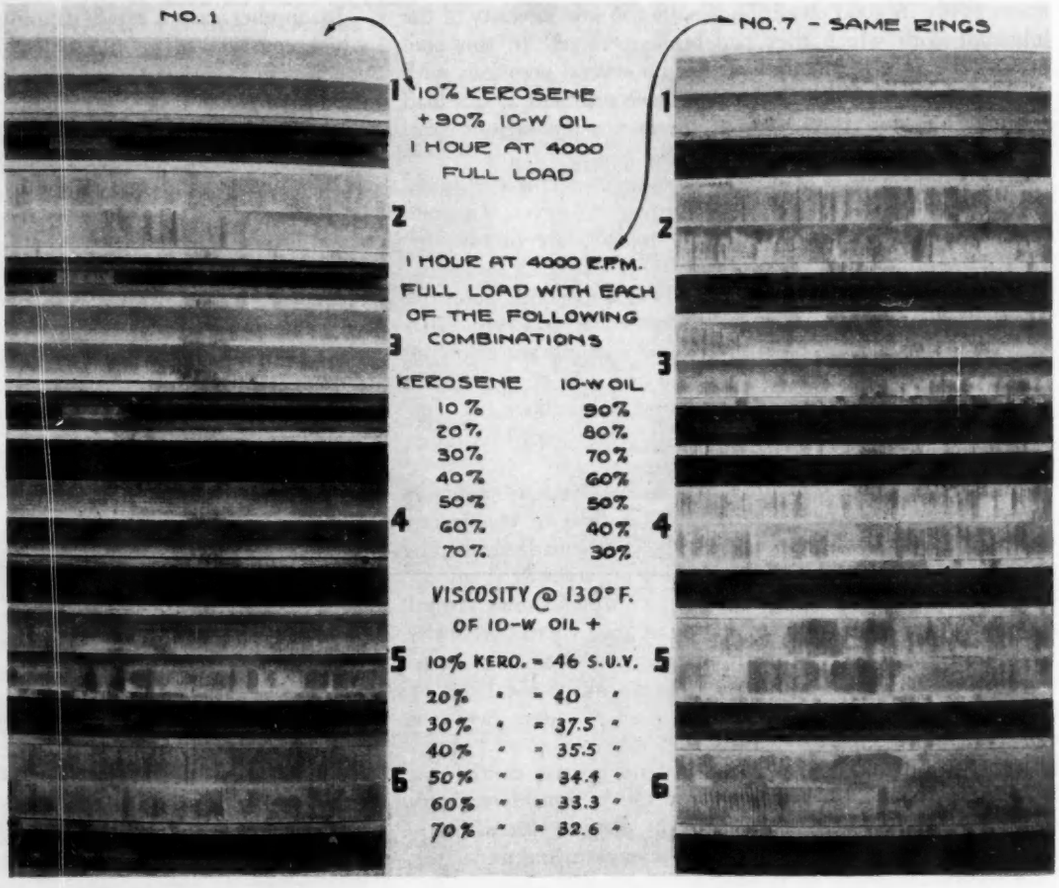
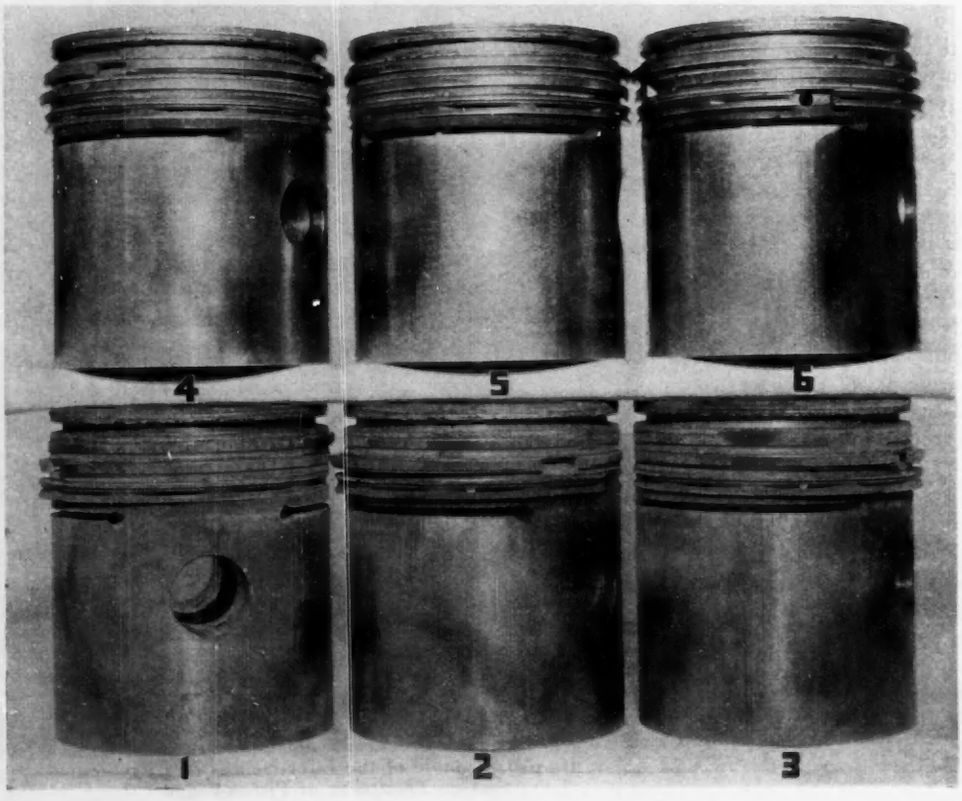


Fig. 17 – In This Test a Mixture of 90 Per Cent 10-W Oil and 10 Per Cent Kerosene Was of Unsafe Viscosity for Cylinders 5 and 6, Whereas a Mixture of 30 Per Cent 10-W Oil and 70 Per Cent Kerosene Provided a Safe Viscosity for No. 1 Cylinder – This Condition Is Probably the Result of a Difference in Surface Finish

design had been developed to a point which provided a normal factor of safety against scoring. Nevertheless, the results show that the viscosity of the lubricant on the cylinder walls may be an important factor in determining piston life.

Cylinder Bores – No special comment need be made on cylinder bores since a viscosity which provides adequate protection for pistons and piston-rings will be satisfactory for the cylinder walls.

Cylinder-Wall Finish – It is recognized generally that smooth bores are essential to the safe employment of oils of



Nos. 1, 2, and 3 Started 100 Times

Nos. 4, 5, and 6 Not Firing

Fig. 18 – Water Condensation on the Cylinder Walls Causes Rusting of Piston-Rings – There Was No Corrosion in the Cylinders Which Were Not Fired

low viscosity. Perhaps the extent to which the finish may affect viscosity requirements may not be appreciated so generally. Fig. 17 shows two photographs of a set of rings which were subjected to a series of tests with lubricants ranging from a mixture of 90 per cent of 10-W oil and 10 per cent of kerosene to a mixture of 30 per cent of 10-W oil and 70 per cent kerosene. For the rings in several of the cylinders the viscosity of the first mixture was woefully inadequate, whereas for the rings in No. 1 cylinder, even the mixture containing 70 per cent of kerosene appears to be of safe viscosity. As temperature conditions in this engine are not such as to favor No. 1 cylinder, its better showing is attributed to smoother cylinder walls, although no difference in surface finish was noticed prior to the test. Whatever the cause, the item of chief interest is the wide difference in the viscosity requirements of the different cylinders.

The owner of a new car is cautioned against driving it at high speeds until it has been run in. Few owners, however, realize that the lack of proper precautions in winter operation may leave the cylinder bores in such condition that the danger of scoring is greater than with new bores. Fig. 18 shows the nature of what may occur, although in this experiment conditions were made abnormally bad. This engine was started 100 times and was allowed to run only until it fired regularly in the three cylinders to which spark-plug wires were connected. The test was made at 70 deg. fahr., but winter conditions were simulated to the extent of using an oil too viscous to be thrown freely on the cylinder walls. Although the lubrication of Cylinders 4, 5 and 6 must have come from the gasoline thrown on the walls, these cylinders, which did not fire, were not harmed. In the cylinders which did fire, however, the water, which was condensed when the burned charge came in contact with the cold walls, caused the rings to rust badly and left the bores in the badly rusted condition shown in Fig. 19.

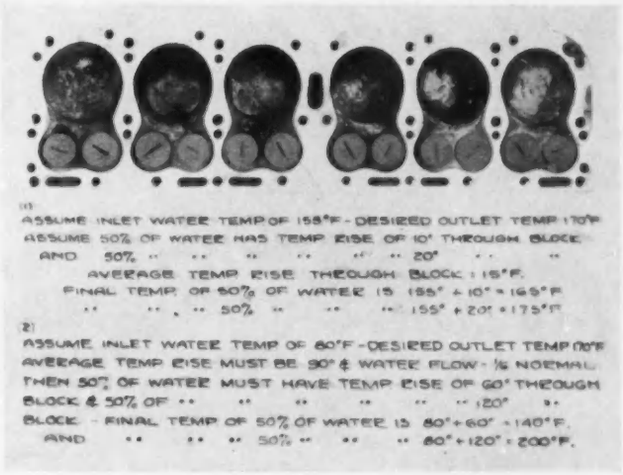


Fig. 21 – When a Thermostat Restricts the Flow of Water, There Is an Increase in the Magnitude of the Differences Between the Temperatures at Various Points in the Jacket

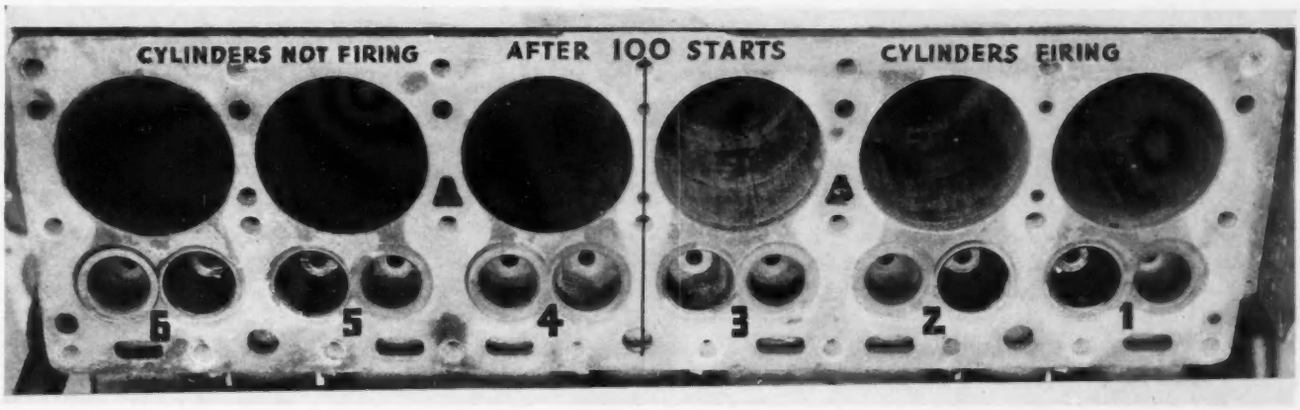


Fig. 19 – Water Condensation Causes Rusting of Cylinder Bores – There Was No Corrosion in the Cylinders Which Were Not Fired

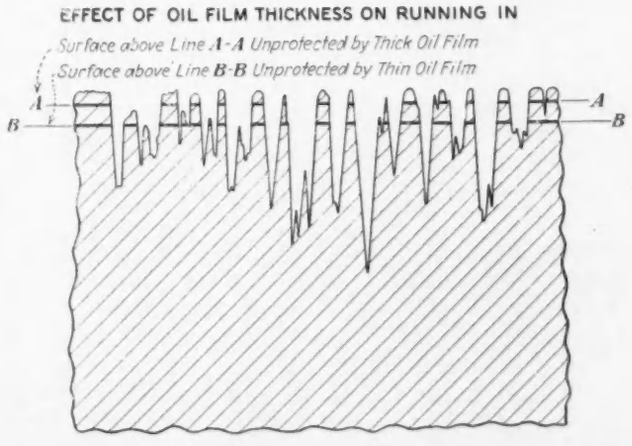


Fig. 20 – Low Viscosity Decreases the Time Required for Running In But Increases the Danger of Scoring Because of More Rapid Rate at Which Heat Is Generated by Metal-to-Metal Contact

Running-In – It is the author's belief that the viscosity of the oil used in a new engine should be at least as high as that which is specified for a well run-in engine. Nevertheless, there are good arguments in defense of the common practice of breaking-in an engine with an oil of low viscosity. If we consider the much magnified section of a cylinder wall shown in Fig. 20, it is reasonable to assume that more of the high-spots will break through the film when an oil of low viscosity is used. This condition means that the wearing down of the high-spots will be accomplished at a more rapid rate, and this action is desirable provided the heat generated is not sufficient to cause scuffing or scoring. From such evidence as is available it is believed that the greater protection against scoring which is provided by a more viscous oil more than compensates for the added time required to wear away the high-spots of the cylinder-wall surfaces. These remarks apply chiefly to pistons, piston-rings, and cylinder walls. In present-day engines, the main and connecting-rod bearings require very little running-in.

Winter Operation – Engines which have operated satisfactorily throughout the summer frequently develop trouble with pistons and rings during the winter months. Even Republicans and Democrats will agree that it is colder in the winter than in the summer. Hence, it is but natural that low temperatures should be blamed for such failures. Often the accusation is just for, as has been pointed out, a great deal of damage can be done by the corrosion produced by a combination of faulty lubrication and water condensation on the cylinder walls. It is our belief, however, that a large proportion of such failures are due to a combination of high jacket-water temperatures and lubricants of low viscosity. If a car has adequate cooling capacity and thermostatic control of the jacket-water temperature, then the average temperature of the jacket is likely to be as high in winter as in summer. In fact, if a car heater is installed, it is not uncommon to use an adjustable thermostat and to set it to maintain a higher temperature in the jacket while the heater is in service. Furthermore, if the temperature distribution throughout the jacket is not fairly uniform, the temperature differences will increase if the flow is restricted by means of a thermostat. Fig. 21 illustrates how a drop in the inlet temperature might even increase the temperature at the hot-spots. It is admitted that the latter possibility is somewhat remote and that the situation is not particularly alarming since a large share of winter driving is at moderate speeds. Nevertheless, it is well to recognize the fact that, as far as pistons and rings are concerned, to be safe, the viscosity of the lubricant should be as high for winter operation as for summer.

Conclusion – As the reader was warned at the outset, this paper is in no sense a report of an investigation of safe viscosity. It is a collection of miscellaneous information which seems to have a bearing on that subject and which has been accumulated as a byproduct of routine engine tests and development. It indicates that rather low viscosities may be safe for bearings if and when we can be sure that the amount of lubricant which reaches the bearings will be adequate. It indicates the extent to which safe lubrication of the cylinder bores depends upon the ability to produce and maintain smooth surfaces on pistons, piston-rings, and cylinder walls. In short, it indicates that, not until our dictionary defines the motor-car engine as a mechanism, perfect in construction and design, will it be wise to label the term viscosity, obsolete.

Advantages of Substratosphere Flight

WHY must we fly so high? So far, most airplane operations have been confined to levels of about 12,000 ft. or below, chiefly because, at greater elevations than this, atmospheric pressure and temperature are so low that people begin to experience some physical discomfort. At 12,000 ft., which is something like the average of the higher peaks in the Rockies, atmospheric pressure is about two-thirds that at sea level and, since the air constituents remain in the same proportion, there is available but two-thirds of the oxygen partial pressure to which people are accustomed. Under this condition, a few people feel slightly sub-normal through want of oxygen and, as we go higher, the number who experience the feeling rapidly increases. At 24,000 ft., there is only a bit more than one-third of the normal air available, very few being capable of remaining conscious without an extra oxygen supply. Then, too, as we go up, we find that the average air temperature drops nearly 4 degrees Fahrenheit for each thousand feet so that, whereas at 12,000 ft. we can expect an average 16 degrees Fahrenheit, at 24,000 ft. this drops to –27 degrees Fahrenheit and, at 36,000 ft. about where it stops dropping, it reaches an average of –67 degrees Fahrenheit. As a matter of interest, this point where the temperature stops falling is called the torpopause, and it varies in this country between the altitudes of 30,000 and 36,000 ft. The region below the torpopause is called the torposphere and that above, where temperature remains constant, is called the stratosphere. The term substratosphere now so casually thrown about might be defined roughly as that region of the atmosphere between the altitudes of 20,000 and 36,000 ft. At present, when we speak of high flight, we are referring to operation within this arbitrary substratospheric range.

From a standpoint of airline transportation, there are three good reasons for wanting to fly higher. These are, first, that we must do so if we expect to break free from the hazard of striking mountains with the attendant regrettable effects on airplane and occupants. Second, we must fly higher to avoid bad weather. Storms ordinarily do not extend above about 20,000 ft. and, as altitude is increased, the frequency of fog and bad weather diminishes. Just how high storm areas ever extend is still unknown, but we do know that they seldom climb up enough to encroach on the base of the stratosphere. Third, we can get considerably better cruising speed. The amount of improvements varies considerably with the airplane design but, as a broad average, it is about 1 per cent per 1000 ft. of increased altitude. Thus, an airplane which cruises at 200 m.p.h. with a given thrust horsepower at 12,000 ft. altitude might be expected with the same power and at twice the altitude to do at least 224 m.p.h. To explain this speed increase accurately would require considerable discussion of aerodynamic principles, but it can be explained approximately by saying that at higher altitudes, the air having a lower density offers less resistance or aerodynamic drag even though the wings must be trimmed to a slightly higher incidence to support the airplane's weight. These three points all have their qualifications and corollaries and there are a number of other reasons but, in view of the obstacles which must be overcome, these items may be neglected.

From the military standpoint, there are the dual advantages of speed increase and the possible freedom of bombardment types from harassment by ground or air-defense units. For private flying, there seems to be little apparent advantage. Even in transport and military operations, the use of high-altitude flight is limited to fairly long-haul work because of the handicaps presented by the time required to climb to and descend from such altitudes and the weight and cost of necessary equipment.

Excerpt from the paper: "High-Flight Engineering," by Kendall Perkins, Curtiss-Wright Corp., presented at the Chicago Section Meeting of the Society, May 20, 1938.

Credit Line Omitted from Taub Paper

The second part of the paper: "Motor-Car Engines in England," by Alex Taub, Vauxhall Motors, Ltd., dealing with carburetion, published in the June, 1938, Transactions Section of the S.A.E. JOURNAL, pp. 234-242, was taken from the paper: "Carburation," which Mr. Taub had presented previously before the Institution of Automobile Engineers, and was published on pp. 65-99 of the March, 1938, *Journal of the Institution of Automobile Engineers*. A footnote stating that the second part of the paper was published through the courtesy of the I.A.E., was erroneously omitted when Mr. Taub's paper was published in the S.A.E. JOURNAL.

Effect of Application on Maintenance of Automotive Electrical Equipment

By R. M. Critchfield

Chief Engineer, Delco-Remy Division, General Motors Corp.

MR. CRITCHFIELD attacks the problem of securing automotive electrical equipment that will operate satisfactorily at the lowest original as well as upkeep cost.

Integral parts of the electrical supply system, the ignition system and the cranking system are discussed with notes on maintenance of each. Particular attention is given to the various items which affect the sizes and types of equipment necessary. Mr. Critchfield points out that misapplications from both the capacity and operation standpoints are often the source of the most serious maintenance problems, and he gives specific examples to illustrate.

In the tables accompanying the paper the author presents application charts to serve as a rough guide in considering the purchase of new equipment.

WHEN the request was received to present a paper on maintenance, it was believed that the "Effect of Application on Maintenance" was a subject the writer was better qualified to discuss and one that involves several problems of considerable importance and interest. Operators can develop personnel and technique in handling maintenance problems, but they cannot reduce these problems to a minimum unless the equipment has been selected properly and applied originally.

In considering how best to treat this subject in view of the interdependence of the various electrical units, it seemed that a division into three groups would permit a logical segregation, and would simplify discussion of the different problems. These groups are:

The electrical supply system.

The ignition system.

The cranking system.

Each group will be examined and the details influencing its application discussed, followed by conclusions as to the correct equipment for various classes of service, together with observations on maintenance data submitted by fleet operations.

The Electrical Supply System

The electrical supply system, as its name indicates, consists of those elements necessary in the supply of the electrical energy on the vehicle. The electrical supply system is made up of the generator, generator control, driving means, the battery, and the necessary wiring.

The Generator

The following items should be considered in making a generator application:

The Amount and Usage of the Electrical Load – Only those loads should be considered which are thrown on the system for appreciable periods of time. Lights, heaters, marker lights, radio, ventilators, and so on, fall in this category. Electrical loads have increased during the past several years on all types of automotive vehicles. For example, radios, heaters, defrosters, and improved lighting have increased this load on passenger cars from a value of 7 amp. to around a maximum of 25 amp. These same items, plus clearance and marker lights, have increased the truck load from 7 amp. to a maximum of 30 amp. (6-volt system), while heaters, ventilators, and lighting requirements have increased the total electrical loads on motor-coach systems from 20 to a maximum of approximately 70 amp. (12-volt system).

The usage or percentage of running time during which these electrical loads are thrown on the system will determine the size of the generator. Obviously, if a fairly heavy load were thrown on the system only a small percentage of the total running time, the generator would not have to be as large as if this load were on the system practically all of the running time. If the maximum load is thrown on the system only a part of the time (for instance, the lighting load which is necessary only during time of night operation), energy stored in the battery can be used to help carry the peak load. If a battery of sufficient capacity is provided to serve this purpose, a slightly smaller generator can be used although, as will be pointed out later, the maintenance on the generator will be increased since it will be operating a greater percentage of the time at, or near, its capacity. This type of operation, of course, will result in a relatively high rate of cycling for the battery as it will be discharged appreciably each night and recharged the following day. Since the number of cycles is a measure of the battery life, this type of operation will reduce the length of available battery service. The average heavy-duty bus-type battery will give 400 to 600 cycles of life, depending on the extent or range of cycling. The cycling life of the smaller batteries will run from this figure down to about 200 cycles.

The Type of Service – City bus, interurban bus, interurban trucking, door-to-door delivery, taxicab, police car, and other types of operations must be considered. The type of service will determine the miles per day, month, or year of operation; the percentage of idling to running time; and the amount the operator could pay logically to avoid maintenance difficulties.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 13, 1938.]

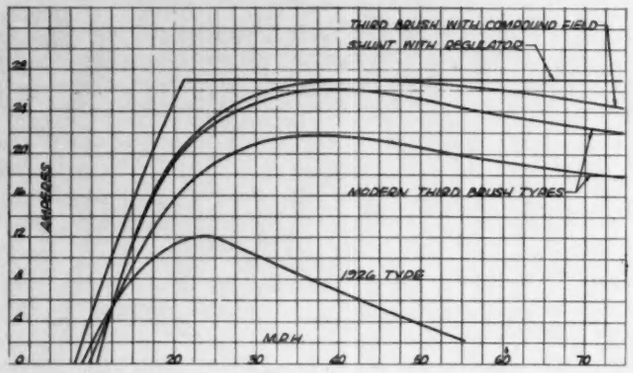


Fig. 1

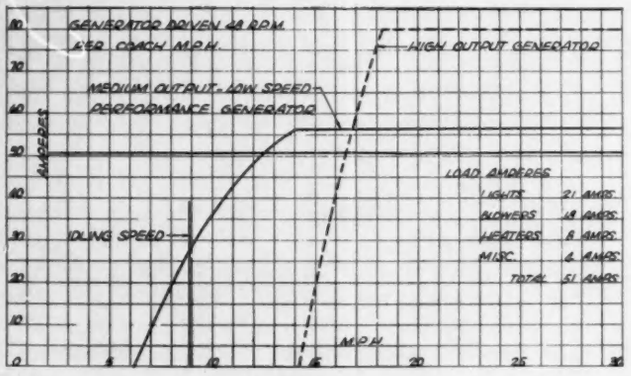


Fig. 2

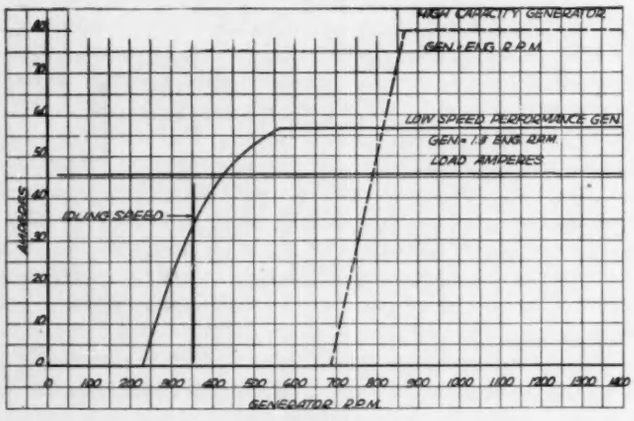


Fig. 3

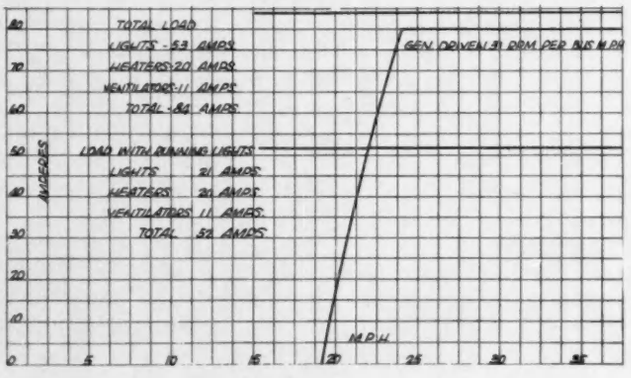


Fig. 4

Fig. 1 – Generator Hot-Output Curves

Fig. 2 – Effect of Low-Speed Performance on City-Coach Generator Operation

Fig. 3 – Another Example of the Effect of Low-Speed Performance on City-Coach Generator Operation

Fig. 4 – Electrical Loads and Generator Performance – 36-Passenger Interurban Bus

It also will determine whether the unit should be ventilated or enclosed. Examples of enclosed applications are those for street sweepers, farm tractors, and vehicles to be used for hauling highly explosive materials. It is obvious that, where possible, the ventilated unit is to be preferred since, with approximately the same size and cost, a much greater output can be obtained.

The Type of Drive and Mounting – The type of drive (gear, belt, or coupling) affects maintenance in that improper applications are a constant source of trouble and expense. The driving means should be adequate to carry the load and should permit proper alignment and adjustment.

The mounting on the engine and location of the generator in the engine compartment will influence the choice of the generator size, since the temperature of the surrounding air will affect its capacity. It is evident that operation in a high-temperature atmosphere will reduce the capacity of a generator of a given size.

A smaller generator operated at a high r.p.m. per vehicle m.p.h. will deliver the same output as a larger, more expensive generator operated at a lower speed. Obviously, other things being equal, the brushes, bearings, and commutator of the slower-speed machine will last longer. One large operation reports: "The larger generators require much less maintenance due to lower temperatures, consequently longer brush, commutator, and armature life. We find the life of the larger units to be almost double that of units with small surplus capacity. We ask for 20 per cent surplus capacity of generator over load."

Operating a generator over long periods of time at or above its rated output will not only affect brush and commutator life, but will, due to the higher temperatures obtained under these conditions, shorten the life of bearings and of the armature and field insulation. Overloads or operation at high ambient temperatures will reduce greatly the life of the insulation due to deterioration of this material at an increasingly higher rate as higher temperatures are obtained.

Considering the foregoing facts, the following observations are made with reference to different types of generator applications:

For Passenger Cars and State Police – The generator capacity should approximately equal the total load, including accessories. In order to have adequate lighting for night driving, voltage must be maintained at the headlight bulbs. This condition requires that the generator should have sufficient capacity to carry the total night load and keep the battery reasonably well charged. This requirement means that the generator should have a capacity of from 150 to 200 watts. The drive ratio and the generator low-speed performance should be such that this load will be carried from 20 m.p.h. on up. Fig. 1 gives the performance curves of various modern passenger-car generators compared with generator performance prevalent several years ago.

City Police Cars with One-Way Radio Equipment and Taxicabs – Generator capacity should be ample to carry the entire load. However, for this service, the low-speed output and drive ratio should be such as to carry this load at from 10 to 12 m.p.h. on up. Generator capacity for this type of service generally runs from 150 to 200 watts. Although this is the same capacity as supplied for passenger cars, generators will be larger due to the necessity of developing their output at lower speeds.

City Police Cars with Two-Way Radio Installations – These cars should have generators with additional 20 to 40 watts capacity. The two-way radio, when transmitting from the car, increases the power requirements much more than this amount, but the transmitting feature, and hence the increased power demand, is used only a small percentage of the time.

Door-to-Door Delivery, Trash and Garbage Collection – When lighting capacity or other extra load is required, a generator of 100 to 150 watts capacity is required. Due to the fact that a large percentage of the total operation is with engine idling, the generator low-speed performance and drive ratio should be such as to carry the lighting and ignition load at idling speeds. Since this type of service is generally at low speeds, and the engine is governed to prevent damage which otherwise would occur to the generator when the engine is raced in gear, this performance can be obtained with a small generator driven at from 2 to 2.5 times engine speed.

City Trucking – Generator capacity should be sufficient and drive ratio such as to carry the electrical load at from 12 m.p.h. on up. A generator of 100 watts capacity should be satisfactory for most applications.

Intercity Trucking – Generator output should be such as to take care of marker and clearance lights, headlights, heater, sometimes a radio and, in the case of a tractor unit, the extra marker and clearance lights for the trailer. The generator should carry this load above 20 m.p.h. Capacity of 200 to 600 watts generally is required. The drive ratio should be such that maximum generator speed be kept below 5000 r.p.m. – preferably 4000 r.p.m.

City Bus – For this service the total possible *continuous* load should not exceed the generator capacity. The exact rating will depend to a great extent on drive ratio, freedom desired from maintenance troubles, cost, and so on. Thus, with a generator that will not cut in until 8 to 10 m.p.h., the load should not exceed 60 per cent of the generator capacity as there will be frequent periods of idling and low-speed operation during which the battery will be called upon to supply power. The generator should have excess capacity at the higher speeds to be able to replenish the battery. If a larger generator is considered having better low-speed performance – for instance, with a cut-in of around 4 to 5 m.p.h. – the load can be as great as 90 to 100 per cent of its capacity.

Fig. 2 illustrates the importance of selecting the proper low-speed performance and drive ratio for the generator, especially for service in congested city areas where the idling periods average 40 per cent of the total operating time and may run as high as 50 per cent. Curves on this chart show the generator output in amperes plotted against bus speed in m.p.h. The ampere load also is shown, as well as the output obtained with the engine idling. The dotted curve indicates the performance of an 80-amp. generator driven at engine speed. Difficulty was experienced in maintaining the battery in a charged condition, although the generator apparently provided an ample margin of output over the amount of electrical load. Investigation disclosed that this operation resulted in a very high percentage of idling time and, since the generator did not charge at idle, the amount of battery discharge during idling and at low speeds was more than the generator could make up for at higher speeds. Changing the generator so that the low-speed performance was improved, even though its capacity was reduced, as indicated by the solid line, permitted the battery to be maintained in a charged condition. With this change, the low-speed performance permitted a charge of approximately 27 amp. at idle.

Operators try to set engine idle speed at the lowest possible point in an effort to reduce fuel consumption during idling periods. Automatic transmission and gear-shifting devices generally require low idling speed for satisfactory operation. Hence, the low-speed performance and drive ratio of the generator should be selected so as to harmonize with these other requirements.

Fig. 3 illustrates another example of the advantage of proper low-speed performance of a city-bus generator application. Difficulty was experienced in keeping the battery charged with a high-output generator (performance indicated by dotted line). Due to the high percentage of idling time in operation, the cause of battery trouble can be traced to the fact that there was no generator output at idling speed. In this case, a generator of improved low speed performance was substituted and driven 1.3 engine speed as against the original drive ratio of 1:1. With this combination, the generator produced approximately 33 amp. at idling, and the battery trouble was cleared up immediately.

Generator capacity for city-bus service should run from 600 to 1200 watts.

Interurban Bus – For this type of service, the generator size should be such that the load will not exceed 80 to 90 per cent of its capacity. The generator low-speed performance and the drive ratio should be such that the generator will cut in around 20 m.p.h.

Fig. 4 presents data for generator performance and electrical loads for a modern 36-passenger interurban bus. Although the total load exceeds slightly the maximum generator output, this total is made up, among other items, of both dome and individual seat lights and this combination is used rarely in service. The generator cuts in at slightly over 19 m.p.h. and carries the running electrical load above 23 m.p.h.

Generator capacity should run from 600 to 1200 watts, and the units should be built to withstand high mileages without failure.

Particular attention should be paid to drive ratios: first, to see that low-speed performance is satisfactory for low-speed operations and for those involving a high percentage of idling as indicated by the examples cited previously; second, to see that for high-speed applications the drive ratio is not unnecessarily high, resulting in excessive generator rotative speeds at the higher vehicle speeds with attendant short life of brush, bearings, and so on. The writer has in mind a case of a trucking concern engaged in intercity operation. Two different make trucks were employed using the same 200-watt generator, one driven at approximately 60 generator r.p.m. per m.p.h., the other at approximately 85 generator r.p.m. per m.p.h. Both makes of trucks had essentially the same load and were used in the same type of service. The batteries in each case were maintained in a charged condition. However, from 50,000 to 70,000 miles brush life was obtained on the trucks using the lower drive ratio as against 20,000 to 25,000 miles on the trucks with the unnecessarily high drive ratio.

Frequent reference to the necessity of obtaining the correct generator drive ratio in making applications stresses the desirability of having a driving means which can be changed readily to give different ratios. The V-belt drive is about the only type that meets this important requirement. The use of this type of drive also affords the generator a certain amount of protection from engine torsional vibration. It is necessary that a V-belt drive be designed properly to handle adequately the power required to drive the generator. This point is sometimes overlooked in applying a high-capacity generator on an engine normally supplied with a small unit. The sizes and angles of belt drives now in use vary to an unnecessary extent. In an effort to reduce the sizes required and thus simplify the service problem, the Society has adopted V-belt standards, and these standards should be adhered to where possible.

In an effort to summarize as briefly as possible the generator-application problem, Fig. 5 has been prepared. Various applications are listed, together with information on electrical loads, generator capacities, drive ratios, weights, and battery capacities.

In examining data on bus battery applications, it was noted that capacities tend to vary with passenger capacity. For city buses, battery capacity at the 4-hr. rate varies from 3 to 4½

	LOAD AMPERES	VOLTAGE	GENERATOR OUTPUT WATTS	GENERATOR MAX. SPEED	GENERATOR CRUISING SPEED	GENERATOR RPM/MPH	GENERATOR WEIGHT POUNDS	* BATTERY AH CAPACITY	NOTES
PASSENGER CAR & STATE POLICE	12-25	6	150-225	8500	4000	80-90	20	90-125	CARRY LIGHTING LOAD ABOVE 15 M.P.H. CARRY TOTAL LOAD ABOVE 25 M.P.H.
CITY POLICE & TAXICABS WITH RADIO	18-23	6	150-200	6000	2500	90-110	32	115-130	CARRY 50% OF LOAD AT IDLE
CITY POLICE WITH 2 WAY RADIO	20-25	6	200-250	6000	2500	90-110	40	120-140	CARRY 50% OF LOAD AT IDLE
DOOR TO DOOR PICKUP AND DELIVERY	6-10	6	100-150	7500	2500	120-135	25	90-120	CARRY 75% OF LOAD AT IDLE
CITY TRUCKING	8-15	6	100-150	7500	3000	80-90	20	80-120	CARRY LOAD ABOVE 12 M.P.H.
INTERURBAN TRUCKING	20-30	6	200 275	5000 4500	3000 2500	70 60	32 40	100-140	CARRY TOTAL LOAD ABOVE 20 M.P.H. AVOID EXCESSIVE GENERATOR SPEED
INTERURBAN TRACTOR & TRAILER	15-30	12	250 600	5000 4500	3000 2500	70 50	32 50	70-100	CARRY TOTAL LOAD ABOVE 20 M.P.H. AVOID EXCESSIVE GENERATOR SPEED
CITY BUS	20-65	12	375 1200	5000 4000	2500 2000	75-100	40 120	85-280	CARRY 50% OF LOAD AT IDLE
INTERURBAN BUS	20-80	12	375 1200	4000 3000	2500 2000	50 30	40 120	100-240	CARRY TOTAL LOAD ABOVE 25 M.P.H. AVOID EXCESSIVE GENERATOR SPEED

* BUS BATTERIES ARE LISTED AT 4 HOUR RATE ALL OTHERS AT 20 HOUR RATE

Fig. 5 (*left*) – Generator Application Chart

Fig. 6 (*below*) – Effect of Charging-Rate Variation on Terminal Voltage of 112 Amp-Hr. Battery

amp-hr. per passenger whereas, with interurban buses, the range is from 4 to 6½ amp-hr. per passenger, the 6½ amp-hr. figure applying to coaches used in overnight service.

Recommended capacities and drive ratios apply to average conditions found in service and should enable the generator to give satisfactory performance and brush life.

Notes on Generator Maintenance – A study of maintenance data obtained from various operations bears out statements made previously to the effect that the use of adequate capacity generators results in longer life and lower upkeep expense. Thus, with 600- to 750-watt bus generators, brushes are scheduled for replacement at from 50,000 to 60,000 miles whereas, with the higher capacity 1000-watt units, brushes are replaced at from 90,000 to 150,000 miles. Inspection periods vary from 10,000- to 25,000-mile intervals. In some cases, generators are blown out and brushes freed in holders at this time and, in others, the units are checked for performance only.

In truck service also, it apparently has been found necessary to provide different maintenance routine for the lower-capacity units. One large operation reports that it is necessary to check the small units (125 to 150 watts) at from 7000- to 10,000-mile intervals whereas, the larger units (approximately 200 watts), it is only necessary to inspect every 25,000 to 30,000 miles.

In most cases, the periodic inspections are supplemented by some kind of driver's report on failures whereas, in a few instances, drivers' reports are relied upon entirely.

Regulation

Even with the lowest capacity generators in use today, there is no question as to the necessity of providing regulation or some form of control of the generator output. If the electrical load on a system were always constant, a generator of a capacity to just balance this load could be selected and there would be no need of output control. In actual practice, however, practically all vehicles operate a percentage of the time with ignition load only. Without regulation the excess generator capacity would have to be absorbed by the battery and, under some conditions, it would overcharge it seriously, causing overheating, excessive water usage, buckling of the plates, loss of active material, and premature failure. A large trucking operation reports: "With generators voltage-regulated, batteries take about 40 per cent as much water as is used without regulation even though the generator capacity may be much lower with the non-regulated type." In addition, as a result of the overcharge, high voltages would be

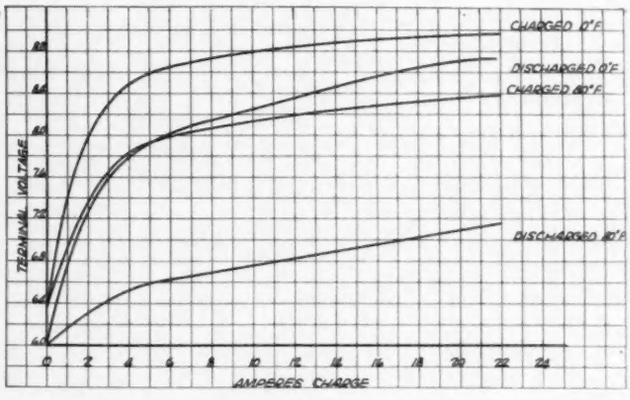


Fig. 6

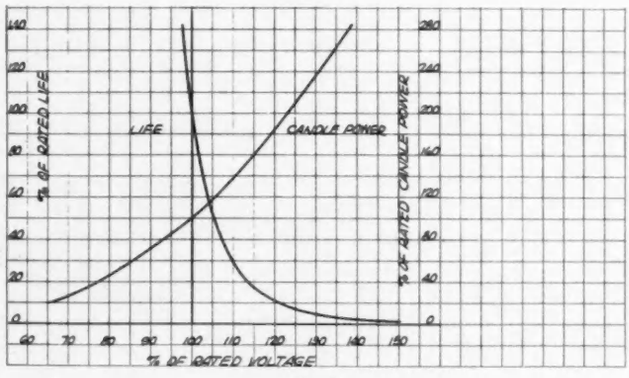


Fig. 7

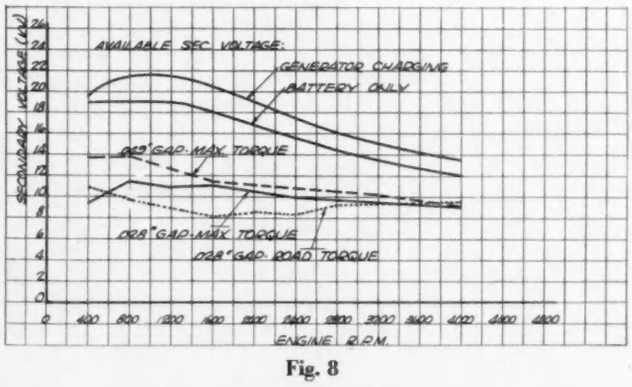


Fig. 8

Fig. 7 – Effect of Voltage Variation on Light-Bulb Life and Candlepower

Fig. 8 – Secondary Voltage Requirements on 1937 Eight-Cylinder Engine

obtained in the system causing oxidizing of the ignition contacts and short lamp and radio-bulb life.

Fig. 6 shows characteristic charge voltage curves of 112 amp-hr. capacity storage battery, and indicates the voltages reached under different conditions of charge and temperature without regulation. Thus, although only 6.9 volts are obtained in charging at 15 amp. at 80 deg. fahr., 8.5 volts are obtained while charging at a similar rate of 0 deg. fahr. The use of a larger battery will not reduce materially the unregulated voltage during overcharge periods.

Fig. 7 gives the relationship between voltage and lamp-bulb life. The reduction in life with increase in voltage is very evident; thus with a voltage 10 per cent higher than rated, the life will be reduced by 70 per cent.

Lack of voltage regulation also results in the generator operating at full output whether needed or not, with resulting unnecessary brush wear and deterioration of insulation.

The simplest type of control consists of a lighting switch arranged with extra contacts controlling the generator field resistance so that, when the lights are on, the field resistance is shorted out and full generator output is obtained whereas, when the lights are off, the resistance is inserted in the field circuit and the output is reduced to a reasonable value. This type of control is used extensively on farm-tractor applications and on some truck and commercial units.

Voltage regulation alone is applied ordinarily to the third-brush-type generator wherein the third-brush action limits the maximum output to a safe value. This arrangement provides the least expensive regulated system and, for passenger car work, it is proving very satisfactory. Yearly production of all vibrating voltage regulators has increased from around 200,000 units in 1935 to over 2,000,000 in 1938.

Both voltage and current regulation is required with the shunt-type generator (practically all high-capacity types). The current regulator unit is added to limit the maximum safe output of the generator as, without this regulation, the shunt-generator characteristic is such that, with increasing speeds, the output would continue to climb far beyond the safe capacity.

The voltage regulator setting is a compromise between that necessary to maintain the battery in a charged condition and that which will protect bulbs, ignition contacts, and other current-consuming devices from over-voltage.

Inasmuch as the generator field design seriously affects the regulator operation, regulators always should be used with the generators for which they are specified. The current-carrying elements of the regulator unit, which are the current regulator winding, relay series winding, and load contacts, bus bars, and so on, must be capable of handling the generator output without overheating. The regulator contacts, voltage windings, and field resistances must be capable of holding the generator output down to a value necessary to obtain the specified settings. Inasmuch as generator field windings, brush neutral settings, speed ranges, and so on, all affect the ability of the regulator to do this job, it is important that the proper regulator be used with each type of generator. For this reason, the promiscuous addition of regulators to non-regulated generators is inadvisable. Some non-regulated generators require so much field current as to make satisfactory regulator operation impossible.

Regulator Maintenance Notes – When a regulator is operating, the control points are breaking a highly inductive generator field circuit. It is to be expected that some wear of these contacts will result. This wear changes the calibration gradually, so that a periodic check is necessary. Reports from operations indicate that these periods run from 2500 to 25,000 miles, at which time the points are cleaned and the regulator is adjusted, whereas the points are replaced at from 40,000 to 100,000 miles.

In cleaning contacts, use of sand or emery paper should be avoided as grains of sand or emery are apt to be imbedded in the point material. Care must be taken to see that oil, grease, and dust, as well as foreign materials such as iron or brass filings, are not allowed on the contacts.

Connections between the regulator, generator, battery, and ground should be made permanently (no slip connectors) with adequate size wire provided with good insulation which will withstand oil, moisture, mechanical abrasion, and other abuse.

The importance of thorough maintenance of regulators cannot be overemphasized. Failure to provide for this maintenance will result in no end of expense with generators, batteries, bulbs, and so on, as well as with the regulator units themselves. With proper regulation, bulb life should be normal, although a condition of body vibration sometimes is encountered, which causes premature bulb-filament failures.

Batteries

The necessity of selecting a battery of the proper size already has been mentioned. For smaller vehicles having only the normal lighting and accessory load, the size of the battery is determined by the cranking requirements of the engine. With buses, however, having considerable additional electrical load, the battery size is determined largely by the size of this load and the generator capacity. In general, the greater the margin of the generator capacity over the electrical load, the smaller the capacity required in the battery, as long, of course, as the battery is not so small as to affect the cranking-system performance.

Batteries should be located so as to be easily accessible for inspection and service operations. Mountings should be such as to protect the battery from heat, excessive vibration, and mechanical shock.

Battery Maintenance Notes – It seems futile to add anything to the information already available on this subject. Batteries should be checked regularly and records kept of gravity and water consumption as these figures soon will indicate whether or not the electrical supply system is performing satisfactorily. Low gravity readings indicate improper regulator adjustment, inadequate generator capacity, or carelessness on the part of the operator in the use of electrical loads when not necessary. Operation of the battery over long periods at low gravity will, of course, cause sulphation and result in the permanent lowering of its capacity and premature failure of the plates. High water consumption indicates improper regulator adjustment or a ground in the generator field circuit, thus short-circuiting the regulator. High water consumption results from overcharging, with consequent injurious gassing, overheating, and dislodging of the active material from the plates. Battery temperatures should not exceed 125 deg. fahr. in normal service. Temperatures as high as 150 deg. fahr. would be considered extremely harmful. In watching the gravity of a battery, allowance should be made for normal decrease in gravity as it ages, even when fully charged. For instance, a battery that has a fully charged balanced gravity of 1.285 when new, after a couple of years, will show a gravity reading around 1.260 under similar conditions.

Wiring and Connections

Wiring and connections serve the important function of connecting together the operating units of the electrical system. Skimping on wiring is a poor way to save. When the cost of the generator, regulator, battery, and other units is taken into account and it is realized that skimping in an effort to save a few cents may impair seriously the operation

of these expensive units, this practice will be discontinued.

Undersize wire or poor connections in the circuits between the generator, regulator, battery, and ground tend to overload the generator and thus reduce the useful output obtainable without damage due to overheating, and so on. High-resistance connections in the regulator tend to cause overheating with charring and failure of the insulation and windings.

Similarly in the lighting circuits, undersize leads or poor connections may result in poor lighting or variation in lighting efficiency from one end of the coach to the other.

Wire used should be provided with adequate insulation protected by tough braid or braids to withstand abrasion and resist action of oil, gasoline, water, and so on.

For bus operations where heavy loads are handled, all connections should be positive and of sufficient size to carry the load and resist mechanical breakage. Slip-type connections should be barred entirely. Wiring should be installed properly and should be inspected regularly to avoid road failures or loss due to abnormal voltages or short circuits with the resultant fire hazards and other risks.

The Ignition System

The ignition system includes: the distributor circuit-breaker unit, the coil, the condenser, low- and high-tension wiring, the spark-plugs, and a source of electrical energy.

The ignition system comprises two circuits. The primary low-tension circuit includes the primary of the ignition coil, the condenser, and the circuit-breaker contacts, and it controls the time at which the spark occurs in the engine cylinder. The secondary high-tension circuit includes the secondary of the ignition coil, the rotor and distributor cap, the high-tension wiring and the spark-plugs.

Ignition equipment is classified sometimes as standard and heavy duty. As generally applied, the standard type refers to units produced in large quantities for passenger-car or light-commercial-car equipment. Heavy duty refers to equipment for large commercial vehicles.

Fundamentally, there is no difference in performance between standard and heavy-duty equipment, the distinction being principally due to the mechanical construction of the units and to the increased factor of safety in the insulation of the heavy-duty units.

An attempt will be made to discuss the various factors affecting the ignition-system requirements which sometimes dictate the use of equipment other than standard, since the latter type is usually the most economical.

Type of Service – Whether or not a vehicle is operated occasionally, as a passenger car, or whether it is operated many miles a month hauling passengers or freight on scheduled runs where freedom from failure and maintenance difficulties is of paramount importance, probably will determine the justification of extra expense for ignition equipment which will provide still a greater margin of safety and reliability than that provided by the average passenger-car equipment. Obviously, with a motor operating 125,000 miles per year, service is more severe and failures are of greater consequence than would be the case with a small delivery truck or passenger car operating at perhaps one-tenth of this annual mileage.

Operation consistently within fairly narrow speed ranges also affects the ignition-system performance requirements.

Engine Ignition Requirements – The compression ratio, carburetion, manifolding, throttle opening, spark-plug location, spark-plug gap, engine temperature, arrangement of high tension leads – all affect ignition requirements. Fig. 8 illustrates the effect of engine load and plug gap on ignition requirements. Curves are plotted showing the voltage required to fire a 0.028-in. gap with road and maximum torque and an 0.049-in. gap with maximum engine torque. As would be expected, these curves indicate that a larger plug gap, which is sometimes necessary to fire a lean mixture, requires a higher voltage than a smaller gap and that, with the same gap, full-throttle operation requires a higher voltage than does part throttle. Thus, an engine operated at full throttle most of the time or with a lean or poorly distributed mixture, requiring a larger plug gap, will impose higher voltage requirements on the ignition system.

The automatic advance curve is determined from dynamometer and road tests so as to obtain maximum power without objectionable spark rap.

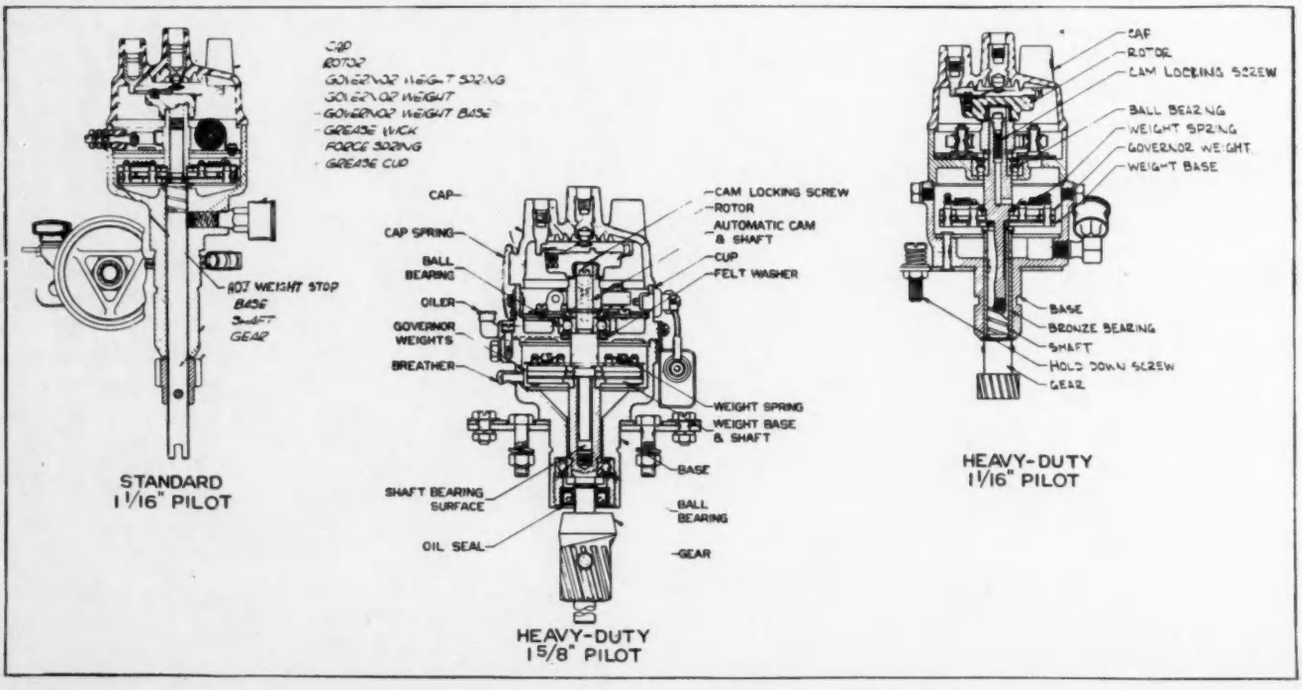


Fig. 9 – Standard and Heavy-Duty Distributors

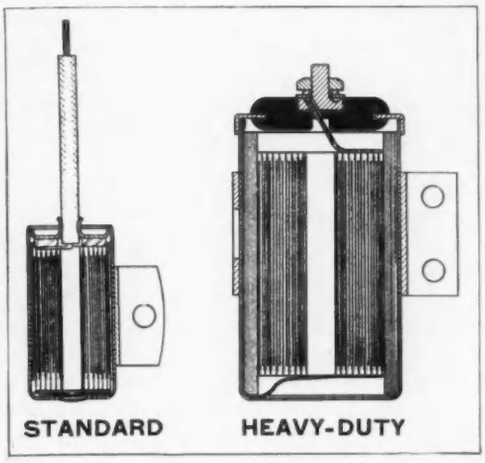


Fig. 10 – Standard and Heavy-Duty Condensers

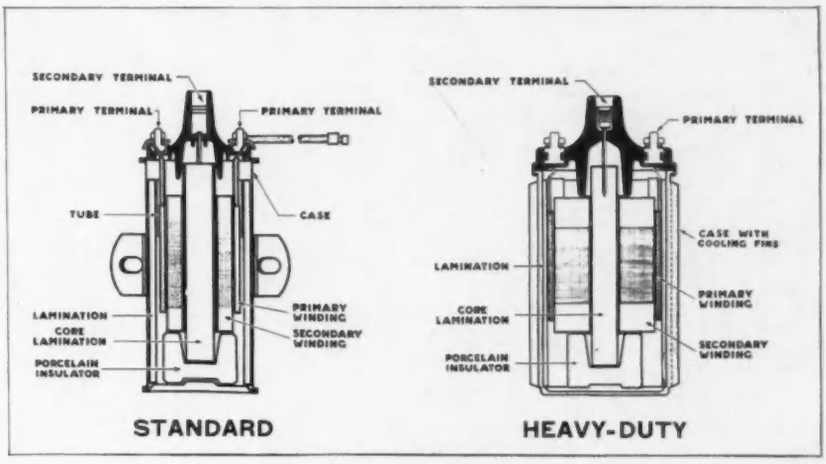


Fig. 11 – Standard and Heavy-Duty Ignition Coils

Prevailing part-throttle operation with the resulting lower compressions and slower rate of flame propagation, as encountered in passenger-car and some types of intercity-bus service, requires additional spark advance for best performance and fuel economy. Operations of this type indicate the need of vacuum spark control to obtain best economy. The method of the application of vacuum spark timing control has been treated in considerable detail by the author in a recent paper.[1]

The Distributor Mounting – The S.A.E. standard drive in either the 1 1/16-in. or 1⅜-in. diameter pilot is usually the most economical for general applications since it can be produced in small quantities without special tooling. Occasionally some particular mechanical feature in the engine may make it advisable to use special mountings, but this condition should be the exception.

For average service requirements the 1 1/16-in. pilot distributor is supplied with a plain cast-iron bearing. Where higher mileage applications are considered, some forms of bronze bearing may become advisable. For still more severe service conditions, a ball bearing is added at the top of the shaft just below the circuit-breaker cam. These units sometimes are designated as semi-heavy-duty, and have proved to be very satisfactory in motor-coach operation. A still more durable and costly construction is obtained by the use of two ball bearings in the 1⅜-in. pilot diameter unit. Another advantage of the larger pilot is that it permits the use of a larger diameter gear in applications where the maximum diameter gear in a 1 1/16-in. pilot has been found to wear excessively. Fig. 9 illustrates typical examples of the three designs just described.

Engine Torsional Vibration – Some engines impose excessive vibration or torsional impulses on the distributor unit. Resulting damage generally shows up in the automatic-advance mechanism. In order to withstand this type of service it is sometimes necessary to increase sections, use alloy steels, braze assemblies, and carburize and harden parts in the automatic-advance mechanism. Even with these precautions the unit still may be subject to excessive gear or coupling wear and may reproduce engine impulses into spark-advance variation with possible impairment of engine performance.

The Distributor Cap and Rotor – These parts, comprising the high-tension parts of the unit, usually are made of phenol-resin compounds. The exact material to be used will depend upon service conditions. Standard phenol-resin compounds may fail under certain conditions. Repeated surface moisture from condensation, rain, or washing may cause leakage which, if continued, will cause permanent failure by carbonization of the surface of the material. Frequent applications of water, particularly if the parts are hot, also may cause cracking of standard phenol-resin caps. Where it is impossible to eliminate this type of treatment, these parts can be molded with special compounds which better resist the actions just described.

Parts usually are made of this special material for heavy-duty equipment, but may be supplied on standard equipment at additional cost if necessary to meet special conditions. This special material is non-cracking and does not suffer permanent injury if occasionally subjected to heat, moisture, and electrical leakage. Distributor caps can be furnished with push connections for the high-tension leads or, to insure against accidental removal of leads, they can be furnished with screw connections at additional cost.

The Distributor Condenser – Condensers are classed as either standard or heavy-duty, depending upon the amount of insulation used and not upon the capacity. For some heavy-duty units manufacturers have specified mica in place of impregnated paper for insulation between the condenser elements. Mica condensers are much higher in price and, in view of the present development in the processing and impregnation of the paper insulation, it is questionable if the additional cost is justified. Condenser life and performance also are affected by heat and moisture. Fig. 10 illustrates the construction of standard and heavy-duty condensers.

The High-Tension Wiring – Although the high-tension wiring installation is not often given a great deal of attention, it plays quite an important part in the proper functioning of the ignition system. The electrostatic capacity between adjacent leads should be kept small with respect to their capacity to ground, otherwise cross-firing will result, that is, energy will be transferred to the wrong plug, resulting in firing at the improper time. This condition is particularly true with the wide plug gaps necessary in some engines due to the higher voltages required. Where it is not possible to separate the leads sufficiently to keep the capacity between leads to a low enough value, they should be enclosed in a well-grounded metal conduit. Adding this shielding tends to reduce the maximum voltage available at the plug. In some cases it may be advisable to use special low-capacity, high-tension cable.

The Ignition Coil – Standard units and those for heavy-duty service are not necessarily different in performance. The characteristics of the heavy-duty units are sometimes modified to emphasize their performance at the lower and intermediate speeds where heavy-duty equipment generally operates. Insulation, both in the winding and in the molded parts, is generally special, the better to withstand high mileage service and provide a greater margin of safety. Heavy-duty coils

[1] See S.A.E. TRANSACTIONS, August, 1937, pp. 358-369: "Modern Automotive Electrical Equipment," by R. M. Critchfield.

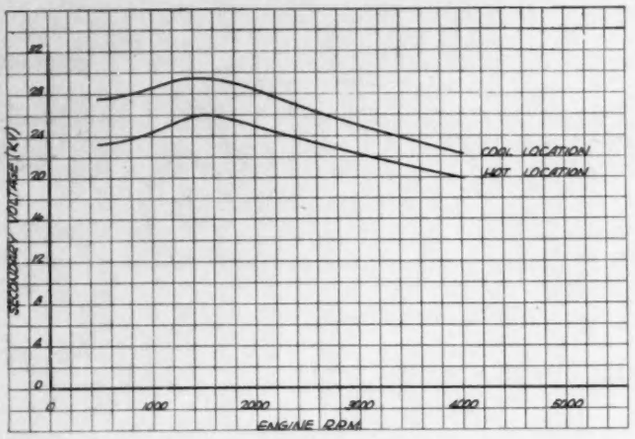


Fig. 12

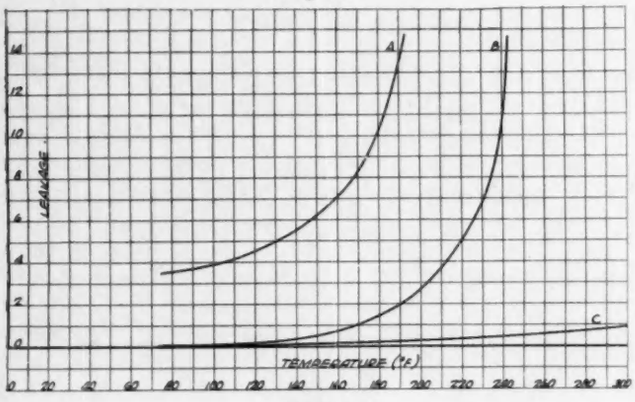


Fig. 13

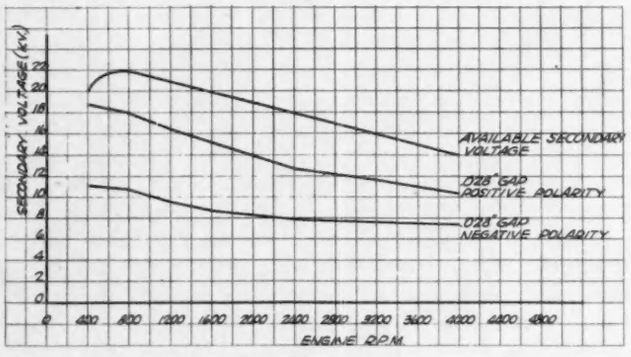


Fig. 15

Fig. 12 – Effect of Temperature on Performance of 12-Volt Ignition Coil

Fig. 13 – Effect of Temperature on Conductance or Leakage of Various Synthetic Plastic Molding Materials

Fig. 15 – Effect of Center Electrode Polarity on Voltage Requirements at Spark-Plug Gap

generally are provided with additional heat-dissipation surface to enable them to operate at lower internal temperatures and thus give not only better performance but also the longer life which lower temperatures permit.

Fig. 11 shows the construction of a standard coil compared with that of a heavy-duty unit. Increased insulation and leakage distances are evident in the latter. Cooling fins also are provided on the heavy-duty unit.

Effect of Temperature and Moisture – In dealing with the various units of the ignition, frequent reference has been made to the harmful effect of temperature and moisture on their performance and life.

Curves on Fig. 12 indicate the loss in performance if a coil is operated in a hot location. It may be seen that, at 2000 r.p.m., the available secondary voltage may be reduced over 10 per cent due to high coil temperature.

Fig. 13 shows the effect of temperature on the conductance or electrical leakage of the insulating materials generally used. Curve *A* for the standard material indicates a noticeable increase in leakage with increase in temperature. Curve *B* for one special compound shows marked improvement of this temperature effect, whereas Curve *C* for a new special material indicates practically no increase in leakage at higher temperatures.

High temperatures also cause difficulty with the proper lubrication of the circuit-breaker cam, resulting in short breaker lever rubbing-block life.

High-temperature operation will reduce the life of practically any insulation, including that of the high-tension cable. One operation reports: "Heat causing rapid deterioration of high-tension insulation is our biggest maintenance problem."

Moisture difficulties generally result from mounting the units so they are subject to road splash or moisture from washing, particularly if occurring while the units are hot. The hot parts are cooled by the water and, due to the resulting contraction of the assembly, tend to draw the moisture inside.

High-temperature operation is due to mounting the unit near the manifold, exhaust pipe, muffler, and so on. If the mounting is otherwise satisfactory, it is better to protect the unit by lagging or insulating the heat-radiating part. If it is also necessary to protect the coil or distributor from moisture conditions, these units can be shielded. In this event, however, it should be borne in mind that the heat-dissipating surface of the coil should be ventilated properly and that ventilation should be arranged for the inside of the distributor cap.

Experience indicates that the internal operating temperature of the coil should not exceed 240 deg. fahr. Since heat is generated internally, it is essential that the coil be so located and shielded that the surrounding air temperature will not exceed 185 deg. fahr.

Dirt is also harmful to the ignition equipment, not only causing bearing difficulties, but providing low-resistance leakage paths over the surface of insulating materials.

Summarizing the foregoing discussion on the various items affecting the application of ignition equipment, the following conclusions can be drawn:

Standard equipment should be used wherever possible due to its lower cost and to the fact that it has been proved thoroughly for use with the engine to which it is applied. Engine vibration may require special construction which otherwise would be unnecessary.

Exposure to high temperatures or moisture may require special insulation.

Certain types of service justify specially constructed equipment in view of the increased factor of safety and freedom from road failures obtained.

The application chart of Fig. 14 attempts to classify different types of service and list the equipment most suitable. Due to all the factors influencing the correct application, such a chart can at best only serve as a rough guide to the proper equipment.

Ignition-System Maintenance Notes – Causes of contact point pitting:

Thorough discussion of this subject would require treatment of highly involved electric-circuit transient phenomena. It can be said generally, however:

That pitting is a transfer of metal from one contact to the other; that, under certain conditions, points will tend to transfer in one direction whereas, with other conditions, the transfer will be in the opposite direction. Satisfactory operation is obtained when circuit constants are adjusted so that

SERVICE	BEARINGS	MOUNTING	CAP & ROTOR MATERIAL	COIL VOLTAGE	COIL TYPE	CONDENSER	H.T. CONNECTIONS
PASSENGER CAR	*	* *	STANDARD	6	STANDARD	2 PAPER	SLIP
POLICE CAR	BRONZE	* *	SPECIAL	6	STANDARD	3 PAPER	SCREW
TAXICAB	BRONZE	* *	SPECIAL	6	STANDARD	3 PAPER	SCREW
LIGHT TRUCKING	C.I.	S.A.E	STANDARD	6	STANDARD	2 PAPER	SLIP
HEAVY DUTY TRUCKING	BRONZE OR BALL	S.A.E	STANDARD	6 OR 12	STANDARD OR HEAVY DUTY	3 PAPER HEAVY DUTY	SCREW
SMALL CITY BUS	BRONZE	S.A.E	SPECIAL	12	HEAVY DUTY	3 PAPER HEAVY DUTY	SCREW
LARGE CITY BUS	BALL &, OR BRONZE	S.A.E	SPECIAL	12	HEAVY DUTY	3 PAPER HEAVY DUTY	SCREW
INTERURBAN BUS	BALL &, OR BRONZE	S.A.E	SPECIAL	12	HEAVY DUTY	3 PAPER HEAVY DUTY	SCREW

NOTE * PLAIN C.I. ON SHORT SHAFTS OR COUPLED LONG SHAFTS AND BRONZE OR DUREX ON ONE PIECE LONG SHAFT
** S.A.E. PREFERRED UNLESS QUANTITY WARRANTS SPECIAL TOOLS

Fig. 14 – Ignition Application Chart

these tendencies balance each other. In case of pitting difficulties, various units of equipment can be modified to obtain the necessary correction.

For transfer of metal to the positive (pitting of the negative) contact, the following changes tend to be corrective:

1. Increased condenser capacity.
2. Shorter condenser lead.
3. Separation of low- and high-tension coil distributor leads.
4. Reduction of primary current by use of resistance in primary circuit.
5. Decrease in the amount of engine idling time.

For transfer of metal to the negative (pitting of the positive) contact, reversal of the foregoing suggested changes will tend to be corrective.

Causes of point oxidizing:

Contacts may develop a high resistance coating because of excessive arcing due either to incorrect electrical constants or to the presence of foreign materials on their surfaces.

In cases of point oxidation, incorrect electrical constants usually permit too much primary current. Coils with too low primary resistance, such as most so-called high-capacity service types, require too much primary current for the contacts to handle.

High battery voltage due to various battery conditions, unregulated generators, or poor connections in regulator to generator circuits, may raise coil voltage sufficiently so that increased coil current will oxidize the contacts.

A combination of cold battery with cold coil windings as obtained in sub-zero weather operation may allow too much primary current through the contacts.

High resistance or loose connections in the condenser circuit also will cause oxidized contacts.

Condensers in which leads do not make good contact with the foil also will cause this trouble, and yet test O. K. on most condenser testers. Measurement of series impedance or so-called "damping" is the only way of detecting this trouble.

Oil and crankcase vapors accelerate contact oxidation and wear. Oil seals in the distributor bearing and adequate cap ventilation correct this condition.

Iron, copper, brass, and various other particles lodged on the contact surface will cause trouble.

Cap Failures – As pointed out in discussing applications, heat and moisture account for most distributor-cap failures. If shielded, ventilation still should be provided.

Coil Failures – Here again heat and moisture, and particularly the latter, account for most failures. Again, it should be emphasized that, if the coil is shielded, this shielding should be confined to the coil top so that the case surface will be free to dissipate heat generated within the coil. The coil should be mounted in a vertical position with the top up as it best withstands heat and moisture in this position.

Under some conditions unnecessarily high voltage requirements are imposed on the coil. Fig. 15 illustrates a condition resulting from incorrect polarity of the spark-plug center electrode which, under certain conditions, increases the voltage necessary to fire the gap over 50 per cent. Even though the coil will produce this higher voltage, the insulation factor of safety will be reduced and contact points will be worked harder.

This condition can be corrected by providing a coil connected for the proper polarity.

Ignition coils should be mounted with the mounting bracket or case directly grounded to the frame or other well-grounded part.

There is no doubt that coils and condensers can be made waterproof and fairly resistant to high-temperature operation. However, increased cost would be involved which, together with the fact that performance would be no better than that obtained with the standard-equipment model located so as to be protected from these injurious conditions, emphasizes the advisability of giving more attention to the proper application of these units.

It is important that the correct type of tester be used in checking the coil condition.

A satisfactory service ignition-coil tester must be accurate, sensitive, easy to operate, and easy to maintain.

Testers which have been brought to our attention can be divided roughly into three groups:

The first type uses a variable spark gap as the measuring instrument. An attempt is made to establish a gap length for each coil at different speeds. This type is unsatisfactory because the spark gap fails to hold its calibration. The voltage at which the gap breaks down changes with operation due to the change in the shape of the points. Another fault is that the insulation of the gap absorbs moisture and thus provides a leakage path sufficiently great to lower materially the voltage which the coil can produce.

The second type of tester uses a milliammeter in series with a fixed spark gap. This type is an improvement over the first type, mainly because of the better construction. The spark gap is still sensitive to the same troubles and, in addition, in order to obtain a steady meter reading, it is necessary to operate the coil on a lower secondary voltage than will be encountered during actual service. Consequently, a coil that had a breakdown which occurred above the voltage at which the gap operates will check O. K., and yet will cause missing of the engine at low speeds or upon acceleration.

The third type tester does not measure the secondary voltage, but makes a measure of the primary voltage. It operates the coil with the secondary terminal disconnected so that the full secondary voltage is applied across the insulation of the coil and no extra variable losses due to tester insulations are added to the system. In this way, it measures only the losses in the coil itself. This tester is much more sensitive than either of the other types and will detect partial failures which the other two types will not reveal. For this reason it is a

better tester for maintenance organizations because defects can be found before they have progressed to the point of causing trouble, and such defective coils can be removed before they cause a failure on the road. Failures such as leaky insulation, wet secondaries, one turn shorted in the primary and ten turns shorted in the secondary, can be detected easily. This tester is not as sensitive to variations in coil temperature as are the other types, and a small defect such as one turn shorted in primary makes a much greater change in the reading than is made by having the coil hot, whereas the other two types of testers will not even detect one shorted turn, but show a large change between hot and cold coils. Further proof of the superiority of the third type is the observation that the use of the first and second type of testers did not decrease the percentage of good coils returned to our plant for replacement. Whereas, use of the latter type has decreased the percentage of good coils returned to one-third of the former number. This tester is calibrated easily and is easy to operate.

Condenser Failures – Heat and moisture account for most failures of this unit also. In checking questionable units a suitable tester should be provided.

Since the condenser is a very essential part of the ignition equipment, it is necessary to provide a tester which will determine if the condenser is up to standard in regard to the various characteristics which it must maintain. First, the condenser must be able to withstand the voltage applied to it by the ignition coil. Since this voltage is usually not more than 200 volts and rarely more than 350 volts, a peak voltage of 500 volts is considered a suitable value for a break-down test. This voltage is high enough to show up weakness and low enough not to cause damage to a good unit.

Second, the condenser should have the desired capacity. It has been our practice to adjust the capacity of the condenser to suit the operating conditions in order to obtain the best breaker contact condition. We use condensers with capacities from 0.15 mfd. to 0.50 mfd. The tester should be capable of measuring the capacity with an accuracy within 10 per cent.

Third, it is essential that the tester detect if extra resistance has developed in the condenser, such as corroded connections between the terminal and lead or between the leads and the foils. This trouble is one which is difficult to find since the connection may be good enough to allow the condenser to take a charge slowly, but may interfere when the condenser operates at the speed required in the ignition circuit. This extra resistance is detected most easily by applying a high-frequency voltage to the condenser; the higher the frequency, the more easily the extra resistance can be detected. This characteristic has been called "damping" by several tester manufacturers. It is desirable to shake the condenser while making this test.

The fourth characteristic is insulation resistance, which is a measure of the leakage through the insulation. Leaky condensers have but little effect on ignition performance. Leakage is of importance because a high-leakage condenser is more apt to break down. This test, then, is more of an insurance against future trouble than a means of locating prevailing trouble. The insulation resistance decreases rapidly with increased temperature so that a hot condenser is more likely to test low than a cold one. Most testers are set to test cold condensers, and good condensers may test low if tested hot.

A study of maintenance information received from various types of fleet operations indicates that:

Distributors should be lubricated at least every 2000 miles.

Distributors are removed for inspection and checking of contact condition and point opening at from 2000-mile to 25,000-mile intervals, the average being around 10,000 miles. The distributors are removed, overhauled, and all worn parts replaced at from 45,000 to 150,000 mile intervals, the average being around 85,000 miles.

The Starter System

The starter system consists of the cranking motor, the motor controls, the battery, and the engine, together with the connecting cables.

Items to be considered in making an application of a starter system to an engine are:

The Engine Size – The torque requirements of the engine vary with its size. It has been customary to consider the engine displacement in determining the starter size. However, a direct relationship has not been found to exist between cranking torque and displacement.

Cranking motor requirements are usually most severe under cold-weather starting conditions, due mainly to the increased engine friction resulting from the higher viscosity of the lubricating oil at low temperatures. From this condition it would appear that there should be a relation between cranking torque and the cylinder area wall wiped by the piston, as the greatest amount of engine friction at low temperatures exists at this point. Torque requirements from a number of cold cranking tests on various size engines were plotted against cylinder area wiped, and it was found that a fairly consistent relation existed. Fig. 16 shows this relation plotted at three different oil viscosities. It is believed that the values indicated will apply within fairly close limits to any four-cycle gasoline engine after it has been broken in.

Cranking torque also varies considerably with speed, especially below the knee of the compression-pressure speed curve. The values as shown on the chart are for approximately 35 r.p.m., which represents average cold-weather cranking practice. Examination of the lines on the chart showing torque requirements for different oil viscosities will indicate the influence of the latter on cranking requirements. For example, with an engine having 400 sq. in. cylinder area, a torque of 160 ft-lb. will be required to crank at 0 deg. fahr. with 20,000-sec. oil whereas, with 45,000-sec. oil, 220 ft-lb. will be required, or an increase of over 35 per cent.[2] It is, therefore, important in considering cold-cranking requirements to provide for the use of the lowest practical oil viscosity.

Engine Design as It Affects Ease of Starting – Carburetion, manifolding, choking, spark-plug location, all affect ease of starting. Poorly constructed hand chokes or automatic chokes which are faulty in operation impose unnecessarily severe conditions on the cranking motor due to the prolonged cranking periods which oftentimes result. Some engines are very difficult to start, particularly at low temperatures. To illustrate, cold-starting tests on two engines of different manufacture each with about 400 cu. in. displacement indicated widely divergent cranking requirements. One engine started consistently at 0 deg. fahr. when cranked around 20 r.p.m.; the other engine would not start when cranked at 50 r.p.m. Not only a larger starter, but also a larger battery will be required for the hard-to-start engine, entailing not only higher initial cost, but increased dead weight and maintenance costs.

The engine design should provide a rigid mounting for the starter which will provide and retain the proper pitch-line clearance between the pinion and ring-gear. The design should provide for the proper reduction between the motor and engine so that the peak motor output will be available under the most severe cranking requirements. Examination of the cranking motor curves in Fig. 21 indicates the characteristic speed output performance of these units. The reduc-

[2] 20,000 sec. representing the low-viscosity limit, and 45,000 sec. the high-viscosity limit, of S.A.E. 20W oil.

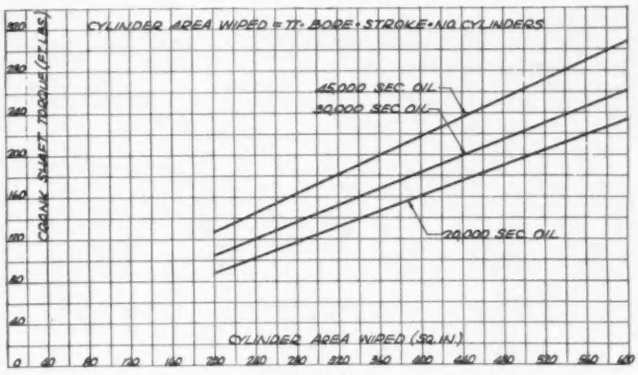


Fig. 16

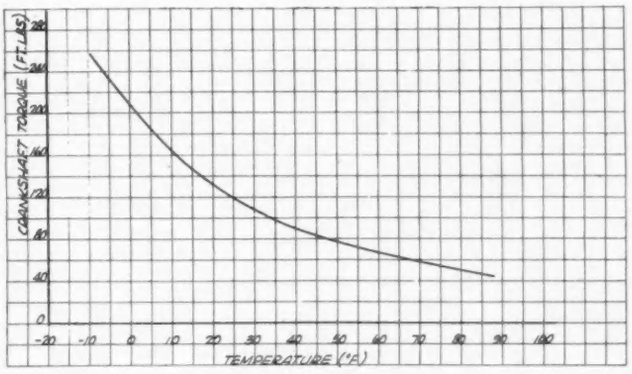


Fig. 17

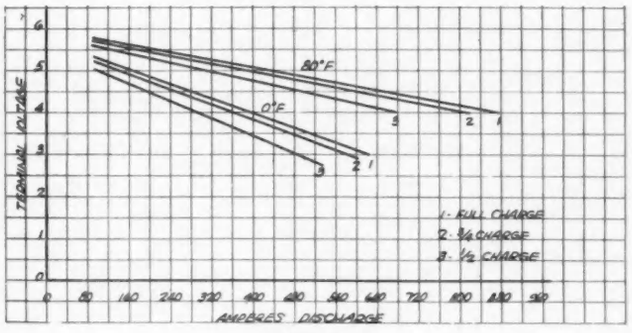


Fig. 18

Fig. 16 – Relation between Engine Size and Cranking Torque Required at 35 R.P.M. and 0 Deg. Fahr.

Fig. 17 – Relation between Temperature and Cranking Requirements for a 426 Cu. In. Engine Using 20W Oil

Fig. 18 – 5-Sec. Voltages – 115 Amp-Hr. Capacity Battery – 4.3 Min. at 300 Amp. at 0 Deg. Fahr.

tion should be such that the maximum output will occur at a speed equivalent to the engine cold cranking speed.

Service Requirements – In addition to limitations imposed by engine size and design on the cranking requirements, the type of service provides additional limitations. It is necessary that passenger cars and certain other types of vehicles be capable of starting in sub-zero temperatures. This requirement is generally not necessary in the case of most fleet operations.

Cold starting requires a higher capacity cranking system. The engine friction increases due to increase in oil viscosity, and hence increases with decrease in temperature.

Fig. 17 shows the relation between cranking torque and temperature for a 426 cu. in. engine. From the curve it is seen that 50 ft-lb. torque is required to crank at 80 deg. fahr., whereas over 200 ft-lb. is required at 0 deg. fahr.

The storage battery which provides the energy for the cranking motor is much less efficient at lower temperatures. Fig. 18 shows the relation between cranking current and battery terminal volts with a battery of given capacity at 80 deg. fahr. and at 0 deg. fahr. It will be observed that, with a 300-amp. draw which represents average cold-weather cranking requirements for medium-size engines, the voltage at the terminals is approximately 5.3 at 80 deg. fahr. and drops to 4.4 at 0 deg. fahr. with the battery fully charged. Since the energy available for the cranking motor is equal to the product of amperes times volts, the loss in energy at low temperatures is evident. The lower voltages obtained with a battery only partly charged emphasize the desirability of providing sufficient generator capacity so that the battery will be maintained in a well-charged condition.

As indicated by data in Fig. 16, changing the grade of oil for winter use has helped the cold cranking problem considerably. Changing to lower viscosity oils for cold-weather operation not only helps reduce the cranking requirements but, due to reduction in engine friction horsepower, lowers the speed at which the engine will fire and run.

Fig. 19 illustrates this point for a particular engine. Friction horsepower values at two different oil viscosities and indicated horsepower are plotted against engine r.p.m. Assuming that an initial fire brings the engine up to 150 r.p.m. with the 50,000-sec. oil, it is evident from the curve that, since the friction horsepower is greater than the indicated horsepower, the engine will not continue to run. With the 7500-sec. oil, the friction horsepower is reduced to the point where it is less than the indicated horsepower and the engine will continue to run after the first explosion.

The type of service also may affect the cranking motor construction. Although a certain design may be perfectly satisfactory in all respects for normal service, some operations, especially delivery trucks, taxicabs in larger cities, and so on, require starting the engine hundreds of times per day. This practice generally results from an effort to save fuel by stopping the engine during what otherwise would be idling periods. Such motors probably will require special bearing and lubrication arrangements.

Battery Size – Data presented indicate that, with larger engines, more torque is required to crank; also that, for cold-weather cranking, more torque is required. Since more torque means more energy out of the cranking motor and since the motor depends upon the battery for this energy, large batteries should be supplied with larger engines, especially where cold starting is involved. Fig. 20 illustrates the effect of battery capacity on cranking speed. The curves in the lower-left corner show the effect of increasing the cranking ability of a small 4½-in. 6-volt motor by increasing the battery capacity from a 2.3 min. cold rating to 5.5 min. With a torque at crankshaft of 75 ft-lb. (15:1 reduction), the cranking speed will be increased from 42 to 58 r.p.m. Similarly, with a large 5-in. 12-volt motor, changing from a battery with a 2.5-min. to one with 7.0-min. cold rating will increase the speed from 38 to 53 r.p.m. with a crankshaft torque of 250 ft-lb. and a 13:1 reduction. Improvement in cranking speeds with larger batteries is evident. In most types of service the battery capacity is determined by cranking motor requirements; however, in bus service where a heavy lighting load is carried, larger batteries generally are supplied than would be required for cranking.

Type of Cranking Motor Pinion Engagement – Three types of pinion engagements are in use at the present time. The Bendix drive and the Dyer drive both can be shifted against the flywheel ring-gear with the engine in operation without damage to the pinion or ring-gear. Both of these types also provide automatic disengagement when the engine starts. The fact that the Dyer-type operation is such that the starting motor cannot be energized until the pinion is in engagement with the ring-gear makes it suitable for heavy-duty service as

no extra shock is thrown upon the gear teeth. The Dyer type is arranged so that, if the teeth butt upon attempting to engage, the pinion will be rotated into an engaging position. Due to the principle of this action it is necessary that the flywheel ring-gear be chamfered from the opposite side of the tooth to that required for the Bendix drive, or that a compromise round-nosed tooth be provided which can be used for either type. The overrunning-clutch type cannot be shifted into the engaging position with the engine running without damage to the teeth, and, therefore, is not suitable for rear-engine applications where it is sometimes difficult for the operator to determine whether or not the engine is operating.

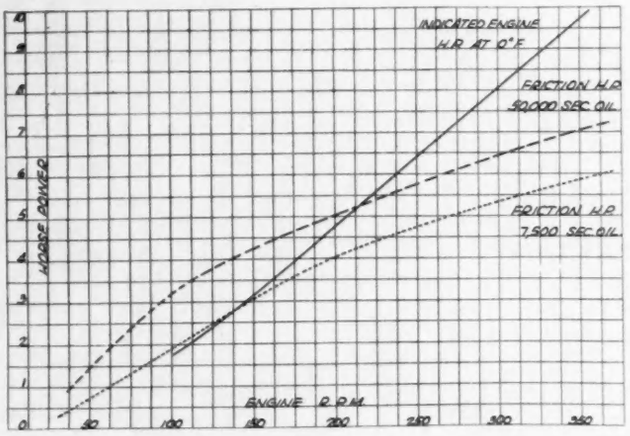


Fig. 19

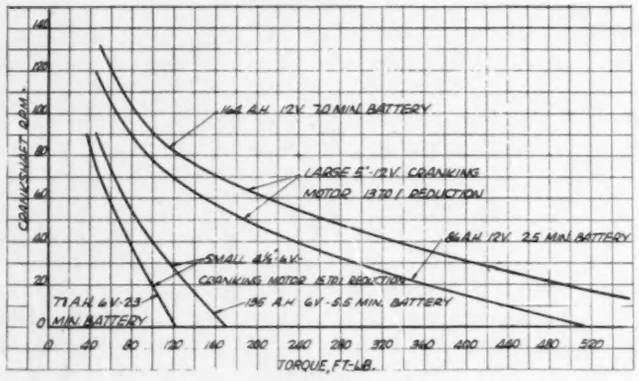


Fig. 20

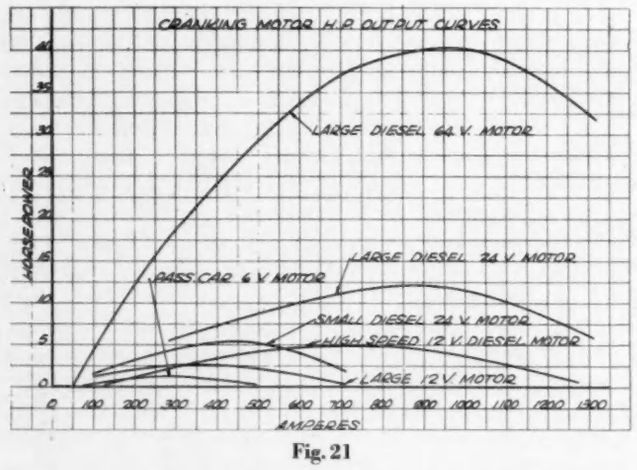


Fig. 21

Fig. 19 – Relation between Indicated Engine Horsepower and Friction Horsepower with Different Viscosity Oils at 0 Deg. Fahr.

Fig. 20 – Effect of Battery Capacity on Cranking Motor Speed

Fig. 21 – Cranking Motor Horsepower-Output Curves

It does possess the advantage, however, of staying in mesh until manually released so that it is possible to crank an engine through periods of weak explosions without having the pinion kick out.

Diesel-Engine Cranking Requirements

Many of the items just covered on gasoline-engine cranking requirements also apply to the Diesel, or compression-ignition type. A Diesel engine, however, requires more cranking-motor capacity for a given size, due to:

Higher compression ratio used, which in itself requires higher torque, but which also requires heavier construction to withstand the higher pressures, thus appreciably increasing the engine friction. The main fact, however, contributing to increased cranking capacity is the self-ignition feature of this engine which requires that the engine be cranked over fast enough to enable the compression pressure to cause sufficiently high temperature to ignite the fuel. Cranking at lower speeds not only reduces the compression pressure, but permits dissipation of the heat generated into the cylinder walls at a faster rate than that at which it is generated. As an example, a 250 cu. in. Diesel engine, in addition to requiring a higher cranking torque, also needs a cranking speed of approximately 200 r.p.m. to start at freezing temperatures as against 25 r.p.m. required for the gasoline engine.

Due to the fact that the ratio between displacement and cooling cylinder-wall area becomes greater with larger displacement cylinders thus increasing the ratio of heat generation to its dissipation to the cylinder walls, the cranking speed required for self-ignition fortunately becomes less as the displacement per cylinder increases. Thus, with a large engine having several hundred cubic inches per cylinder, starting at freezing temperatures can be accomplished with a cranking speed of around 40 r.p.m.

Without help from preheated air, use of glow plugs, and so on, it is necessary that the speed be high enough to get above the knee of the speed-compression pressure curve to insure self-ignition, a temperature of around 575 deg. fahr. due to compression pressure being necessary. For a given engine design, it is impossible at any reasonable speed to get compression temperatures high enough to fire with extremely low ambient temperatures. Thus, one engine at 0 deg. fahr. cranked up to 200 r.p.m. with the compression temperature leveling off at 525 deg. fahr., or a value too low to fire. Glow plugs or preheating of the intake air is necessary in order to start under such conditions.

Due to the high-speed, high-power cranking requirements of the compression-ignition engine, starting motors generally are designed for higher voltage operation, 12 volts being required for the smaller types, 24 volts for the medium size, and 64 volts for some of the larger stationary types. Fig. 21 shows the horsepower outputs of various cranking motors. The small curve in the lower left-hand corner indicates the performance of the average passenger-car starting motor which has a maximum output of about 1 hp. The next higher curve gives the performance of the heavy-duty truck and bus 12-volt cranking motor which has a maximum output of approximately 3 hp. The other curves are for 12-, 24-, and 64-volt cranking motors used with Diesel engines, the largest motor having a maximum output of about 40 hp.

These high-capacity cranking motors require batteries capable of delivering high current values without excessive internal drop. To accomplish this purpose without excessive weight and cost, batteries have been provided with a large number of medium-thickness plates and with high-capacity connectors and other current-carrying parts.

ENGINE TYPE	DISPLACEMENT CU IN	MOTOR H.P.	VOLTAGE	FLYWHEEL REDUCTION	TYPE * PINION	TOOTH SIZE	BATTERY MIN. AT 300A AT 0°F	A H CAPACITY
GASOLINE	UP TO 250	.8	6	14-17 : 1	B-C	10-12 PITCH	3.0 - 4.0	90 - 115
GASOLINE	250 - 325	1.0	6	14-17 : 1	B-C	10-12 PITCH	4.0 - 4.75	115 - 125
GASOLINE	325 - 400	1.0	6	21 - 28 : 1	B-C	8-10 PITCH	4.75 - 5.5	125 - 140
GASOLINE	400 - 650	2.5	12	12 - 15 : 1	B-C-D	8-10 PITCH	3.0 - 4.5	95 - 110
DIESEL	UP TO 250	4.5	12-24	8 - 10 : 1	B-C-D	6-8 PITCH	9 - 15 3.5 - 5.0	12V 200-300 24V 100-120
DIESEL	250 - 400	5.0	24	8 - 10 : 1	B-D	6-8 PITCH	4.0 - 5.5	110 - 140
DIESEL	400 - 650	11.0	24	8 - 10 : 1	B-D	6-8 PITCH	5.5 - 10.0	140 - 200
DIESEL	650 - 800	15.0	32	8 - 12 : 1	B-D	6-8 PITCH	6.0 - 10.0	170 - 200
DIESEL	4000	30	64	15-20 : 1	D	6-8 PITCH	10.0	200

NOTE * PINION TYPE B = BENDIX, C = OVERRUNNING CLUTCH, D = DYER

Fig. 22 – Cranking Motor Application Chart

Starting Motor Controls

Where cost is a factor and the units can be made accessible, foot-operated switches provide the simplest means of starter control. Where remote control is necessary, pushbutton-operated magnetic switches can be used with the Bendix drive and pushbutton-operated shifting solenoid switches can be used with the overrunning-clutch and Dyer-type drives. For automatic operation, the Startix unit made by Bendix is also available as well as the accelerator and clutch pedal starting supplied in one form or another by Delco-Remy and Auto-Lite.

The necessity for using 24 volts for some Diesel-engine cranking motor requirements already has been covered. It has been found desirable in some cases to maintain 12-volt lighting, accessory, and generator circuits, due to the general availability of 12-volt bulbs and accessories and the increased mechanical efficiency of the 12-volt bulb. To accomplish this purpose, manually and remote-control operated switches are available which permit the application of 24 volts to the motor when cranking, while allowing the remainder of the system to operate at 12 volts.

Wiring and Connections

Since the energy available at the cranking motor is proportional to the voltage at its terminals, the cranking ability with a given capacity battery will vary with the drop in the cables and connections; therefore, it is good practice to reduce this drop to a minimum if full advantage is to be taken of the cranking motor and battery specified.

Fig. 22 lists engines by groups and attempts to classify starter requirements, pinion to ring-gear reductions, pinion tooth sizes, voltages, and battery capacities, and should serve as a rough guide in making applications.

Maintenance Notes

Unnecessary maintenance of cranking motors can be avoided if they are mounted so as to be protected from road splash, mud, fuel, oil drippings, and so on.

In order to mount in the available space and keep weight and cost at a minimum, cranking motors are designed to get the maximum output from the material used. They are essentially intermittent-service units and, due to internal losses, heat up very rapidly while in use. This teaches:

First, that the cranking motor should not be used over prolonged periods. If an engine will not start, prolonged cranking very seldom helps and, in some cases, delays conditions necessary for starting due to running down the battery, flooding the cylinders, and so on. In cases of vapor-lock, prolonged cranking is of little value. As a general rule, a cranking motor should never be used over 30 sec. without determining what is the trouble with the engine.

Second, the hotter the parts and air surrounding the cranking motor, the less time it will take to reach a critical temperature while cranking; therefore, like everything else electrical, more efficient and trouble-free operation is secured if the motor is kept as cool as possible.

It already has been pointed out that increased cranking performance could be obtained with a motor of given size by using a larger battery.

It also was pointed out that higher capacity batteries than are needed for cranking are used sometimes for certain types of operations. With installations of this type, it is all the more necessary to avoid abusing the cranking motor as the battery provides sufficient excess of energy to permit overloading and overheating the motor under prolonged cranking conditions.

One source of expense in cranking-motor maintenance results from damage due to backfires when starting. Correct timing, which is important in obtaining proper performance and economy, is also important at low speeds to avoid backfiring.

Pinions should be kept lubricated and free from dirt, gravel, clutch dust and other material tending to gum up the threads, splines, and so on. The pitch-line clearance should be maintained at the proper value, and the clearance in the direction of the shift between pinion and ring-gear in disengaged position should be adjusted to the value recommended for the particular type of shift being used.

Ring-gears should be made of alloy steel and heat-treated so that, after assembly on the flywheel, the Rockwell C hardness will run from 42 to 50.

Fleet maintenance data indicate that cranking motors are checked and inspected at intervals ranging from 3000 to 25,000 miles and that they are removed, overhauled, brushes and bearings replaced if necessary, and the commutator turned, at intervals from 25,000 to 150,000 miles.

Conclusion

An attempt has been made in the foregoing to point out some of the factors of electrical equipment application which affect maintenance.

From the writer's experience, the main items of maintenance at the present time result from:

Improper drive ratio, for generator applications.

Heat, moisture, and engine vibration, for the ignition applications.

Application of under-sized cranking motors and batteries, and misapplication of pinion engagement devices, for starter applications.

It is hoped that some of the information just presented will be of assistance in this connection.

Reference has been made to the necessity of supplying special performance and, in some cases, specially constructed electrical units to insure trouble-free service. At times, operators are inclined to blame vehicle manufacturers for supplying unsuitable equipment; yet they will purchase it under competitive-bidding conditions without knowing what equipment will be furnished by the successful bidder. The purchaser should be careful to specify the performance required, but should avoid trick specifications and unnecessary constructions that will result only in higher prices without a corresponding increase in performance.

If standard equipment can be used, it is very desirable from a cost standpoint. However, if special or oversize equipment is needed, it is generally less expensive to have it included at the time of purchase of the vehicle.

Correlation of Road and Laboratory Octane Numbers

Report from the Cooperative Fuel Research Committee

Presented by J. R. Sabina

Chairman Correlation Group, Motor Fuels Division

NO change in the method for correlating road and laboratory ratings is recommended, this report concludes. The failure to achieve better correlation, it explains, is not due primarily to shortcomings of the data, but to the failure to take cognizance of all the factors influencing car ratings. Furthermore, it reports, no material improvement can be expected from accumulation of additional laboratory data involving only speed, temperature, and spark advance.

At the November, 1936, meeting of the Cooperative Fuel Research Committee the study of the correlation between laboratory knock ratings and road ratings in the then-current automobiles was authorized. The outcome of a preliminary study, based on information submitted by individual companies in 1934, 1935, and 1936 cars, indicated the desirability of collecting cooperatively road and laboratory data under controlled conditions for a more accurate evaluation of the correlation problem. The results of the work of the Road-Test Analyzing Group and the Laboratory Group already have been presented. This report presents the findings of the Correlation Group.

Evaluation of the various laboratory methods, from the standpoint of correlation, is based on:

1. The deviations of the average laboratory ratings from the road ratings.

2. The deviations of the average laboratory ratings from a 45-deg. line displaced from the origin to give the lowest average deviation.

AT the November, 1936, meeting of the Cooperative Fuel Research Committee, action was taken authorizing the study of the correlation between laboratory knock ratings and road ratings in the then-current automobiles. A preliminary study was made based on information submitted by individual companies on 1934, 1935, and 1936 cars. The outcome of that work indicated the desirability of collecting cooperatively road and laboratory data under controlled conditions for a more accurate evaluation of the correlation problem.

The results of that work have been presented in the reports of the Road-Test Analyzing Group,[1] September, 1937, and the Laboratory Procedure Group, October, 1937,[2] respectively. This report presents the findings of the Correlation Group.[3]

An analysis of the report submitted by the I.P.T. Knock-Rating Panel based on British cars and fuels leads to the conclusion that the same factors seem to influence both the British and American data, and in about the same degree, there being no tendency for the two sets of data to follow different relationships. It is probably safe to assume, therefore, that a testing method which gives improved correlation in this country will give improved correlation with British fuels and cars.

Participation

Organizations which participated in the Correlation Group program are listed below:

Ethyl Gasoline Corp.
General Motors Corp.
Gulf Oil Corp.
Gulf Research and Development Co.
National Bureau of Standards.
Phillips Petroleum Corp.
Sinclair Refining Co.
Skelly Oil Co.
Socony-Vacuum Oil Co., Inc.
Standard Oil Co. of Calif.

This paper and the three that follow comprise four of the five papers presented at a Detonation Symposium held at the 1938 Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 16, 1938. The fifth: "Knock-Testing – In the Laboratory and in Service," by Graham Edgar, was published in the September 1938 issue of the S.A.E. JOURNAL, *pages 7 to 12 and 17 to 20.*

[1] See S.A.E. TRANSACTIONS, June, 1938, Vol. 42, pp. 244-252; or Proceedings of the American Petroleum Institute, Section III, Vol. 18, 1937, pp. 98-112: "1937 Road Knock Tests – Report from Cooperative Fuel Research Committee," presented by T. A. Boyd.

[2] See S.A.E. TRANSACTIONS, February, 1938, Vol. 42, pp. 63-72; or Proceedings of the American Petroleum Institute, Section III, Vol. 18, 1937, pp. 112-125: "Effect of Test Conditions on Fuel Rating – Report from Cooperative Fuel Research Committee," presented by A. E. Becker.

[3] Personnel of the Correlation Group: J. R. Sabina, chairman; C. B. Veal, secretary; W. G. Ainsley, D. P. Barnard, H. J. Gibson, A. E. Becker, J. M. Campbell, L. E. Heil, W. M. Holaday, J. B. Macauley, Jr., J. R. MacGregor, Neil MacCoull, A. W. Pope, H. M. Trimble.

Standard Oil Co. (Ind.)
Standard Oil Development Co.
Sun Oil Co.
The Atlantic Refining Co.
The Pure Oil Co.
The Texas Co.
Tide Water Associated Oil Co.
Union Oil Co. of Calif.
Universal Oil Products

A similar group organized by the I.P.T. Knock-Rating Panel submitted a report containing data obtained by six participating companies.

Object

The object of this study was to coordinate the results obtained by the Road and Laboratory Groups and to develop, if possible, a better method for correlating road and laboratory ratings.

Procedure

The first step undertaken by the Group was a critical examination of the road results. This analysis revealed that a small percentage of the road ratings was reported as having been made in cars with spark advances considerably in excess of the maximum provided by the octane selectors or similar scales on the distributors. If the ignition timing at the speed of maximum knock were advanced by the same increment as that indicated by the static or low-speed settings, the conclusion was that the cars in which such ratings were made were not in proper condition or adjustment for knock-rating purposes. It is possible, however, at least in some cases, that an abnormal advance under static or low-speed conditions did not result in an abnormal advance in the speed range of maximum knock but, as this factor was unknown, the selection was made on the basis of the static settings.

In the absence of definite recommendations by the motor-car manufacturers covering the maximum permissible spark advance, correlation was based arbitrarily on the average of road ratings at spark advances not in excess of 10 deg. over the manufacturers' specifications for initial spark timing. The effect of this limitation on the number of cars used, and on the road averages is shown in Table 1. These modified ratings have been used throughout this report and are shown for all fuels in the second column of Table 2.

Study of the various methods investigated by the Laboratory Group indicated that improved correlation might be obtained by using combinations involving 150 and 200 deg. fahr. intake mixture temperatures at speeds of 600 and 900 r.p.m., and 17 deg. fixed spark advance.

In view of the fact that two of these methods had been run on relatively few fuels and that the other two combinations had not been tried, members of the Correlation and Exchange Groups were asked to rate the summer and winter fuels by the four methods. The combinations of variables chosen were as follows:

Method No.	R.P.M.	Intake Mixture Temperature, deg. fahr.	Fixed Spark Advance, deg.	Conditions for Motor-Method Knock Intensity: Corr. Compression Ratio	Per Cent C-10 in A-5
12	900	150	17	6.1	61.5
32	900	200	17	5.8	62.5
30	600	150	17	5.4	56.0
27	600	200	17	5.25	59.0

The use of the predetermined compression ratios and reference-fuel blends indicated in the table resulted in Motor Method knock intensity. The spark linkage was disconnected and the spark advance maintained constant at 17 deg.

The effect of spark advance on ratings was investigated over a wide range of conditions by the Laboratory Procedure

Table 1 – Effect of Elimination of Ratings Made with High Spark Advances on the Average Road Ratings of the Preferred 1937 C.F.R. Fuels

	ALL CARS ALL SPARK ADVANCES *		ALL CARS WITH SPARK ADVANCE 10° OR LESS		
Fuel	Rating	No. Cars	Rating	No. Cars	Change From All Car Rating
WA-1	67.8	58	67.6	51	-0.2
WB-2	68.8	54	68.7	46	-0.1
WB-3	77.6	47	77.4	38	-0.2
WB-4	74.3	39	74.4	35	+0.1
WB-5	74.4	46	74.1	41	+0.3
WD-2	69.1	41	69.2	34	+0.1
WE-2	71.2	51	71.4	42	+0.2
WF-2	70.4	54	70.4	46	0.0
WG-1	73.9	55	73.7	46	-0.2
WH-1	79.0	39	79.1	33	+0.1
AVG.					+0.01
SA-1	67.5	55	67.5	48	0.0
SB-2	69.2	47	69.1	40	-0.1
SB-3	77.8	37	77.5	24	-0.3
SB-4	70.2	38	70.1	34	-0.1
SB-5	70.6	45	70.2	39	-0.4
SE-2	70.7	51	70.5	43	-0.2
SF-1	72.6	47	72.6	37	0.0
SF-2	70.3	53	70.2	45	-0.1
SH-1	73.9	46	73.7	38	-0.2
AVG.					-0.14

* The ratings obtained in one car in Group "D" were omitted as no spark advance was reported.

Table 2 – Average Ratings – Octane Numbers*

Fuel	Road	Motor	Ave. Res.	Res. + Motor / 2	32	12	30	27	L-3
xWA-1	67.6	69.1	68.7	68.9	69.8	69.2	68.9	69.3	70.6
B-1	65.5	66.6	67.9	67.2	67.9	68.6	69.4	69.6	66.9
x B-2	68.7	68.5	70.9	69.4	70.2	70.6	71.4	71.9	70.1
x B-3	77.4	77.9	79.8	78.8	79.5	79.9	80.7	80.6	79.4
x B-4	74.4	72.5	75.5	74.0	75.3	76.1	77.3	77.1	73.8
x B-5	74.1	71.0	74.4	72.7	73.2	74.5	76.2	75.7	72.9
C-1	71.1	71.2	71.8	71.5	71.7	72.0	72.2	72.0	71.8
C-2	69.1	71.5	70.3	70.9	71.3	70.9	70.1	70.8	72.5
x D-2	69.2	69.1	71.2	70.2	71.1	71.2	72.0	72.8	70.7
E-1	69.4	68.6	70.6	69.6	72.0	73.2	74.4	74.7	69.5
x E-2	71.4	70.5	73.3	71.9	72.9	73.6	74.7	74.8	72.5
F-1	73.1	71.2	78.0	74.6	75.6	75.9	77.6	78.7	74.1
x F-2	70.4	68.9	71.0	70.0	70.2	70.3	72.6	71.4	70.7
x G-1	73.7	69.4	77.4	73.4	74.2	75.1	79.2	78.2	71.7
x H-1	79.1	76.1	81.8	79.0	79.5	80.4	82.9	82.1	77.8
H-2	81.2	79.7	85.7	82.7	82.6	84.0	86.0	85.4	82.0
I-1	69.7	70.9	73.1	72.0	74.0	74.3	76.6	76.0	73.3
I-2	68.2	70.6	69.7	70.2	71.0	70.7	71.3	70.9	71.4
xSA-1	67.5	67.9	68.5	68.2	68.4	67.9	67.6	68.5	69.9
B-1	63.7	62.4	65.7	64.0	64.1	64.0	63.8	64.7	64.5
x B-2	69.1	69.4	72.6	71.0	72.1	72.2	72.4	72.9	72.1
x B-3	77.5	77.4	80.8	79.1	79.4	79.7	80.4	80.3	79.9
x B-4	70.1	69.8	74.3	72.0	73.0	73.7	74.5	74.2	71.6
x B-5	70.2	67.0	72.9	70.0	70.1	70.7	71.8	71.0	69.6
C-1	73.2	72.0	73.3	72.6	72.7	72.8	73.0	73.0	73.2
C-2	70.4	72.1	72.0	72.0	72.2	71.8	70.8	71.4	73.7
D-3	69.8	70.7	72.1	71.7	72.7	72.9	72.8	73.7	72.1
E-1	68.7	69.0	71.2	70.1	72.3	72.7	73.3	74.4	69.4
x E-2	70.5	70.6	73.5	72.0	72.9	73.7	74.1	74.3	72.4
x F-1	72.6	70.7	75.6	73.2	74.8	75.6	76.3	76.7	72.9
x F-2	70.2	69.2	72.6	70.9	71.6	71.7	71.7	72.5	71.5
x H-1	73.7	71.7	76.9	74.3	75.2	76.2	77.5	77.5	73.7
H-2	79.3	79.4	85.0	82.2	82.3	83.3	84.6	84.3	81.9
I-1	67.4	67.4	70.2	68.8	70.1	70.0	70.1	71.1	70.1
I-2	68.2	69.3	69.4	69.4	69.8	69.1	69.0	69.5	70.4

x Preferred Fuels.
* Motor Method Conversion

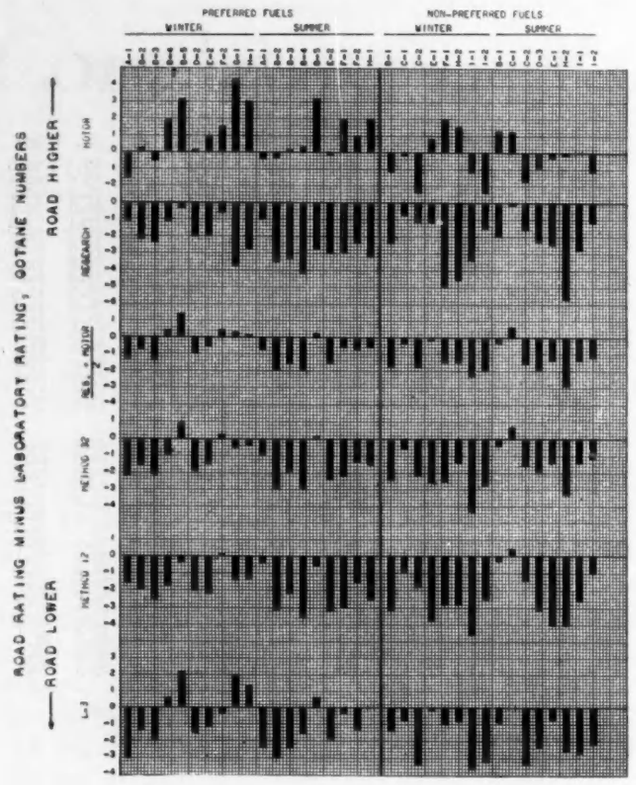


Fig. 1 – Deviations of the Average Laboratory Ratings from the Road Ratings for All Fuels When Rated by the Motor, Research, 50 Per Cent Computed Value, Methods 32, 12, and L-3

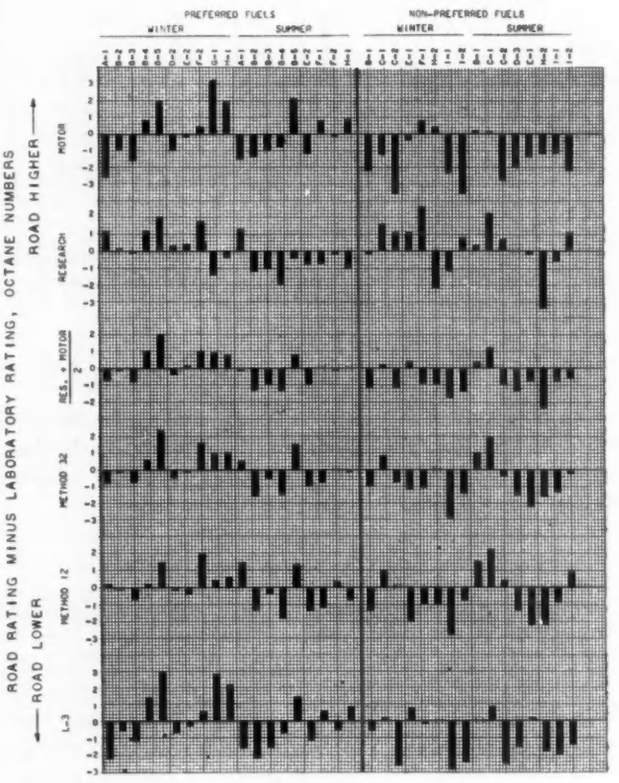


Fig. 2 – Deviations of the Average Laboratory Ratings from the 45-Deg. Line Shifted from the Origin To Give the Lowest Average Deviation on the *Preferred Fuels*

Table 3 – Deviations from Road Ratings

Fuel	Motor	Research	Res. + Motor / 2	32	12	L-3
xWA-1	-1.5	-1.1	-1.3	-2.2	-1.6	-3.0
B-1	-1.1	-2.4	-1.7	-2.4	-3.1	-1.4
x B-2	+0.2	-2.2	-0.7	-1.5	-1.9	-1.4
x B-3	-0.5	-2.4	-1.4	-2.1	-2.5	-2.0
x B-4	+1.6	-1.1	+0.4	-0.9	-1.7	+0.6
x B-5	+3.1	-0.3	+1.4	+0.9	-0.4	+2.1
C-1	-0.1	-0.7	-0.4	-0.6	-0.9	-0.7
C-2	-2.4	-1.2	-1.8	-2.1	-1.8	-3.4
x D-2	+0.1	-2.0	-1.0	-1.9	-2.0	-1.5
E-1	+0.8	-1.2	-0.2	-2.6	-3.8	-0.1
x E-2	+0.9	-1.9	-0.5	-1.5	-2.2	-1.1
F-1	+1.9	-4.9	-1.5	-2.5	-2.8	-1.0
x F-2	+1.5	-0.6	+0.4	+0.2	+0.1	-0.3
x G-1	+4.3	-3.7	+0.3	-0.5	-1.4	+2.0
x H-1	+3.0	-2.7	+0.1	-0.4	-1.3	+1.3
H-2	+1.5	-4.5	-1.5	-1.4	-2.8	-0.8
I-1	-1.2	-3.4	-2.3	-4.3	-4.6	-3.6
I-2	-2.4	-1.5	-2.0	-2.8	-2.5	-3.2
xSA-1	-0.4	-1.0	-0.7	-0.9	-0.4	-2.4
B-1	+1.3	-2.0	-0.3	-0.4	-0.3	-0.8
x B-2	-0.2	-3.5	-1.9	-3.0	-3.1	-3.0
x B-3	+0.1	-3.2	-1.6	-1.9	-2.2	-2.4
x B-4	+0.3	-4.2	-1.9	-2.9	-3.6	-1.5
x B-5	+3.2	-2.7	+0.2	+0.1	-0.5	+0.6
C-1	+1.2	-0.1	+0.6	+0.5	+0.4	0.0
C-2	-1.7	-1.6	-1.6	-1.8	-1.4	-3.3
D-3	-0.9	-2.3	-1.9	-2.9	-3.1	-2.3
E-1	-0.3	-2.5	-1.4	-3.6	-4.0	-0.7
x E-2	-0.1	-3.0	-1.5	-2.4	-3.2	-1.9
x F-1	+1.9	-3.0	-0.6	-2.2	-3.0	-0.3
x F-2	+1.0	-2.4	-0.7	-1.4	-1.5	-1.3
x H-1	+2.0	-3.2	-0.6	-1.5	-2.5	0.0
H-2	-0.1	-5.7	-2.9	-3.0	-4.0	-2.6
I-1	0.0	-2.8	-1.4	-2.7	-2.6	-2.7
I-2	-1.1	-1.2	-1.2	-1.6	-0.9	-2.2
Alg.Avg.(all)	+0.46	-2.35	-0.95	-1.70	-2.09	-1.27
Alg.Avg. (preferred)	+1.09	-2.27	-0.64	-1.37	-1.84	-0.82
Arith.Avg.(all)	1.26	2.35	1.14	1.78	2.12	1.65
Arith.Avg(preferred)	1.38	2.27	0.93	1.49	1.85	1.51

x Preferred Fuels

Table 4 – Deviations of Corrected Ratings from Road Ratings (Corrections Based on Preferred Fuels Only)

Fuel	Motor	Research	Res.+ Motor / 2	32	12	L-3
Correction	-1.1	+2.3	+0.6	+1.4	+1.8	+0.8
xWA-1	-2.6	+1.2	-0.7	-0.8	+0.2	-2.2
B-1	-2.2	-0.1	-1.1	-1.0	-1.3	-0.6
x B-2	-0.9	+0.1	-0.1	-0.1	-0.1	-0.6
x B-3	-1.6	-0.1	-0.8	-0.7	-0.7	-1.2
x B-4	+0.8	+1.2	+1.0	+0.5	+0.1	+1.4
x B-5	+2.0	+2.0	+2.0	+2.3	+1.4	+2.9
C-1	-1.2	+1.6	+0.2	+0.8	+0.9	+0.1
C-2	-3.5	+1.1	-1.2	-0.7	0.0	-2.6
x D-2	-1.0	+0.3	-0.4	-0.5	-0.2	-0.7
E-1	-0.3	+1.1	+0.4	-1.2	-2.0	+0.7
x E-2	-0.2	+0.4	+0.1	-0.1	-0.4	-0.3
F-1	+0.8	+2.6	-0.9	-1.1	-1.0	-0.2
x F-2	+0.4	+1.7	+1.0	+1.6	+1.9	+0.5
x G-1	+3.2	-1.4	+0.9	+0.9	+0.4	+2.8
x H-1	+1.9	-0.4	+0.4	+0.7	+0.5	+2.1
H-2	+0.4	-2.2	-0.9	0.0	-1.0	0.0
I-1	-2.3	-1.1	-1.7	-2.9	-2.8	-2.8
I-2	-3.5	+0.8	-1.4	-1.4	-0.7	-2.4
Correction	-1.1	+2.3	+0.6	+1.4	+1.8	+0.8
xSA-1	-1.5	+1.3	-0.1	+0.5	+1.4	-1.6
B-1	+0.2	+0.3	+0.3	+1.0	+1.5	0.0
x B-2	-1.4	-1.2	-1.3	-1.6	-1.3	-2.2
x B-3	-1.0	-1.0	-1.0	-0.5	-0.4	-1.6
x B-4	-0.8	-1.9	-1.3	-1.5	-1.8	-0.7
x B-5	+2.1	-0.4	+0.8	+1.5	+1.3	+1.4
C-1	+0.1	+2.2	+1.2	+1.9	+2.2	+0.8
C-2	-2.8	+0.7	-1.0	-0.4	+0.4	-2.5
D-3	-2.0	0.0	-1.3	-1.5	-1.3	-1.5
E-1	-1.4	-0.2	-0.8	-2.2	-2.2	+0.1
x E-2	-1.2	-0.7	-0.9	-1.0	-1.4	-1.1
x F-1	+0.8	-0.7	0.0	-0.8	-1.2	+0.5
x F-2	-0.1	-0.1	-0.1	0.0	+0.3	-0.5
x H-1	+0.9	-0.9	0.0	-0.1	-0.7	+0.8
H-2	-1.2	-3.4	-2.3	-1.6	-2.2	-1.8
I-1	-1.1	-0.5	-0.8	-1.3	-0.8	-1.9
I-2	-2.2	+1.1	-0.6	-0.2	+0.9	-1.4
Alg.Avg.(all						
" " (preferred)						
Arith.Avg.(all)	1.42	1.03	0.84	1.01	1.05	1.27
" " (preferred)	1.28	0.89	0.70	0.84	0.83	1.32

x Preferred Fuels

Group. However, having chosen the most promising combinations of variables, it was believed advisable to obtain additional information on the influence of spark advance on the ratings of a few selected fuels. This work indicated that changing the spark setting from 17 deg. advance does little toward improving average correlation with the road ratings, and tends to increase the maximum spread about the best average line through the points.

Inasmuch as an appreciable length of time had elapsed between the original fuel ratings by the Motor Method and this later work, it was deemed advisable to obtain new Motor-Method ratings concurrently with the experimental methods Nos. 12, 30, 32, and 27. The Motor values used in this report therefore are not necessarily equal to those given either in the Road or Laboratory Reports. The Research and L-3 values are those reported by the Laboratory Procedure Group.

In addition to the single engine methods, correlation was investigated using varying percentages of the difference between the Research and Motor Methods added to the Motor Method for each fuel. The best results were obtained by using a factor of 50 per cent of the preceding difference.

Table 2 presents the average ratings of all fuels by the road and laboratory methods under consideration.

Discussion of Results

The evaluation of the various laboratory methods, from the standpoint of correlation, was based on:

1. The deviations of the average laboratory ratings from the road ratings.

2. The deviations of the average laboratory ratings from a 45-deg. line displaced from the origin to give the lowest average deviation.

In this analysis only the preferred winter and summer fuels were used. The non-preferred fuels were omitted due to the relatively small number of road ratings used in obtaining the averages.

The results of the new methods investigated indicated that the two methods at 900 r.p.m. were superior to the methods at 600 r.p.m., hence the latter two methods were not included in the subsequent analysis.

Fig. 1 represents diagrammatically the deviations of the

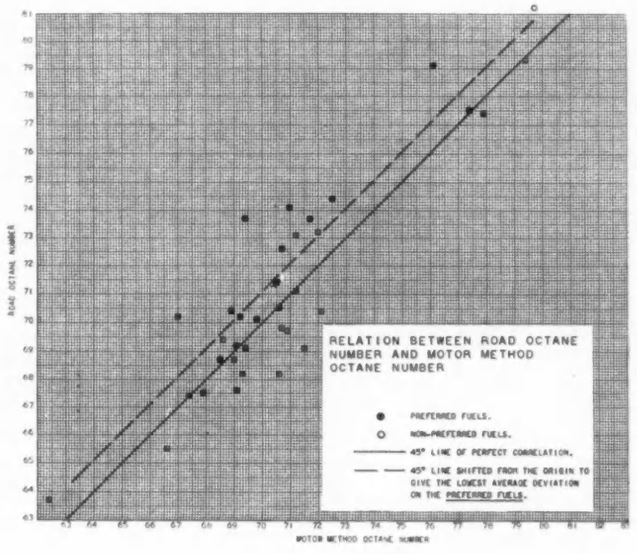


Fig. 3

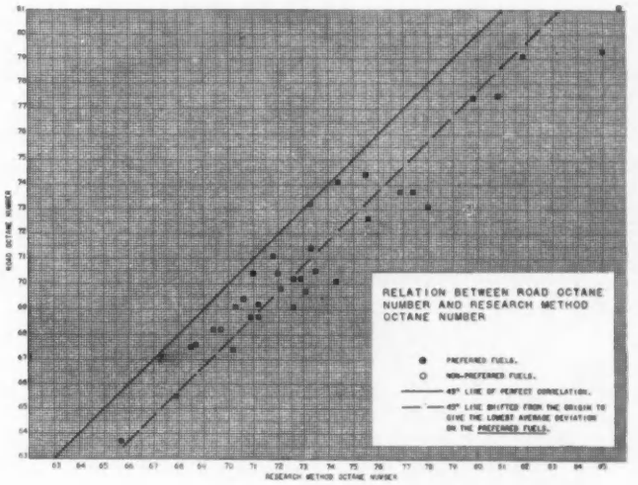


Fig. 4

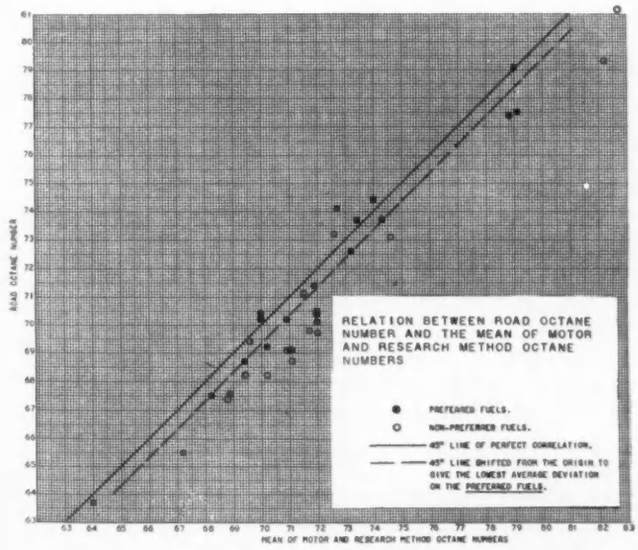


Fig. 5

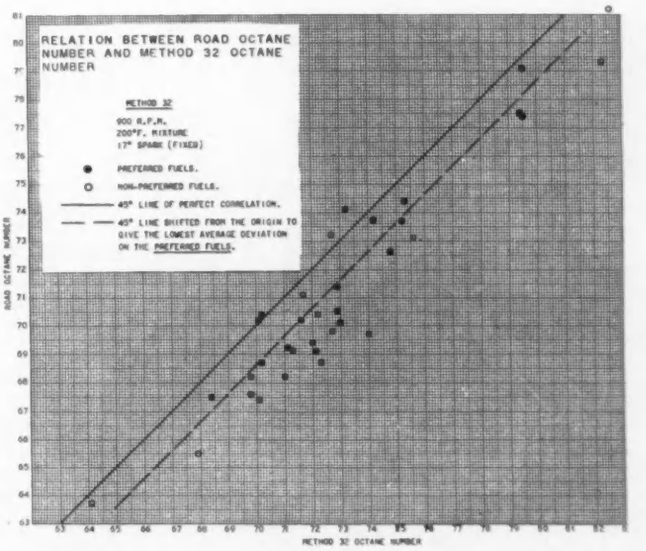


Fig. 6

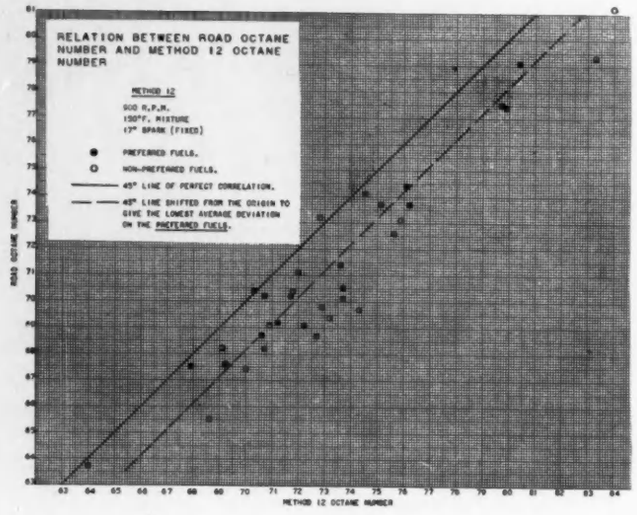


Fig. 7

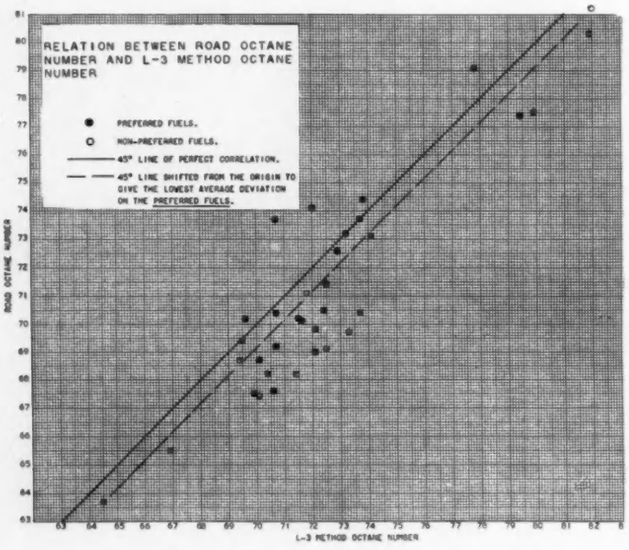


Fig. 8

average laboratory ratings for each fuel from its average road rating. Table 3 gives the same values numerically as well as the overall average by each method. The arithmetic average of the deviations of the preferred fuels is shown in the first column in Fig. 9.

It will be noted that the least deviation was obtained by the mean of the Motor and Research Methods, whereas the largest deviation occurred with the Research Method. If the Motor Method is used as the basis for comparison, the two methods mentioned show deviation percentages of 67 and 165 respectively.

Judging correlation on the basis of the arithmetic average alone gives little information as to the distribution of ratings about the line of perfect correlation. The algebraic average, on the other hand, gives a measure of the average deviation from perfect correlation, but no measure of the maximum deviation or scattering of the individual points.

In order to combine the advantages of both the arithmetic and algebraic averages, the deviations of the individual ratings were determined about a 45-deg. line displaced from the origin to give the lowest average deviation. These are given in bar form in Fig. 2, and numerically in Table 4. The amount by which this line is displaced should obviously be equal to the algebraic average of the ratings for the particular laboratory method in question. For simplicity in presentation the slope of this line was not changed from that of perfect correlation. This line differed from the best average line through the points by such a small angle that it is believed that the results are not seriously affected. The displaced line is indicated by dashes on Figs. 3, 4, 5, 6, 7, and 8. Examination of these figures will show that, in all except that for the Research Method, the spread of fuels includes the perfect correlation line.

Although it is desirable that the best average line through the points coincides with the line for the reference fuels (perfect correlation line), this condition is not necessary to predict the relative ratings of fuels on the road. Since there are commercial fuels in this country similar to the reference fuels, it is, however, desirable that the line for the reference fuels fall between the points representing fuels with the maximum deviation on either side of the displaced line. This view is not in accord with the British idea that the line for perfect correlation may be disregarded. The bars on the right of Fig. 9 show the width of the 45-deg. band necessary to include all points for the various methods analyzed. The group of bars in the center of the figure represent the average arithmetic deviations from the displaced line for each Method.

Conclusions

Judging the methods purely from the standpoint of deviation and maximum spread, there is little to choose between the Research, 12, and 32. However, the mean of the Motor and Research Methods furnishes somewhat better correlation.

In addition to these criteria for selecting an improved laboratory method, there are other factors to be considered, for example:

1. The complication of two engine methods.
2. Relative reproducibility.
3. The extent to which these fuels are representative on a weighted basis of commercial gasolines. The latter consideration becomes of greater significance as the adjusted correlation line is moved further from the line of perfect correlation.

In view of these factors and present information, no change of method is recommended. Both road and laboratory data that were available to the Group are the most reliable obtained to date. It is believed that no material improvement is likely to be effected by the accumulation of additional laboratory data of the type considered in this analysis, that is, data involving only speed, temperature, and spark advance. The failure to achieve better correlation is not due primarily to shortcomings of the data, but to the failure to take cognizance of all the factors influencing car ratings.

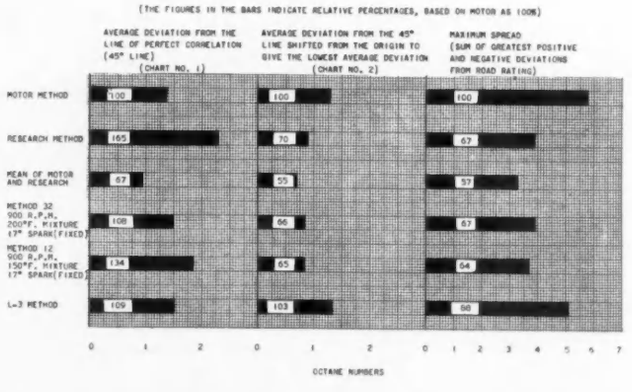


Fig. 9 – Average Deviations of the Laboratory Ratings from the Road Ratings and Maximum Spread of the Deviations – Based on *Preferred Fuels* Only

Application of Statistical Concepts to the Knock-Rating Problem

By John M. Campbell and W. G. Lovell
Research Laboratories Division, General Motors Corp.

CONCLUSIVE evidence that any one make of car has a real, as distinguished from accidental, tendency to rate fuels higher or lower than the average for all cars is difficult to find in existing data, the authors report.

The growing economic and technical importance of antiknock quality, they point out, is making it increasingly desirable to estimate the random distribution of errors that occur in making measurements of knock, either in the laboratory or in road tests. Statistical analysis, they explain, offers a means of appraising the probability of occurrence of errors of various magnitudes. To make a beginning in such a statistical analysis of the available data on fuel ratings is the purpose of their paper, they announce.

The variability, or random error, among successive measurements in road tests was from three to six times higher than the variability among measurements of octane number in the A.S.T.M.-C.F.R. single-cylinder test engine, the paper reports. Also, the variability among ratings made in different makes of cars was only slightly greater than the variability among repeated tests in a single car of one make.

The high variability in road ratings also makes it necessary to obtain a relatively large number of road ratings on any particular gasoline before a reliable comparison can be made between it and other gasolines with the same degree of precision that is obtained in the mean of three or four laboratory ratings on each fuel, the paper concludes.

ONE of the fundamental problems in connection with the making of measurements of any kind is the estimation of the reliability of the measurements themselves. In this respect the measurement of antiknock quality is no exception, and the increasing economic and technical importance of antiknock quality makes it increasingly desirable to have available suitable methods of appraising the significance of an octane-number rating.

Random variations in knock ratings either in the laboratory or in road tests may be ascribed to experimental error or to factors either known or unknown which are beyond control in present methods. Statistical methods of analysis make it possible to evaluate these random variations in terms of the probabilities that certain ratings will be obtained or might have been obtained on the basis of past experience.

The statistical method is based essentially on a study of the distribution and frequency of variations from the mean of a representative sample of data. Provided the data follow a random distribution, it is possible to determine a number of useful types of information about any particular set of data. Among them are:

1. Evaluation of experimental error with respect to other related data.
2. Determination of how the random error varies with experimental conditions.
3. Probability of errors of various magnitudes.
4. Approximate amounts of data required for any desired degree of precision.

It is the purpose of this paper to make a beginning in the statistical analysis of the available data on fuel ratings with these ends in view. At most points the data are not as complete as might be desired, but are sufficient to illustrate the principles involved and the general type of information obtainable by such methods of analysis.

Concept of Variability – In making repeated knock ratings on a given gasoline, whether in a laboratory test engine or in a car, it is a matter of common experience that repeated ratings obtained are not all identical, and only by averaging a number of independent observations is it possible to obtain an estimate of what the octane number really is. Emphasis is placed on the word *estimate*, because we never know exactly what the octane number is. At best, the octane number can be expressed only as an average of a certain number of independent observations and, as such, its meaning is inherently associated with this particular mathematical operation. The statistical methods discussed in this paper deal with the interpretation of data which have been subjected to this averaging process.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 16, 1938.]

Table 1 – Comparison of Six Independent Sets of Laboratory Knock-Ratings on One Fuel* (A.S.T.M. Method D357-36T)

	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	All
	70.5	69.7	70.5	71.0	71.0	69.5	.....
	71.9	70.5	70.7	70.5	71.3	70.6	.....
	71.0	70.4	71.0	71.2	70.8	71.5	.....
	71.5	70.2	70.5	70.8	70.7	70.6	.....
	71.1	71.0	70.3	70.1	69.8	70.2	.....
	70.1	71.0	71.2	70.8	70.5	70.6	.....
	69.8	71.4	70.1	71.4	70.6	71.0	.....
	70.5	70.5	71.0	71.0	70.0	70.8	.....
	70.0	70.8	70.4	71.0	69.9	71.4	.....
	71.1	70.9	70.0	70.6	70.8	70.1	.....
	.....	70.5	71.1	70.6	.....	70.2	.....
	.....	71.2	71.0	70.4	.....	.....	.....
	.....	.....	71.4	.....	.....	.....	.....
	.....	.....	71.0	.....	.....	.....	.....
	.....	.....	71.0	.....	.....	.....	.....
	.....	.....	71.2	.....	.....	.....	.....
	.....	.....	70.4	.....	.....	.....	.....
Number.....	10	12	17	12	10	11	72
Mean.....	70.8	70.7	70.8	70.8	70.5	70.6	70.7
Standard Deviation σ....	0.65	0.45	0.41	0.35	0.47	0.56	0.49

* 1937 C.F.R. road test fuel W-SE-2, each rating in each set made independently by different laboratories. Repeated ratings made at later dates by a single laboratory sometimes occur in succeeding sets of data.

A typical series of actual laboratory knock ratings is given in Table 1. The mean of all these ratings is 70.7 (70.6903 ....). A measure of variability of the data may be obtained from the relation:

$$\sigma = \sqrt{\frac{\Sigma \bar{v}^2}{n}} \qquad (1)$$

where σ = variability (standard deviation).

$\Sigma \bar{v}^2$ = summation of squares of individual deviations from mean.

n = total number of ratings.

A more convenient means of calculating σ is by the use of Equation (2) which is derived from Equation (1):

$$\sigma = \frac{1}{n} \sqrt{n\Sigma V^2 - (\Sigma V)^2} \qquad (2)$$

where ΣV^2 = summation of squares of individual ratings.

$(\Sigma V)^2$ = square of summation of individual ratings.

[1]See Report of Analyzing Group of Motor Fuels Section of Cooperative Fuel Research Committee, September, 1937: "Analysis of Car Knock Ratings in 1937 Cars."

[2]See S.AE. TRANSACTIONS, June, 1935, pp. 215-220: "C.F.R. Committee Report on 1934 Detonation Road Tests Discussed," discussion by Donald B. Brooks.

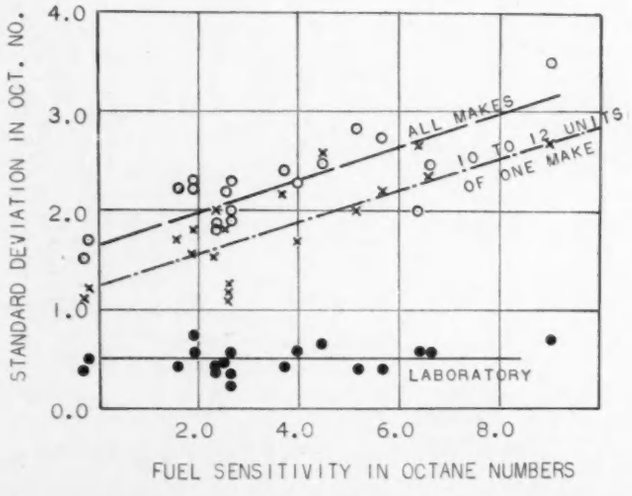


Fig. 1 – Analysis of Variability in Knock-Ratings

In this particular example $\sigma = 0.49$. This quantity is similar in meaning to the average deviation which has been used commonly heretofore in the tabulation of knock ratings. However, calculated in this way, the "standard deviation" or "standard unit" of variability is much more useful than the average deviation in that it can be used to evaluate approximately the probability of random deviations of any given magnitude from the mean. The evaluations possible in most of the practical problems at hand are approximate because of the limited amount of data usually available and because of certain limitations in the statistical theory itself.

Representative Values for Variability in Laboratory and Road Knock-Testing – There are four conditions found in connection with knock-rating work where a knowledge of variability is desirable:

1. Laboratory tests.
2. Repeated road ratings in a single car.
3. Road ratings in different cars of same make.
4. Road ratings in cars of different make.

Data for calculating variability in Items 1, 3, and 4 of this list are available in the report of the 1937 C.F.R. road knock tests.[1] The standard deviations for each of the preferred fuels are listed in Table 2 and are shown in Fig. 1 plotted against fuel sensitivity (Research minus A.S.T.M. rating). Fig. 1 shows a tendency for the standard deviation in road tests to increase with increasing fuel sensitivity. Laboratory variability was not affected appreciably by fuel sensitivity. No particular significance is attached to the variations in variability of the laboratory ratings of different fuels because, as shown in Table 1, considerable spread in variability was found among different samples of the same fuel.

Table 2 – Variability in Knock Tests Expressed in Octane Numbers

Fuel	Sensitivity	A.S.T.M. Variability[1]	Road Variability in One Car Make[2]	Road Variability in All Makes[3]
WA-1	−0.3	0.37	1.1	1.5
WB-2	1.8	0.41	1.7	2.2
WB-3	2.3	0.43	2.0	1.8
WB-4	4.0	0.61	1.7	2.3
WB-5	3.7	0.42	2.4	2.2
WD-2	1.9	0.64	1.6	2.3
WE-2	2.3	0.41	1.6	1.8
WF-2	1.9	0.75	1.8	2.2
WG-1	9.0	0.73	2.7	3.5
WH-1	6.4	0.59	2.7	2.0
SA-1	−0.2	0.50	1.2	1.7
SB-2	2.5	0.47	1.8	2.2
SB-3	2.6	0.62	1.3	1.9
SB-4	4.5	0.67	2.7	2.6
SB-5	5.2	0.39	2.0	2.8
SE-2	2.6	0.35	1.1	2.3
SF-1	5.7	0.38	2.2	2.7
SF-2	2.6	0.27	1.1	2.0
SH-1	6.6	0.52	2.4	2.5
Average	3.5	0.50	1.8	2.2

[1] July-August 1937 exchange ratings.
[2] 9 to 12 units of Car *A*.
[3] 50 to 65 individual cars representing 23 different makes.

Item 2, were published by Brooks in 1935.[2] In the tests reported at that time, the variability was of the order in magnitude of 1.0 to 1.5 octane numbers in one make of car.

The data in Tables 3-A and 3-B represent the results of some tests made by the Ethyl Gasoline Corp. Laboratories

Table 3-A – Octane Ratings of a Blend of 60 Per Cent C-10 in A-5 (63.3 Octane Number) Obtained in Eight 1937 Cars

Car	Rating Number 1	2	3	4	5	6	7	8	Mean	Standard Deviation
A	63.8	65.7	65.7	65.7	63.8	67.6	67.6	65.8	65.7	1.4
B	68.8	55.0	56.3	61.2	61.2	61.2	61.2	61.2	60.8	4.1
C	60.0	60.0	60.0	63.8	56.3	67.6	67.6	72.8	63.5	5.1
D	58.7	62.2	62.2	63.8	62.2	62.2	62.2	62.2	61.8	1.6
E	60.0	63.8	63.8	63.8	60.0	60.0	63.8	63.8	62.4	1.9
F	60.2	58.2	63.8	63.8	63.8	63.8	63.8	63.8	62.6	2.1
G	63.8	62.5	64.8	64.8	65.8	65.8	63.8	65.8	64.6	1.1
H	63.8	63.8	64.8	63.8	63.8	63.8	65.5	65.5	64.4	0.7
							Combined data		63.2	3.0

Table 3-B – Octane Ratings of a Blend of 90 Per Cent C-10 in A-5 (75.0 Octane Number) Obtained in Eight 1937 Cars

Car	Rating Number 1	2	3	4	5	6	7	8	Mean	Standard Deviation
A	75.0	75.0	73.0	73.0	77.0	76.0	76.0	76.0	75.1	1.4
B	75 0	75.0	75.0	71.4	75.0	75.0	75.0	77.0	74.8	1.4
C	75.0	75.0	75.0	75.0	75.0	75.0	74.3	75.0	74.9	0.2
D	75.0	75.0	77.0	77.0	77.0	77.0	77.0	77.0	76.5	0.9
E	75.0	75.0	75.0	73.3	75.0	77.0	77.0	74 0	75.2	1.2
F	75.1	75.1	75.1	75.1	75.1	75.1	75.1	78.0	75.4	0.9
G	73.0	75.0	75.0	75.0	75.0	75.0	75.0	75 0	74.8	0.7
H	75.0	77.0	75.0	72.3	77.0	75.0	75.0	77.0	75.4	1.5
							Combined data		75.3	1.2

which were made available to the C.F.R. Committee in its study of the correlation problem. In these tests two reference-fuel blends of known composition were rated blind eight successive times in each of eight different makes of car. All the ratings in each car were made on the same day without any change in the adjustments of the car. The two fuel blends which were rated in this way differed by about 12 octane numbers, so that the ratings on the fuel of lower octane number were made at medium to heavy knock intensities, whereas the ratings on the fuel of higher octane number were made at trace to light knock intensities. These data show that the variability in knock ratings was affected materially by knock intensity, the variability becoming less at the lighter knock intensities. Under the most favorable conditions, that is, at light knock, the standard deviations for all the ratings irrespective of car was 1.2 octane numbers. This figure is in good agreement with the data previously published by Brooks.[2] No particular significance is attached to the variations of standard deviations from car to car in Table 3 because of the relatively small amount of data available.

From a study of Fig. 1 and Tables 1, 2, and 3, it appears that the maximum control over conditions producing random variations is had in the laboratory test method. As close control over test conditions becomes less as in road tests, particularly at varying degrees of knock intensity, the magnitude of random deviations becomes more and more apparent.

Variability and Probability – The distribution about the mean of the data in Table 1 containing 72 laboratory ratings of the same fuel is shown in Fig. 2. This distribution is suggestive of data conforming to a normal curve of error. For example, the curve drawn in Fig. 2 is a normal curve of error which corresponds with the data represented in this problem.

[3]See "Probability and Its Engineering Uses," by T. C. Fry, D. Van Nostrand Co., Inc., New York, 1928; "Introduction to the Theory of Statistics," by G. U. Yule, Charles Griffin Co., Ltd., London, 1932; and "Statistical Methods for Research Workers," by R. A. Fisher, Oliver and Boyd, Ltd., Edinburgh, 1936.

Table 4 – Probability of Occurrence of Deviations of a Single Rating from the Mean of a Large Number of Ratings – Standard Deviation 0.5 Octane Number

± Deviation from Mean	Ratio Deviation to Standard Deviation ("t" Expressed in Standard Units)	Probability of Occurrence (Fraction Exceeding ± Limiting Deviation)	
0.0	0.0	1.00	(1.000000)
0.05	0.1	0.92	(0.920344)
0.2	0.4	0.69	(0.689156)
0.33725 ("Probable error")	0.6745	0.50	(0.500000)
0.5	1.0	0.31	(0 317310)
1.0	2.0	0.04	(0.045500)
1.5	3.0	0.00	(0.002700)
2.0	4.0	0.00	(0.000064)

The equation for such a curve is:

$$y = \frac{1}{\sqrt{2\pi}} e^{-\frac{t^2}{2}} \qquad (3)$$

where t is expressed in standard units and is the ratio of the deviation at any point to the standard deviation.

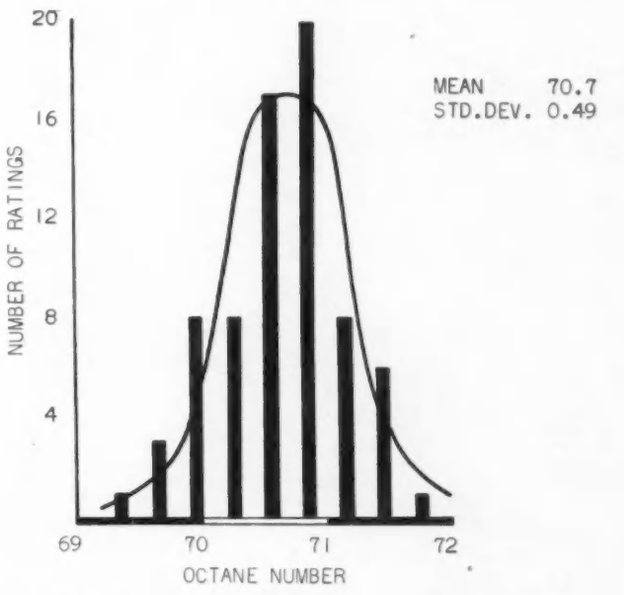


Fig. 2 – Distribution of 72 Independent Octane Ratings on One Fuel

Assuming that the data do follow a normal curve of error – there being no reason to suspect otherwise – the probability that a single laboratory knock-rating will deviate from the mean of a large number of data can be calculated from the standard deviation. The probabilities of occurrence of deviations of various magnitudes may be obtained by reference to tables obtainable in various texts on statistics and probability,[3] of integrated areas beneath the normal curve of error, using for this purpose a standard deviation of 0.5 octane number which appears from the data in Tables 1 and 2 to be a reasonable value to use. The resulting probabilities of the occurrence of deviations from the mean are tabulated in Table 4. This table shows, for example, that there is about one chance in three (probability 0.31) that a single knock-rating will deviate from the mean of a large number by at least ±0.5 of an octane number. Applying this relation to the actual data in Table 1, out of 72 ratings we should expect to find about 72 × 0.31 or 22 ratings outside the range of 70.7 ± 0.5 octane number. By actual count, there were 20 ratings outside

this range, which is within reasonable agreement with the calculated number. Perfect agreement in this case would be accidental because of the limited amount of data available.

Table 4 shows further that only about 3 ratings in a thousand (probability 0.0027) would be expected to deviate from the mean by more than 1.5 octane numbers due to accidental causes. Accordingly, deviations of this order in magnitude and greater are most likely to be in error due to some causes other than those which ordinarily account for experimental error.

The use of more than two significant figures in probability values when dealing with less than 100 observations, which is usually the case in knock problems, of course, is unwarranted. The more exact values are shown in parentheses merely to indicate that the values carried to two decimal places are an approximation.

Sampling – Where greater precision than that obtained from one rating is desired, it is customary to obtain an average of several ratings. Ordinarily it is not practicable to obtain as many ratings on a single fuel as are available in Table 1. Usually it is necessary to compromise with a small number of ratings. The probability of error decreases approximately with the square root of the number of ratings obtained according to the relation:

$$\pm e = \frac{K\sigma}{\sqrt{n}} \qquad (4)$$

where e = deviation of mean of small number of ratings from mean of an infinitely large amount of data.
K = a constant, characteristic of a given probability expressed in standard units.
n = number of ratings actually used.
σ = standard deviation.

For small samples it is sometimes preferable to use a standard deviation determined from other and more extensive data of the same kind rather than that of the particular data at hand, since Equation (4) is based on the assumption that σ was calculated from an infinitely large amount of data. Actually this is not possible, but if σ is based on a sufficiently large quantity of data, it is suitable for practical purposes as a guide in experimental work.

Values of K suitable for substitution in Equation (4), corresponding to various probabilities, are shown in the following table:

Table 5 – Values of *K* Corresponding to Various Probabilities

Probability	K
0.5 (1 chance in 2)	0.67
0.1 (1 chance in 10)	1.64
0.01 (1 chance in 100)	2.58
0.001 (1 chance in 1000)	3.29

The effect of increasing the size of the sample on the probable precision of the mean of a group of laboratory knock-ratings in accordance with Equation (4) is shown in Fig. 3 using a standard deviation of 0.5 of an octane number for laboratory fuel ratings. This figure shows, for example, that there is approximately 1 chance in 10 that the mean of 10 independent knock ratings will deviate from the mean of a large number by more than 0.26 of an octane number. This is in reasonable agreement with the mean values determined for the six samples of data reported in Table 1. According to this figure, there is only 1 chance in 1000 that the mean of 10 laboratory knock-ratings will differ from the mean of a much larger number by more than 0.5 of an octane number. From such information it is possible to estimate how many

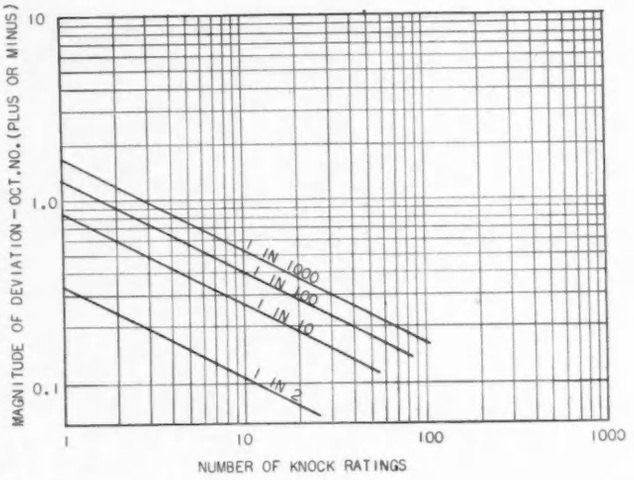


Fig. 3 – Probability of Error in Laboratory Knock-Ratings Using Standard Deviation of 0.5 Octane Number

ratings actually should be made in order that the mean may have the degree of certainty derived in accordance with the individual's judgment of the requirements of the problem.

Probability of Error in Road Knock-Ratings – Tables 2 and 3 show that the standard deviation is from three to six times greater in road tests than in laboratory tests. Accordingly, the probability of deviations from the mean is correspondingly greater. Using a standard deviation of 2.0 octane numbers as representative of an average commercial gasoline in road tests, Fig. 4 shows the relationship between the probability of deviations of various magnitudes and the size of the road-test sample. This relationship may be applied within reasonable limits either to one car, or to different cars of one make, or to cars of different makes since, in each case, the variability is of the same order in magnitude.

Fig. 4 shows, for example, that there is approximately 1 chance in 10 that an average rating based on 50 road ratings in different car makes having a variability of 2.0 octane numbers will deviate from 0.4 to 0.5 of an octane number from the mean of a much larger amount of data. To get the same degree of certainty in laboratory ratings an average of only three independent ratings would be required.

Analysis of Ratings in a Single Car Make – Deviations from the mean in road ratings may be attributed either to random error or to real differences between one car and another. As shown by comparison of the data in Tables 2 and 3, the variability among repeated tests in a single car is of the same order in magnitude as that observed among tests made in different units of the same make and even among cars of different make. Consequently, it is difficult, upon the basis of present information, to distinguish any real tendency of one car make to rate fuels higher or lower than the average for all cars.

In order to get some idea from available data of the degree of certainty that a given make of car rates sensitive fuels appreciably higher or lower than the average, the following analysis has been made. Four of the most sensitive fuels used in the 1937 Road Tests were set aside and adjusted to a common mean, as in Table 6. Then, by making the same adjustment for the average of each of these fuels as rated in one car make, any deviation of this adjusted average from the average for all cars would be the amount representing either a real or chance departure of ratings in that particular car make from the mean of all cars. For example, the ratings obtained in Car *A* are summarized in Table 7. A total of 44 ratings had an adjusted mean of 72.1, which was 0.5 of an octane

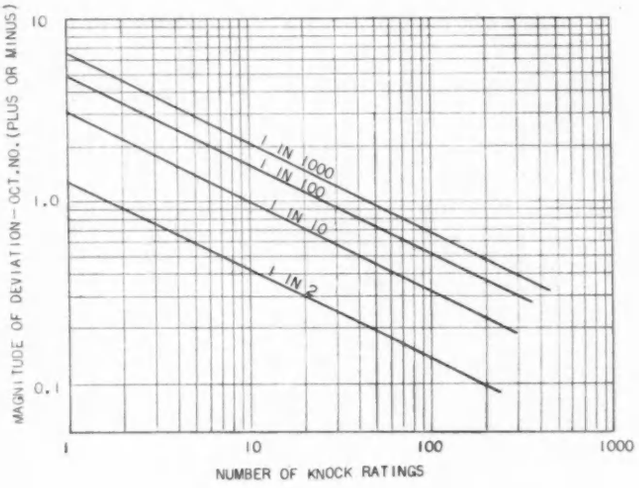


Fig. 4 Probability of Error in Road Knock-Ratings Using a Standard Deviation of 2.0 Octane Numbers

number below the adjusted mean for the 196 ratings representing all cars in Table 6.

Determination of the probability that this deviation was due to chance may be carried out by either of two methods of approach. The first method is based on the use of a modification of Equation (4) to determine the probability of error with respect to an infinitely large sample. The second method involves the determination of the probability of drawing a sample of 44 at random from a population of 196 which would deviate by 0.5 of an octane number from the mean of the parent population.

The calculation by the first method using a modification of Equation (4) is as follows, where t is expressed in standard units:

$$\epsilon = \frac{t\sigma}{\sqrt{n}} = -0.5 = \frac{t \times 2.5}{\sqrt{44}}$$

whence $t = -1.33$. The probability of the mean of the sample exceeding the mean of a large sample by more than -1.33 standard units is 0.09 or roughly 1 chance in 10. (The integrated area from $-\infty$ to -1.33 under the curve represented by Equation (3) is 0.09. See footnote 3.)

The calculation by the second method is a problem in sampling from a discrete number of variates. Where the standard deviation of the parent population numbering 196 is 2.8, the standard deviation for all possible combinations of 44 may be calculated from the equation:

$$\sigma_s = \sigma_p \sqrt{\frac{S-n}{n\,(S-1)}} \qquad (5)$$

where: σ_s = standard deviation of samples.
σ_p = standard deviation of parent population.
S = number of parent population.
n = number in sample.

substituting in Equation (5):

$$\sigma_s = 2.8 \sqrt{\frac{196-44}{44\,(196-1)}} = 0.37$$

The mean of all possible combinations of 44 would be equal to the mean of the parent population, in this case 72.6. Accordingly, the sample we have would be -0.50 divided by 0.37 or -1.35 standard units from the mean of all possible samples. The probability of such an occurrence is 0.09 or, again, about 1 chance in 10. Thus, both methods of analysis agree that the chance is roughly 1 in 10 that the observed deviation of -0.5 of an octane number may have been due purely to chance. Although there are remaining 9 chances out of 10 that the effect was due to some real factor which distinguishes this make of car from the rest, the probability, even after 44 tests, is not high enough to make such a conclusion very certain. On the other hand, if the deviation from the average had been -1.0 octane number – not a very great change in octane number – there would have been only 3 chances in 1000 that the effect was due purely to chance. A deviation of -2.0 octane numbers in the 44 tests from the average would indicate practical certainty that the deviation was the result of a real car characteristic.

Table 6 – Analysis of Car Ratings on Sensitive Fuels

Fuel	Total Ratings	Mean	Adjustment to Mean 72.6	Adjusted Mean	Σv^2	σ	Sensitivity
SF-1	50	72.6	0.0	72.6	375.12	2.7	5.7
WG-1	56	73.9	1.3	72.6	669.23	3.5	9.0
WH-1	40	79.1	6.5	72.6	194.16	2.0	6.4
SH-1	50	74.0	1.4	72.6	328.12	2.5	6.6
Total	196			72.6	1566.63	2.8	6.9

Table 7 – Analysis of Ratings in One Make of Car

Fuel	No. of Ratings	Car Average	Adjustment from Table 6	Adjusted Car Average	ΣV	Σv^2	σ
SF-1	11	72.4	0.0	72.4	796.4	55.49	2.2
WG-1	12	72.1	1.3	70.8	849.6	84.91	2.7
WH-1	9	78.6	6.5	72.1	648.9	67.90	2.7
SH-1	12	74.3	1.4	72.9	874.8	71 73	2.4
Total	44			72.1	3169.7	280.03	2.5

Comparison of Knock Test with Other Methods of Test – Comparison of the observed standard deviations (variability) for various methods of test is one possible criterion of the relative precision obtainable with such tests. For example, Table 8 is a comparison of standard deviations calculated for the A.S.T.M. 10 per cent points, Reid vapor pressures, and knock-ratings reported by 17 laboratories on C.F.R. Exchange samples R-125, R-126, and R-127.

From these data the variability of the knock-ratings appears to occupy the smallest percentage of the working range of any of the three tests analyzed. In spite of this condition, however, because of the economic significance of octane ratings, the random distribution of errors in knock measurements is a matter of considerable importance. In order to make an accurate appraisal of the reliability of any given data, it is desirable to have a statistical background by means of which the probability of random variations can be evaluated. Beyond this background, the judgment of the individual must be depended upon to interpret the statistical probabilities into a suitable course of action. Statistical analysis is an aid to, but is in no wise a substitute for, individual judgment.

Acknowledgment – The authors are indebted to Prof. H. C.

Table 8

A. S. T. M. Test	Standard Deviation	Approximate Working Range	Per Cent Working Range Represented by Variability
10 per cent distilled	2.5 deg. fahr.	115–160 deg. fahr.	5
Reid vapor pressure	0.8 lb.	5–13 lb.	10
Knock-rating	0.5 octane number	50 to 80 octane number	2

Carver of the University of Michigan for consultation and advice in connection with the statistical theory involved in these computations and to the staff of the Ethyl Gasoline Corp. Laboratories for permission to use the data in Table 3.

Discussion

Applies "Car Severity Factors"

—J. O. Eisinger
Standard Oil Co. (Ind.)

THE authors' analysis, as we interpret it, shows that, under the present conditions of knock testing in order to obtain the same certainty, it is necessary to make about 14 times as many observations on the road as in the laboratory. The standard deviation of 1.2 octane numbers for repeated ratings *of one fuel in the same car* suggests the need for more rigid supervision and control of road-test methods. The road work of the C.F.R. and A.P.I. this year as currently outlined, calling for more uniform and controlled test technique, certainly is a step in the right direction.

The authors' Fig. 1 shows a tendency for the standard deviation in tests on the same fuel in different cars to increase with increasing fuel sensitivity. Certain cars are known to rate fuels higher than other cars. This condition may be true of individual cars of the same make and model. This tendency, which is identified with a definite car characteristic technically termed "car severity," applies broadly to all fuels and particularly to fuels of the sensitive type. For example, in the 1937 C.F.R. road tests, sensitive fuel WG-1 had a road octane rating of 83.9 in Car *S* as compared to only 67.2 in Car *O*, a difference of 16.7 octane numbers. Such differences are not to be confused with random errors such as arise in making repeated tests of a single fuel in a single car.

These considerations indicate that individual deviations from the mean of a number of ratings of a given fuel in different cars are attributable partly to car-severity characteristics and partly to random errors associated with such tests. The deviations resulting from ratings made in cars of different severities obviously will be greatest for fuels of highest sensitivity. As a result, it would be expected that the standard deviations shown in the authors' Fig. 1 would be greatest for fuels of highest sensitivity. This is true not necessarily because random errors are greater for this type of fuel but, in large part, because of real differences in car ratings. This explanation may be implied in the authors' paper, or is perhaps obvious. However, it may prevent some readers from construing Fig. 1 as inferring that sensitive fuels are more difficult to rate on the road than non-sensitive fuels.

We feel that we must take issue with this statement of the authors which appears under the heading, "Analysis of Ratings in a Single Car Make":

"Consequently, it is difficult, upon the basis of present information, to distinguish any real tendency of one car make to rate fuels higher or lower than the average for all cars."

In our opinion, the writers have not properly substantiated this statement. We have computed severity factors for all cars included in the 1937 C.F.R. road test program, using the ratings of a large number of fuels for this purpose. Car *A*, which was used by the authors in their example, was found to have a severity factor only slightly higher than the average car. Consequently, this car would not be expected to rate fuels much below the mean rating for all cars, as evidenced by the difference of only 0.5 octane number found by the authors. Car *B* was found to be somewhat less severe than the average car. Applying the authors' method, using the same four sensitive fuels, a total of 19 ratings in Car *B* had an adjusted mean of 73.8, which was 1.2 octane numbers above the adjusted mean for the 196 ratings representing all cars (authors' Table 5). The probability of a mean of this sample differing from the mean of a much larger sample by more than 1.2 octane numbers was ascertained as approximately 1 chance in 50 – a somewhat greater certainty than the authors found for Car *A*.

Cars *A* and *B* and all other cars of which large numbers were tested had severities very close to the average car. Unfortunately, only one or two units were tested of those cars which were found to have the highest and lowest severities. Even though there were not sufficient data on these cars to permit the statistical treatment of the writers, there is very little reason to believe that the extent to which they appreciated or depreciated sensitive fuels was due purely to chance. Although each laboratory submitted to the C.F.R. only one rating of each fuel in each car, the probability is that each fuel-car combination was tested several times. Had it been possible for the authors to apply their statistical concepts to all of the original data, they might have reached a conclusion quite different from that in dispute.

Aircraft Propellers of the Future

THE requirements of propellers of the future probably will be along the following lines:

1. Propellers must have automatic synchronizing control systems, capable of holding all engines at the exact same speed, thus relieving the pilot of the necessity of keeping the engines synchronized by manual manipulation of the propeller r.p.m. control levers. This system probably will be solved most efficiently by use of electrical control means.

2. Propellers must have unlimited pitch range and should be equipped with low-pitch, high-pitch and feather-pitch stops. For flying boats of the four or more engine types in particular, there is a very good possibility that, for taxiing in the water, it will be desirable to set the inboard engines in reverse pitch having the outboard ones in normal position, thus simplifying water-handling.

3. It is believed that considerable advantage in certain types of operation will result from the use of propellers having infinitely variable fixed-pitch positions in addition to automatic constant-speed controls. This type of control may be called manual selective. With this control the propellers can be set for the desired conditions and synchronized with micrometer precision. In comparatively smooth air the synchronization will maintain for long periods of flight and any loss in engine power will be noted by a falling of r.p.m. Hence certain types of incipient engine failures may be detected long before they become dangerous.

4. Some discussion of the desirability of a pitch indication to serve as a check on power output has indicated that this instrument might be advantageous. It is believed that, although there are certain theoretical advantages to knowing the actual blade angle in flight, they are not sufficient to compensate for the additional complication of the pitch indicator.

5. For some types of operation, the use of propellers having four or more blades may become desirable in order to improve take-off in installations when high powers are used and diameters are limited.

6. For high-altitude operation, it is probable that propellers of larger diameters than have been standard practice in the past will be required. Also, for certain types of operation, it may be more economical from the weight standpoint to use engines with two-speed reduction gears, although this arrangement probably will cause some headaches to the engine people. Another possibility is the use of propellers having variable diameters. However, compared to the difficulties of designing variable-diameter propellers, the two-speed engine reduction gear is child's play.

7. An appreciable improvement in performance can be anticipated at least on liquid-cooled installations or other types with tapered nose shapes by the use of blades having shanks with improved fairing.

In summarizing it is believed that the most important changes in propellers over the next few years will be in the direction of lighter blade construction, improved constant-speed controls of the automatic-synchronizing type suitable for use on airplanes with two, four or more engines, and the acceptance of manual selective control as a useful adjunct to constant-speed operation.

Excerpts from the paper: "Trend of Controllable Propeller Requirements," by George W. Brady, chief engineer, Propeller Division, Curtiss-Wright Corp., presented at the Chicago Section Meeting of the Society, May 20, 1938.

A Practical Approach to the Road Detonation Problem

By A. J. Blackwood, C. B. Kass, and G. H. B. Davis

Standard Oil Development Co.

PRESENT methods of correlating road knock behavior of fuels with laboratory methods are unsatisfactory and should be discarded in favor of a method that makes a new approach to the problem, the authors contend. Reviewing the causes of this situation, the paper offers evidence to show that the use of mathematical averages applied to the study of the car detonation problem has been very misleading and that the so-called average fuel octane-number requirement of existing cars and its corollary, the octane requirement of the average car, are values having little practical significance.

A procedure is recommended by the authors as one approach to the problem of setting up a simple, practical way of evaluating fuels. It consists in selecting a number of privately owned cars and testing both reference-fuel blends and branded fuels in these cars without making any changes whatsoever in the engine adjustments. The results of two such surveys are reported.

COOPERATIVE road tests conducted at Uniontown in 1932 under the auspices of the Cooperative Fuels Research Committee established the generally accepted technique for conducting antiknock tests of fuels in cars on the road, and provided the basis for standardizing the present A.S.T.M. octane number as a suitable laboratory antiknock value for predicting road performance in cars. However, the automotive and petroleum industries are beginning to feel that a satisfactory solution of the problem does not now exist. During the time that has elapsed since 1932, work along these lines has been conducted actively by a number of individual groups both in this country and in Europe, and it is becoming increasingly evident that an entirely new approach to the problem must be made.

There is ample evidence showing that present laboratory rating methods do not correlate with road performance, and it is not uncommon today to find nationally branded fuels of the same A.S.T.M. octane number differing in the same car, and from car to car, by the equivalent of 5 or 6 so-called road octane numbers. This is more than half the spread between the existing levels for premium and regular grade fuels, and there have been instances of a non-premium regular grade fuel of 70 A.S.T.M. octane number giving less knock in a car than a premium priced fuel of about 78 A.S.T.M. octane number. Although it is a rare combination of car and fuel which gives such unusual performance and the whole phenomena are vastly more complicated than inferred, the point to be emphasized is that an unsatisfactory situation does exist in regard to properly correlating road knock behavior of fuels with the existing laboratory methods.

The material presented in this paper is submitted as one approach to the problem which provides a simple method of obtaining and handling the data and which may assist in arriving at a more satisfactory solution of the car antiknock-requirement and laboratory-correlation problems.

Octane-Number Requirement

Recently the term "octane-number requirement" has come into use and the annual A.P.I. proceedings[1, 2] of the last two years print figures to show the average fuel octane-number requirement of cars or, conversely, the octane-number requirement of the fuel for the average car. It has been stated that nothing has confused engineering progress more than the widespread application of the mathematical average, and although this statement may be an exaggeration, there is no doubt that the use of such averages applied to the study of the car detonation problem has been very misleading. For example, the following two groups of cars each have an average octane-number requirement of 72.3 for no knock. The performance of the individual cars in Group 1 on a 72.3 octane number reference fuel is altogether different from the corresponding results on the cars of Group 2. In this example, the "average" of the cars in Group 1 has no practical value whereas, in Group 2, the "average" would have practical value but only because the deviation from average was small.

	Octane Requirement for No Knock	Deviation from Average	Knock Intensity on 72.3 Octane Reference Fuel
Group 1			
Car A	79.1	+ 6.8	Objectionable
Car B	74.8	+ 2.5	Light minus
Car C	71.9	— 0.4	No knock
Car D	63.4	— 8.9	No knock
Average	72.3		
Group 2			
Car E	72.2	— 0.1	No knock
Car F	71.9	— 0.4	No knock
Car G	72.0	— 0.3	No knock
Car H	73.1	+ 0.8	Trace minus
Average	72.3		

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 16, 1938.]

[1] See A.P.I. Proceedings, 1936, Vol. 17, Section III, pp. 96 and 97: "Report of the Chairman of the Automotive Survey Committee," by E. W. Isom.

[2] See A.P.I Proceedings, 1937, Vol. 18, Section III, pp. 150 and 151: "Report of the Chairman of the Automotive Survey Committee," by E. W. Isom.

Table 1 – Test Data – 1937 Cars – Metropolitan Area

Car Number	Mileage	Miles Since Decarbonization	Spark Adv.	Octane Number Requirement For Indicated Knock Intensity				Knock Intensity on Commercial Fuels						Atmos. Temp.
				No Knock (0)	Trace (1)	Light (4)	Medium (8)	A	B	C	D	E	F	
1	2,109	2,109		75.7	75.0	72.5	69.2	5.0	8.0	7.0	4.0	6.0	2.5	76
2	5,047	5,047		<63	<63	-	-	0	0	0	0	0	0	71
3	1,208	-	2°R.	<63	<63	-	-	0	0	0	0	0	0	73
4	4,336	4,336	0°	81.3	79.8	74.8	67.9	8.0	8.5	8.0	8.0	8.0	6.9	92.0
5	5,743	5,743		<63	<63	-	-	0	0	0	0	0	0	78.0
6	3,394	3,394	4°A.	<63	<63	-	-	0	0	0	0	0	0	81.0
7	15,103	14,428	3°R.	66.0	64.0	<62	-	0	0	0	0	0	0	62.0
8	12,220	12,220	2°A.	77.5	76.0	72.0	66.2	1.0	5.5	4.2	1.6	3.7	2.7	72.0
9	5,879	5,879	1°A.	82.0	81.0	77.8	73.2	6	10.0	9.0	5.0	8.0	5.0	68.0
10	1,491	1,491	1°R.	75.8	74.8	72.2	68.7	2.0	4.5	4	3.0	3.0	1.5	73.0
11	4,600	4,600	-	76.4	75.0	71.0	67.0	4	6	5	3.0	4	2.4	83
12	2,066	2,066	2°R.	71.2	70.4	67.8	64.2	0	4.0	2.0	0	0	0	76
13	6,740	6,740	9°R.	64.0	63.3	-	-	0	0	0	0	0	0	65
14	4,292	4,292	4°R.	72.0	71.3	68.5	64.8	0	1.3	0	0	1.0	2.0	84
15	3,198	3,198	4°R.	73.5	71.8	66.0	-	0.5	1.5	1.5	1.0	1.5	3.0	84
16	2,412	2,412	2°R.	69.4	68.5	66.0	62.8	1.0	1.0	1.0	0	0	0	69
17	4,420	4,420	4°A.	75.9	74.3	69.2	<63	3.0	4.5	4.0	1.5	2.5	2.0	71
18	5,919	5,919		68(App.)	-	-	-	-	-	-	-	-	-	74
19	3,869	3,869	0°	<63	<63	-	-	0	0	0	0	0	0	75
20	4,248	4,248	2°R.	75.0	73.6	67.0	-	1.7	3	2.5	1.5	1.5	2	78
21	7,690	7,690	1°R.	85.0	83.0	75.5	69.7	11	10.5	7.5	6.5	8.0	6	67
22	4,467	4,467	4°R.	<63	<63	-	-	0	0	0	0	0	0	78
23	6,887	6,500	4°A.	77.5	75.0	68.8	66.2	2.0	5.0	3.5	2.0	2.0	1.7	72
24	3,265	3,000	4°A.	72.5	70.9	65.8	-	3.0	6.0	4.0	0	2.0	0.5	78
25	4,712	4,712	4°A.	79.4	77.7	72.1	64.6	4.0	5.0	6.0	5.0	5.0	4.0	85
26	1,818	1,818	6°A.	78.0	73.0	65.5	-	2.0	4.5	3.0	1.0	2.0	3.0	72
27	1,475	1,475	0°	63.0	62.0	-	-	0	0	0	0	0	0	73
28	2,061	2,061		<63	<63	-	-	0	0	0	0	0	0	83

The so-called average fuel octane-number requirement of existing cars and its corollary, the octane-number requirement of the average car, are values having little practical significance as will be shown later, and the use of the word "average" in this connection should be discouraged.

The commonly accepted definition of the term "octane-number requirement" is the A.S.T.M. octane number of that mixture of the then-existent secondary reference fuels which gives a barely audible knock when a particular car is tested while in adjustment as recommended by the car manufacturer. However, both the accumulation of carbon and the antiknock value of the fuel which the owner of the car is accustomed to use, frequently necessitate a change in the spark setting away from the original recommended setting and, in consequence, only a small percentage of cars on the road actually are in the adjustment recommended originally. Wear of parts frequently brings about appreciable changes in the spark timing even though the adjustments never may have been altered manually. It seems apparent that the fuel octane-number requirement of a car in service when adjusted in accordance with the manufacturer's advices has little meaning other than to reflect the antiknock performance with the factory adjustment plus the influence of carbon deposition, accumulation of scale in the cooling system, local hot-spots, distributor peculiarities, wear, and the like. The really important data, namely the octane-number requirement of cars as they actually exist in the customer's hands, are not obtained when the cars are adjusted before making the test.

As mentioned previously, octane-number requirements of new cars have been obtained during the past several years, but even new cars of the same make, with only sufficient mileage for proper break-in, vary considerably in antiknock performance. Distributor characteristics are often such that, when the static spark advance is set according to the manufacturer's recommendation, a wide range in spark timing occurs from cylinder to cylinder at a given speed during actual operation, and this condition varies in magnitude for cars of even the same make. In successive cars from the assembly line, there may be slight variations in cylinder-wall thickness, in compression pressures, in valve timing, in coolant circulation, and in heat-transfer characteristics of the hot-spot, and all of these factors have an influence upon the temperature and pressure of the air-fuel mixture before ignition with resultant noticeable differences in knocking. Therefore, even with new cars, a representative number of units should be tested if the results are to be considered significant.

Road Octane Number

The road octane number of a fuel is defined as the A.S.T.M. octane number of that blend of standard secondary reference fuels which develops the same maximum knock intensity as the test fuel when run in a car. The road rating is obtained by bracketing the unknown fuel between two blends of reference fuels of known octane number such that the intensity of knock falls between the intensities found with the known reference blends. The road octane number of the fuel is then reported as equal to the A.S.T.M. octane number of the reference-fuel blend equalling the unknown fuel. Aside from the fact that atmospheric conditions influence the ratings obtained, three outstanding objections to this method of evaluating fuels are:

First, in order to obtain accuracy, the engine usually is adjusted to produce a light or trace knock intensity. This procedure frequently necessitates a spark adjustment of abnormal value, and the results may be proportionately in error. This point is illustrated in the following table of data for five types of fuels, each of which was run in the same car with two spark adjustments – one to give trace knock and the other a heavy knock. Of particular interest is the fifth fuel where not only did the knock intensity change with a change in

spark advance but, in addition, the maximum knock occurred at a much higher speed resulting in almost 5 points poorer road evaluation of the fuel.

	Road Octane Number When Run With: Trace Knock Intensity	Heavy Knock Intensity	Difference
Pennsylvania Straight Run + 2 cc. $PbEt_4$	68.3	68.8	+ 0.5
California Straight Run	72.2	73.0	+ 0.8
Pennsylvania Cracked	67.2	69.4	+ 2.2
Blend Mid-Continent S. R. and Cracked	73.7	74.2	+ 0.5
Mid-Continent Cracked	77.0	72.2*	— 4.8

*High-Speed Knock

Second, the resulting rating of the fuel depends upon the relative way in which the sample and the secondary reference fuel blends behave in the car being used. Thus a given fuel may have several road octane values depending upon the car being used as shown by T. A. Boyd in his recent Annual S.A.E. meeting paper[3] reporting on the work of the C.F.R. Committee and from which the following data are selected to illustrate this point.

Car	Octane Number of Fuel WC-1	WE-1	WH-2
C	71.3	68.2	84.5
G	73.4	71.2	82.5
O	67.9	67.8	75.0
S	73.0	74.3	

Third, a road octane number is a measure of performance only when the characteristics of a given car are known or when several fuels are being compared in the same car. Road ratings of the same fuel in various cars give no indication whatever of comparative behavior inasmuch as a value of say 70 octane number in one car may have been accompanied by objectionable knocking whereas a rating of 65 octane number on the same fuel in another car may have represented only a trace knock. Further, it has been demonstrated in some cars that a fuel which knocks harder under hot conditions than under cold conditions will have a higher rating for the hot test, due to reference-fuel characteristics. In such instances, octane numbers vary inversely as performance and are no measure whatever of the relative degree of knocking under the two test conditions. Road octane numbers, unless obtained on the same car under identical test conditions, convey a very minimum of information.

[3] See S.A.E. TRANSACTIONS, June, 1938, pp. 244-252: "1937 Road Knock Tests – Report from Cooperative Fuel Research Committee," presented by T. A. Boyd.

Obtaining Data for Suggested Test Method

After considering the foregoing points of view, it appeared desirable at the start of the work described hereafter to discard the existing systems of determining both road ratings and average octane-number requirements in favor of what was considered to be a simple practical way of evaluating fuels. The procedure adopted was to select a fairly large number of privately owned cars and test both reference-fuel blends and branded products on those cars without making any changes whatsoever in the engine adjustments. If the cars selected constitute a representative cross-section of the total cars on the road, the results obtained should then be representative of actual service conditions.

Two such practical surveys were made. The first was conducted on cars in the metropolitan New York area where the level of fuel quality is perhaps higher than in most areas and where car owners in general might be expected to have their cars adjusted to a somewhat better grade of fuel. The second survey was made in the deep South where fuels are not quite

Table 2 – Test Data – 1937 Cars – Southern Area

Car Number	Mileage	Miles Since Decarbonization	Spark Adv.	Octane Number Requirement For Indicated Knock Intensity: No Knock (0)	Trace (1)	Light (4)	Medium (8)	Knock Intensity on Commercial Fuels: G	H	J	K	L	M	Atmos. Temp.
29	6,978	6,978	6°RTD.	55.0	-	-	-	0	0	0	0	0	0	70°F.
30	6,333	6,333	4°RTD.	62.9	59.0	-	-	0	0	0	0	0	0	81
31	5,653	5,653	6°RTD.	72.8	71.0	65.5	-	2.5	0.5	3.0	2.5	3.0	1.0	71
32	3,891	3,891	6°RTD.	60.5	69.0	54.0	-	0	0	0	0	0	0	70
33	9,745	9,745	2°RTD.	59.1	57.0	-	-	0	0	0	0	0	0	78
34	1,305	1,305	4°RTD.	66.8	63.0	-	-	0	0	0	0	0	0	71
35	2,510	2,510	6°RTD.	60.8	59.5	55.0	-	0	0	0	0	0	0	70
36	4,215	4,215	4°RTD.	68.8	67.0	61.7	-	1.0	0	0.5	0	1.5	0	64
37	12,202	12,202	2°RTD.	57.6	56.0	-	-	0	0	0	0	0	0	73
38	8,015	8,015	4°RTD.	61.9	61.0	58.1	54.0	0	0	0	0	0	0	60
39	5,870	5,870	1°ADV.	80.5	79.0	73.8	-	5.5	4.5	5.0	6.0	5.5	5.0	68
40	11,263	11,263	2°ADV.	86.0	85.0	82.1	76.9	11.0	10.0	11.0	10.0	12.0	10.5	74
41	1,640	1,640	1°ADV.	71.8	70.1	63.0	-	2.0	0	2.5	0.5	2.5	0	81
42	2,655	2,655	1°RTD.	73.4	71.0	63.0	-	1.0	0.2	1.5	2.0	1.5	1.0	71
43	7,804	7,804	4°RTD.	66.2	64.0	-	-	0	0	0	0	0	0	77
44	3,848	3,848	8°RTD.	66.2	64.5	59.0	-	0	0	0	0	0	0	81
45	4,035	4,035	9°RTD.	62.2	57.0	-	-	0	0	0	0	0	0	79
46	1,363	1,363	9°RTD.	55.0	-	-	-	0	0	0	0	0	0	73
47	19,826	19,826	1°RTD.	75.0	73.8	69.4	64.5	1.5	1.5	2.0	1.5	2.5	2.0	72
48	9,388	9,388	6°RTD.	71.0	70.5	63.0	-	0.5	0.3	0.5	0.5	1.5	0.5	72
49	9,665	9,665	5°RTD.	62.9	59.0	-	-	0	0	0	0	0	0	66
50	10,736	10,736	4°RTD.	69.6	68.2	63.7	-	0	0	0	0	0	0	80
51	7,095	7,095	-	69.8	68.3	63.7	57.3	2.0	0	3.0	0.5	2.5	0.5	75
52	7,412	7,412	2°ADV.	81.6	78.4	68.3	-	2.5	2.5	3.0	2.5	3.0	3.0	60
53	16,050	16,050	6°RTD.	76.3	75.0	71.0	65.5	3.5	2.0	3.0	4.0	5.5	4.0	73
54	6,198	6,198	2°ADV.	68.3	67.0	63.0	-	0	0	2.0	0	0.5	0	75
55	8,689	8,689	4°RTD.	76.0	74.5	68.9	-	1.5	1.0	2.5	2.0	2.5	2.0	79
56	4,383	4,383	4°ADV.	70.4	69.0	64.6	-	2.0	1.5	2.0	0.5	2.0	1.0	80
57	11,308	11,308	2°RTD.	65.5	64.0	59.0	-	0	0	0	0	0	0	71
58	17,078	17,078	2°ADV.	71.8	69.9	67.9	-	1.5	0	0.5	0.3	0.5	0	78
59	7,513	7,513	3°RTD.	65.5	64.2	60.1	-	0	0	0	0	0	0	70

as high in quality and where, because of certain conditions, a fairly large percentage of car owners use third-grade fuel and probably adjust their cars accordingly. It may be assumed reasonably that the combined surveys totaling some 265 cars about equally divided between the two areas give a fair country-wide picture of the existing situation.

Each survey covered only cars of 1932-1937 models and the results are presented on the basis that these models represent 100 per cent of the cars on the road. Actually this figure was nearer 65 per cent although in certain areas 90 per cent would be a better figure. The atmospheric temperature averaged the same in both surveys – about 76 deg. fahr. The spark-advance setting as found on most of the cars was recorded together with the owner's statement of the mileage since the preceding carbon removal.

The test proper on each car consisted of running a series of about a dozen fuels, six of which were branded commercial non-premium fuels, the others being blends of Standard Reference Fuels C-10 and A-5. The tests were conducted in accordance with the general Uniontown procedure – on level road – and knock intensities were noted carefully to obtain an accurate estimate of the maximum intensity on each fuel. It was possible for one test operator to equip a car with the necessary fuel-handling apparatus, and test the dozen fuels in a total of about three hours. The static spark advance of each car was determined by one technician specially trained for this phase of the work.

Tabulation of Data

The complete pertinent data for the 1937 model cars[4] only are given in Tables 1 and 2. It will be noted that the data on knock intensity are reported numerically rather than as trace, light, medium and heavy knock. This method is entirely for convenience in plotting the data, zero being at no knock, one at trace, four at light, eight at medium, and twelve at heavy knock.

[4] The cars are identified by number only, but car manufacturers who so desire may obtain the identification of their particular cars by communicating with the authors.

[5] See A.P.I. Proceedings, 1936, Vol. 17, Section III, pp. 50-53: "Analysis of Passenger-Car Registrations," by M. Ainsworth and A. L. Clayden.

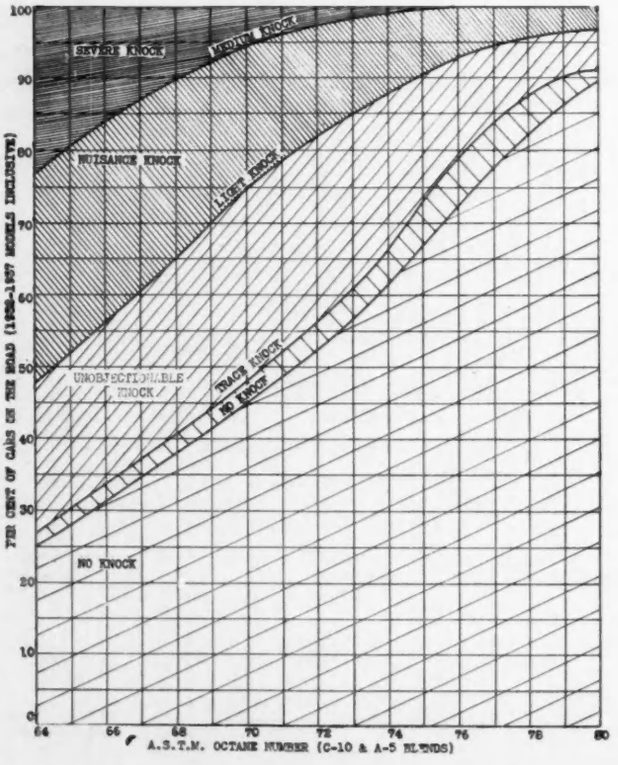


Fig. 1 – Knock Analysis – Combined Metropolitan and Southern Areas, Summer 1937

Statistical Analysis of Antiknock Requirements

As previously mentioned the data are considered on the basis of their representing 100 per cent of the cars on the road. By combining car mortality statistics as based upon the method suggested by M. Ainsworth and A. L. Clayden[5] and yearly production statistics from the trade journals, it is possible to assign, within reasonably close limits and for all practical purposes, a value to each car indicative of its percentage of cars on the road. For example, if X produced, let us say, 25 per cent of all 1934 model cars, and 1934 cars represent, say, 14 per cent of the 1932-1937 cars on the road, then the total number of 1934 X cars included in these surveys would represent $\frac{14 \times 25}{100}$ or 3.5 per cent of the cars on the road. If then, 7 X cars of 1934 model were included, each individual car would represent 3.5/7 or 0.50 per cent of the cars on the road. This method was applied to each of the 265 cars, and an individual percentage assigned to each separate car.

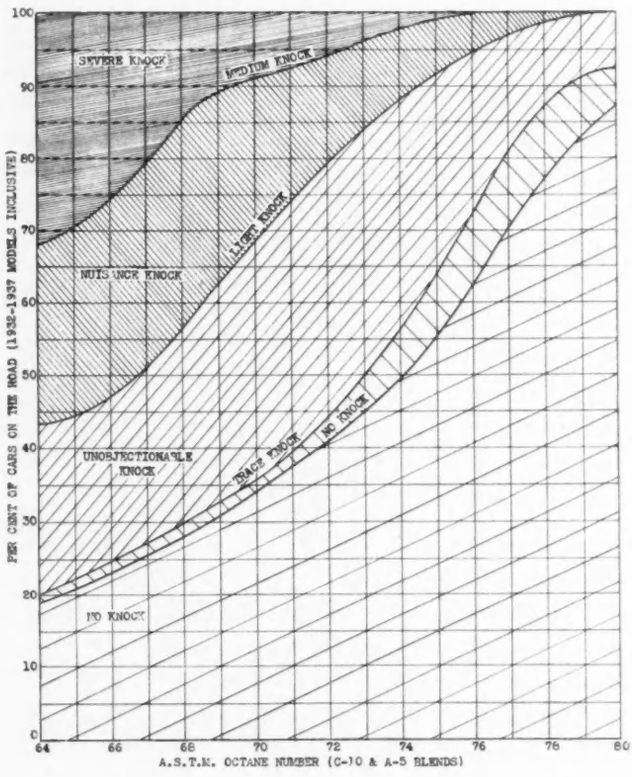


Fig. 2 – Knock Analysis – Metropolitan Area, June-July 1937

Having arrived at these values and having obtained the knock-intensity data for each car when using various reference-fuel blends, it is possible to plot a cumulative curve showing the per cent of cars on the road which will develop no knock, trace knock, light knock, medium knock, and so on, on the reference fuels of a given octane number. Such plots are presented in Figs. 1, 2, and 3, the latter two showing the Metropolitan area data and Southern area data as separate plots.

It is immediately evident from studying these plots that there is no such thing as a significant average octane-number

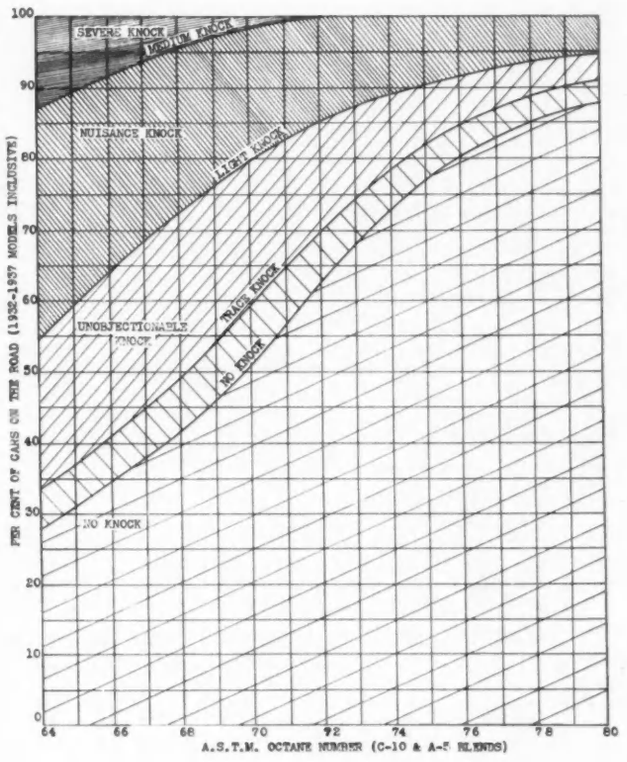


Fig. 3 – Knock Analysis – Southern Area, October-November 1937

requirement. These curves, however, permit of approximating the per cent of cars which will operate satisfactorily with a reference-fuel blend of a given A.S.T.M. octane number on any preselected basis of knock intensity adjudged to represent the "nuisance" level. The authors have chosen light knock as the probable upper limit of public acceptance. The curves indicate also what approximate improvement in market acceptance of a fuel is represented by a change of one octane number at various octane-number levels.

Spark Advance

That the driving public operates with different spark advances in different areas is evident from a study of the observed spark-advance data. Fig. 4 shows in bar-chart form the per cent of cars having various spark advances for the two separate areas. The cars in the South are in general about 3½ deg. more retarded than the cars in the Metropolitan area. The effect of this and other variations in car condition upon road performance is shown by comparing Figs. 2 and 3. Available statistics would indicate that about 20-25 per cent of the car owners in the Southern area survey were using third-grade gasoline as compared with less than 5 per cent of the owners in the Metropolitan area survey. There seems little doubt that the cars on the road are being adjusted to operate on the fuels being used. In view of this condition, it might be necessary to test cars in more than two locations in order to obtain an especially accurate indication of the performance of the cars throughout the country.

Evaluating and Correlating Branded Fuels

Several commercial fuels were included in each survey, but the same fuels were not run at both locations. It is possible, however, to tabulate the per cent of cars giving the different knock intensities and prepare curves for each fuel such as shown in Fig. 5 for the fuels used in the Metropolitan survey. It will be seen that these commercial fuels differ very markedly in the per cent of cars in which they develop a given knock intensity. Since these differences exist, a basis is formed for a practical method of relating road behavior with laboratory ratings. Basically the best laboratory evaluation should be that one which – over a wide range of commercial fuels – shows a corresponding and linear increase in the per cent of cars satisfied as the numerical value of the laboratory rating increases. Fig. 6 shows the data for the Metropolitan survey plotted as a function of three commonly used laboratory ratings of the fuels. The laboratory methods of ratings are referred to by number instead of name since the authors do not hold a brief for any one method and are presenting these data only to indicate a possible solution of the problem of correlating laboratory tests with road performance. The lab-

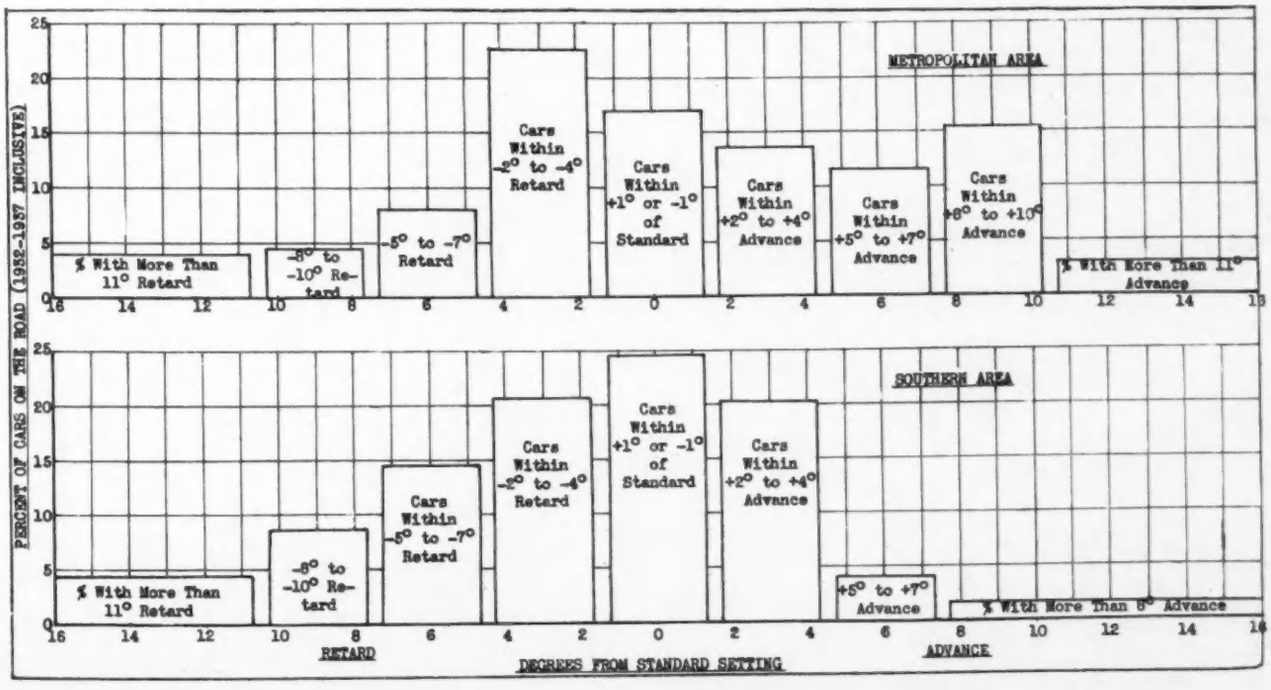


Fig. 4 – Spark Advance Vs. Per Cent of Cars on the Road

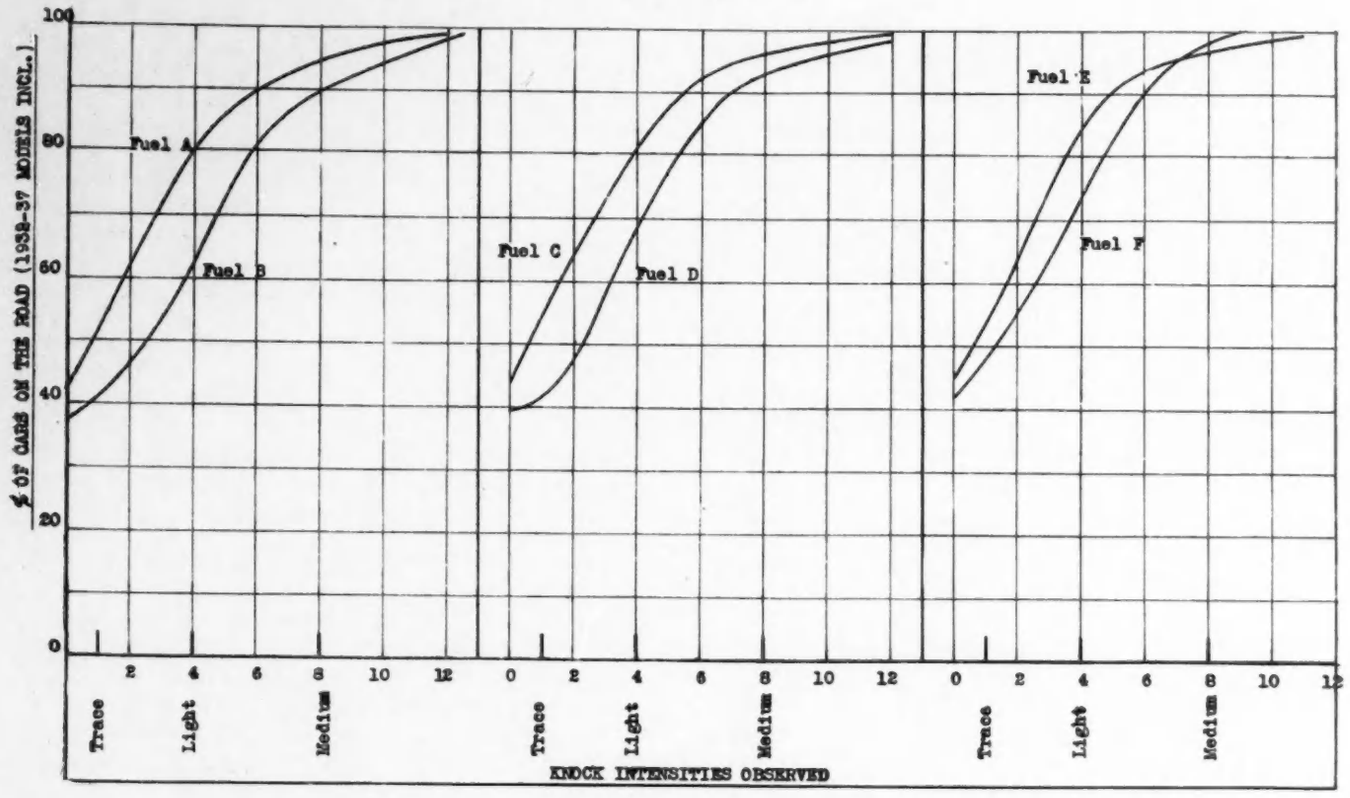


Fig. 5 – Knock Analysis – Six Commercial Non-Premium Fuels – Metropolitan Area, June-July 1937

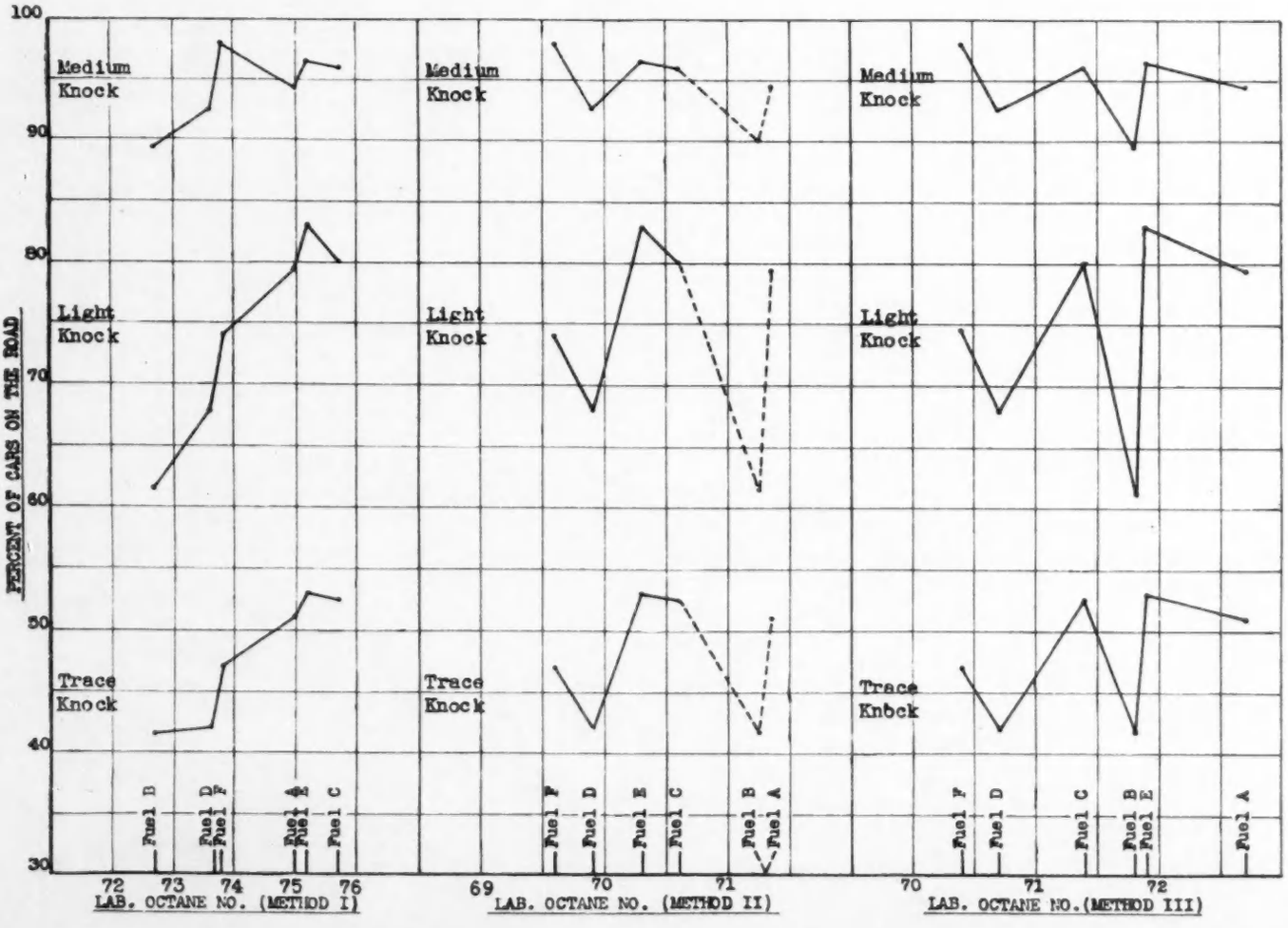


Fig. 6 – Road Knock Analysis – Correlation with Laboratory Ratings

oratory rating designated as Method 1 is very obviously the one most indicative of car performance, whereas the other two definitely do not correlate with road behavior. It is further apparent that the best correlation, as well as the most sensitive test, is obtained when evaluating fuels at light knock intensity. Fuel ratings made at too-light knock intensity may be in error due to valve and exhaust noises being mistaken for incipient knock. Fuel ratings made at too-heavy knock intensity also are apt to give erratic and misleading results. Of particular interest in this method of handling the data is the fact that those cars which do not knock on the fuel being tested are not disregarded but really form an integral part of the analysis.

In closing, it might be pointed out that there is one serious objection to this method of studying the antiknock performance of fuels due to the probability that wide differences may exist on the interpretation of trace knock, light, and medium knock. For these particular tests, the personnel were all trained by a technical man who participated in both the 1932 and 1934 Uniontown tests and who, prior to conducting these surveys, had occasion to check his interpretation of the different knock intensities with two other groups active on road detonation work. Surprisingly good agreement exists between various experienced test men on trace and light knock although, when evaluating in the louder knock ranges – which are actually of lesser importance since this entire range of intensities is unacceptable to the public – wide discrepancies may exist. If this method of test is used by independent laboratories, it may be necessary to cooperate in setting more definitely the knock intensities referred to as trace, light, medium, and heavy.

Conclusion

As a result of these two surveys, it is felt that this method of test yields practical data. Laboratory ratings are useless unless they predict relative performance of fuels in the cars actually being operated by the public, and heretofore the characteristics of the privately owned car have not been considered in attempting to correlate road behavior with laboratory results.

Discussion

Urges Knock-Intensity Standardization

– J. O. Eisinger

Standard Oil Co. (Ind.)

We have felt for some time that the antiknock demands of new cars set to the manufacturers' specifications are of little practical value to those selling motor fuels. In other words, the gasoline marketer is not so much interested in the octane requirement for which a car is designed, but what the requirement is after the car is in the hands of the customer.

During the summer of 1936 our laboratory conducted a similar survey of 70 privately owned cars. These cars were of 1934, 1935, and 1936 models and represented 98.7 per cent of all cars sold in the United States during those three years. All cars were tested as received, no adjustments being made in the course of the work. The data obtained at that time are plotted in Fig. *A*, together with the authors' octane-requirement distribution curve for the New York metropolitan area. In addition, a similar curve is shown for the 1936 and 1937 new cars (manufacturers' standard adjustments throughout) tested by members of the Automotive Survey Committees of the C.F.R. and A.P.I.

An inspection of Fig. *A* reveals that 70 octane-number motor fuels insure freedom from knock to only about 35 per cent of all cars as shown by the authors' data, or about 58 per cent as taken from the data of the Automotive Survey Committees and the Standard Oil Co. (Ind.). Actually, this picture is exceptionally pessimistic. The usual octane-requirement test, high-gear full-throttle acceleration from low speed, is a very severe test since the average motorist rarely will subject a fuel to such an extreme knock-producing situation. Many car owners, as a result of their driving methods, may never experience any knocking trouble despite the fact that the so-called octane requirements of their cars exceed the octane

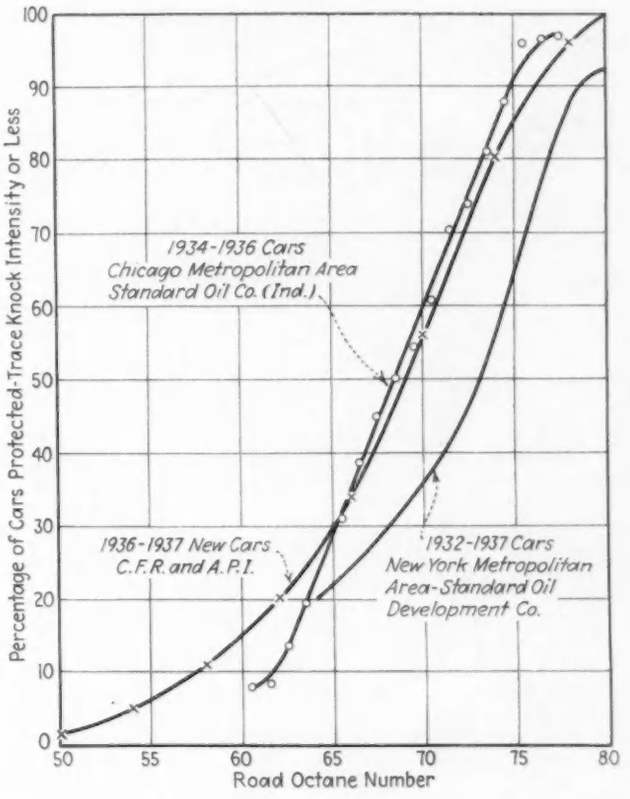


Fig. *A* (Eisinger Discussion) – Comparison of Knock Analyses

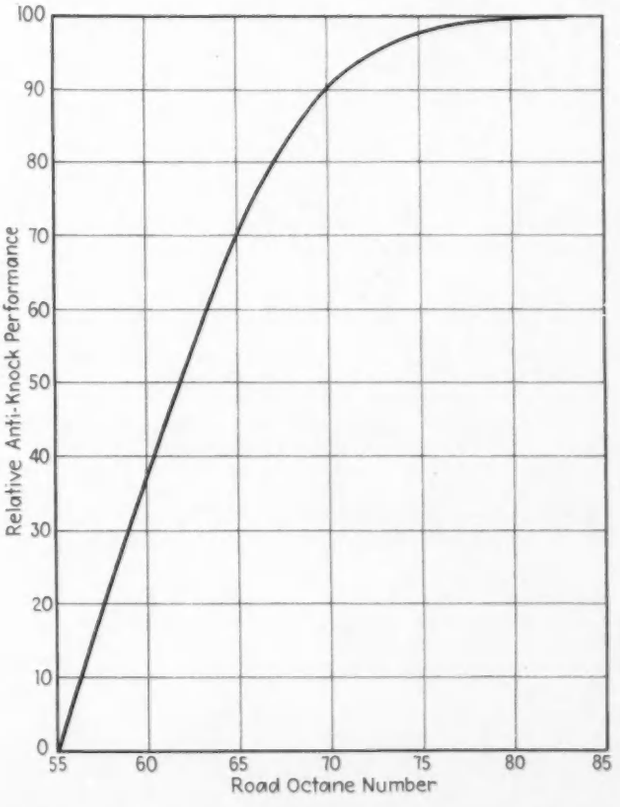


Fig. *B* (Eisinger Discussion) – Relative Antiknock Performance of Different Octane-Number Fuels

Table *A* – Relative Antiknock Values of Different Octane Number Fuels (Sea Level)

10 Per Cent Increment No.	Octane Requirement	Car Octane Requirement Minus Fuel Octane Number: Fuel Octane Number 75	70	65	60	55
1	76.1	1.1	6.1	11.1	16.1	21.1
2	74.0		4.0	9.0	14.0	19.0
3	72.4		2.4	7.4	12.4	17.4
4	70.8		0.8	5.8	10.8	15.8
5	69.2			4.2	9.2	14.2
6	67.6			2.6	7.6	12.6
7	66.0			1.0	6.0	11.0
8	64.5				4.5	9.5
9	62.8				2.8	7.8
10	59.0					4.0
Relative Knock		1.1	13.3	41.1	83.4	132.4
Relative Antiknock $100\left(1-\frac{\text{Relative Knock}}{132.4}\right)$		99.2	90.0	68.9	37.0	0.0

numbers of the fuels used by as much as 5 octane numbers or perhaps even more. Furthermore detonation, even when it occurs, is not altogether intolerable and is not regarded so by many motorists, particularly when manifested only infrequently as a slight knock under conditions of full-throttle acceleration.

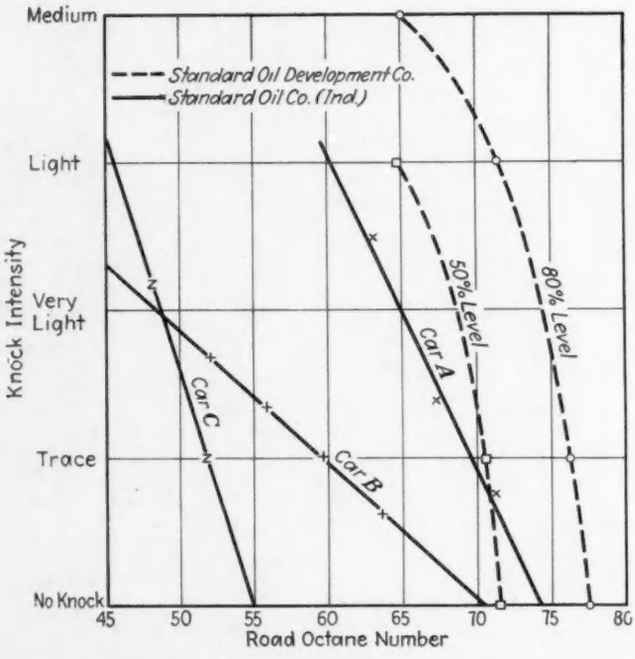


Fig. *C* (Eisinger Discussion) – Octane Requirements at Different Knock Intensities

The authors' diagrams show about 15 octane numbers between trace and severe knock intensity, that is, the octane requirement of a car based on trace knock intensity must exceed the octane number of the fuel used by about 15 octane numbers before severe knocking occurs. Even though a large number of cars will knock on a 70-octane-number fuel, it is evident that the knocking will be comparatively mild from the standpoint of intensity. A 60-octane-number fuel, however, will not only cause knocking in a much larger proportion of cars, but also will induce heavier knocking in those cars which knock only mildly on a 70-octane-number fuel. These considerations suggest that the most equitable plan for evaluating the knocking propensities of different fuels is on the basis of the relative intensity of the knocking they induce in all cars on the road. Table *A* represents our attempt to make such an evaluation. The method used is self-explanatory. Our own octane-requirement distribution curve as plotted in Fig. *A* was used to establish the octane requirement of each 10 per cent of the cars on the road. The resulting curve is plotted in Fig. *B*. Although the importance of knock intensity in this connection has long been recognized, the relative antiknock performance of different octane-number fuels may be appraised more accurately by presenting the data in this way.

R. J. Greenshields, L. E. Hebl and T. B. Rendel[a] recently brought out still another aspect of the problem by stating that part-throttle octane requirements of cars are perhaps of even greater importance than requirements determined at full throttle.

The method proposed by Blackwood et al for establishing correlation between road and laboratory ratings is an interesting and instructive corollary to their work on the relative performance of commercial motor fuels in privately owned cars. However, we sincerely question the advisability of adopting exactly their road-test technique for an investigation which has correlation per se as its exclusive objective. It is believed that the authors would have been able to duplicate their correlation results, probably with assurance of greater accuracy, had they bracketed each test fuel between two reference fuels in only those cars in which the particular test fuel gave a trace knock. It is generally recognized that tests made at heavier knock intensities are less reliable. On the other hand, we do feel that there is justification for the use of privately owned cars in rating fuels on the road. If fuels are tested using customer cars, making very little or no change in spark timing, there can be little doubt that they are being evaluated under the conditions under which motorists themselves appraise fuels.

We wish to endorse the authors' statement that any cooperative work of this kind among independent laboratories would necessitate knock-intensity standardization. Fig. *C* is an illustration of the discrepancies which may exist between the knock-intensity interpretations of different laboratories. Data taken from the authors' Fig. *A* have been superimposed on typical curves of our own showing the knock intensity induced in different cars by different octane-number fuels. Obviously, the differences noted in Fig. *A* between our octane-requirement distribution curve and that of the authors are largely attributable to different concepts as to what constitutes *trace* knock intensity. The authors' trace knock is apparently lighter than our trace knock by an amount equivalent to 4 or 5 octane numbers. We feel that we cannot emphasize too strongly the need for both the oil and automotive industries to cooperate in establishing the knock intensity or intensities at which octane-requirement determinations should be made. Furthermore, there is need to establish what knock intensity level by the conventional octane-requirement test is equivalent to an objectionable knock in the same car when driven under normal "customer" operating conditions. Standardization of the knock-intensity nomenclature (that is, trace, light, medium, and so on) is probably the first step which must be taken. Perhaps the best approach to the standardization of knock-intensity levels is to establish for an "average" car definite octane-number differences between no knock and trace knock, between trace knock and light knock, and so on.

[a] See "Antiknock Requirements of Automobiles," by R. J. Greenshields, L. E. Hebl, and T. B. Rendel, presented at the Eighth Mid-Year Meeting of the A.P.I., Wichita, Kan., May 24, 1938.

Horsepower Loss in Reducing Octane Requirement

– T. B. Rendel

Shell Petroleum Corp.

MR. BLACKWOOD, in presenting his paper, mentioned an equation for correcting the road ratings to agree more closely with their laboratory ratings by the introduction of a term involving sensitivity in the fuel. I should like to agree with Mr. Blackwood in principle as to the use of this equation, but I think it should be pointed out that we do not yet know enough about the measurement of the sensitivity of a fuel to changes in engine conditions to be able to generalize on the use of an equation involving merely the difference between the Research- and the Motor-Method octane number. After all, there is a possibility of fuel reacting quite differently to changes in manifold temperature compared with changes in speed or spark advance or other engine variables. The Research minus Motor Method of measuring sensitivity is a haphazard generality of these variables which may or may not cancel each other out.

Mr. Blackwood's analysis of octane-number requirements is also interesting. Some months ago we similarly investigated the octane-number requirements of cars, using the results of the A.P.I. and C.F.R. cooperative tests. Fig. *A*, taken from our paper to the A.P.I., shows the percentage of cars giving no knock versus the octane number. Also on this chart we have shown by circles the approximate percentage of gasoline marketed below a given octane number. It is interesting to note that these figures agree remarkably well with the percentage of cars giving no knock at that given octane number, indicating that, in general, cars are adjusted to suit the octane-number level supplied in any given district or, putting it another way around, the octane requirement

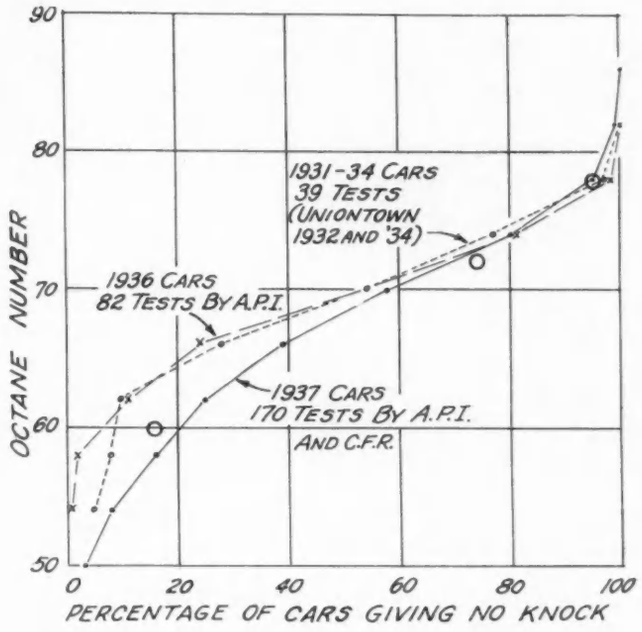


Fig. *A* (Rendel Discussion) – Distribution on Cars of Various Models

of a car is a function of the octane number of the fuel supplied to that car. In this conclusion we heartily agree with Mr. Blackwood, and we also agree with his idea that this phenomenon is caused by variations in spark-advance setting. An interesting point in this latter connection is shown in Fig. *B*. This figure shows the spark-timing degrees retard (or advance) from that required to give maximum horsepower plotted against percentage loss of horsepower obtained with that spark setting on six different cars. The main point about these curves is the remarkable similarity among the various cars and also the remarkably small percentage loss in horsepower for a relatively large number of degrees retard. Retarding the spark 5 deg. will reduce the octane-number requirement quite appreciably; yet it will only involve a loss of about

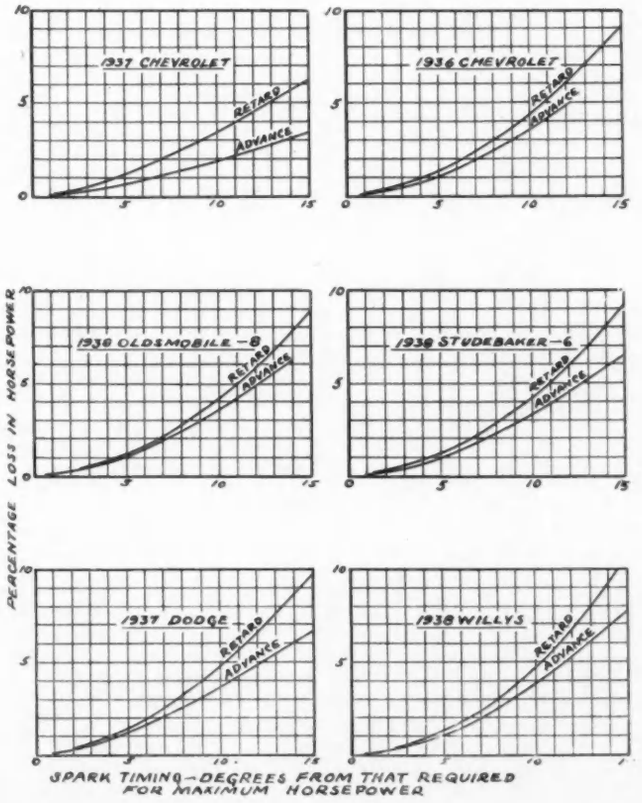


Fig. *B* (Rendel Discussion) – Changes in Spark Timing Vs. Percentage Loss in Horsepower

1 to 1½ hp. Considering the cost of raising the octane number much above present limits and the fact that this increase will be eventually passed on to the consumer, the conclusion from an economic standpoint is obvious.

Authors' Closure Amplifies Explanation of Methods

– A. J. Blackwood, C. B. Kass, and G. H. B. Davis

Standard Oil Development Co.

MR. RENDEL refers to an equation for determining the road octane rating of a fuel from laboratory ratings. (Road Octane Number $= M_1 + C\,(M_1 - M_2) + b$ where M_1 is a laboratory rating under mild conditions, M_2 is a laboratory rating under severe conditions, and C and b are experimentally determined constants for a given set of road and car conditions.) This equation does not appear in the paper but was included in the presentation as a further development along the lines of the method proposed in the paper for studying this problem. This type of equation has been found to work extremely well and, although agreeing in principle with Mr. Rendel that generalizations based on two simple laboratory ratings do not necessarily account for all the engine variables of speed, load, temperature, spark advance, and so on, the fact remains that this type of equation, with the constants worked out experimentally, fits the facts closer than any of the numerous methods which we have tried previously. Incidentally Mr. Rendel has inferred that the equation suggested is based upon the difference between A.S.T.M. and Research ratings, whereas we carefully avoided mention of any specific methods and, as a matter of fact, Research ratings were not used. We have attempted to propose a method of solution rather than the solution itself which we believe might best be based on co-operative results.

Both Mr. Rendel and Mr. Eisinger mention car octane requirements as based upon their study of the A.P.I. and C.F.R. data available in the literature. We too studied these data along the lines proposed in our paper and eventually abandoned them as being inadequate to give the answer desired – namely, a method for evaluating fuels in addition to the per cent of cars satisfied by any given octane number of the reference fuels. Furthermore, these data were practically all obtained under "adjusted" settings and not truly representative of cars as they are found in the customers' hands. It was for these reasons that we found it necessary to run three complete surveys to provide the necessary data under different operating conditions. We imagine the survey which Mr. Eisinger's company made was done for the same reasons, and it is to be regretted that the results of that survey were not made available at that time. We might mention here that the comparisons made with our data on a basis of per cent of cars are not strictly comparable in any case. Our figures were calculated on a basis of per cent of cars on the road at the time of the survey, whereas the A.P.I.-C.F.R. data apparently are calculated from per cent of total cars included in the test and the Indiana data are based on per cent of total cars manufactured over a three-year period.

Mr. Eisinger has called attention to the difference between our Metropolitan area survey and his own data and suggests that this may be due to differences in the respective interpretation of "trace" or "light" knock. We are inclined to believe that our respective test-car drivers would agree very well on trace knock and that the data actually cannot be compared for the reasons already stated. As a matter of fact, if Mr. Eisinger had compared his data with our Southern area results, the per cent of cars satisfied with 70 octane fuel would have agreed exactly. In other words, the difference in requirements between the Metropolitan area and the Southern area results – all obtained by the same test crew – are no greater than the difference between our Metropolitan area results and the Indiana or A.P.I.-C.F.R. results.

Since these differences between various areas actually exist, and since the range of octane number available from third grade up to premium fuel is substantially the same the country over, we cannot subscribe entirely to the thought that the car octane-number requirement is a direct reflection of the octane number of the fuels available as suggested by Mr. Rendel. Other factors such as economic conditions, density of traffic, condition of highways, whether the country is flat or hilly, whether seasonal temperature changes are great or small, and others, all have their influence on the level of octane number at which the public elects to set cars. It is probably much the same set of circumstances which dictates the ratio of third grade to regular grade to premium grade fuels in various parts of the country.

On the question of the importance of part-throttle knock, we are in agreement with the discussers. A survey of how customers accelerate revealed that only about 10 per cent consistently accelerated at full throttle, but the remaining 90 per cent occasionally also accelerated at full throttle. There are, of course, some cars which, under certain conditions, knock more at part throttle than at full throttle.

Spark Advance and Octane Number – A Road-Test Technique

By W. E. Drinkard and J. B. Macauley, Jr.

Research Engineer *Staff Engineer*

Chrysler Corp.

THE road-test technique outlined in this paper, the authors explain, was developed to find a method that would give a broader picture of the car-fuel relationship. Although conceding that the general method employed – using the knock die-out point as a datum in the study of fuels in road cars – is not new, they claim that their technique increases the speed and precision in the method of obtaining borderline advance, and offers data over a wider field than hitherto presented.

A fully manually controlled distributor, a spark-advance indicator, and thermocouples for indicating the necessary operating temperatures comprise the apparatus used, the paper reports. The authors suggest that a more accurate and dead-beat means of indicating car speed than the car speedometer be employed, such as a fifth-wheel electric tachometer.

The general procedure consists in accelerating the car in high gear from approximately 10 m.p.h. and noting the speed and the spark advance at which audible knock disappears, thus locating one point on the curve. The spark advance is then altered by about 2 deg. and the acceleration repeated to locate another point, and so on until enough points are located to complete the curve.

The paper follows with notes on testing; correlation of borderline knock data with data obtained on a knock-intensity basis; a discussion of octane requirement and road rating of fuels; and concludes by pointing out the usefulness of this technique in evaluating car variables.

IN 1932 at Uniontown, the C.F.R. road-test method for rating fuels was developed. In 1936, with the initiation of the C.F.R. Motor Survey, the car octane-requirement method was developed by rather direct modification of the fuel rating technique. The data accumulated during several years of cooperative road tests under these methods have been subjected to careful and painstaking study, but the results have not been altogether satisfactory. The thought arises that perhaps the trees have obscured the woods – that a technique which would give a broader picture of the car-fuel relationship would prove of value. The method here presented is suggested as such a technique.

The method is not new. Prior to the first Uniontown tests, the Ethyl Gasoline Corp. laboratories had suggested the use of the knock die-out point as a datum in the study of fuels in road cars. In January, 1937, Hebl and Greenshield[1], in a discussion presented at the S.A.E. Annual Meeting, suggested the use of borderline knock curves in studying the fuel-car relationship. The present contribution merely adds speed and, we believe, precision in the method of determining borderline spark advance, and offers data by this method over a wider field than that hitherto presented. We believe that the results warrant further investigation of the possibilities of the method.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, June 16, 1938.]

[1] See S.A.E. TRANSACTIONS, April, 1937, pp. 148-150.

The Method

The apparatus used in making the tests here reported consists of:

1. A fully manually controlled distributor, together with a suitable control for operating it from the driver's seat in the car. This device can be made readily from a standard dis-

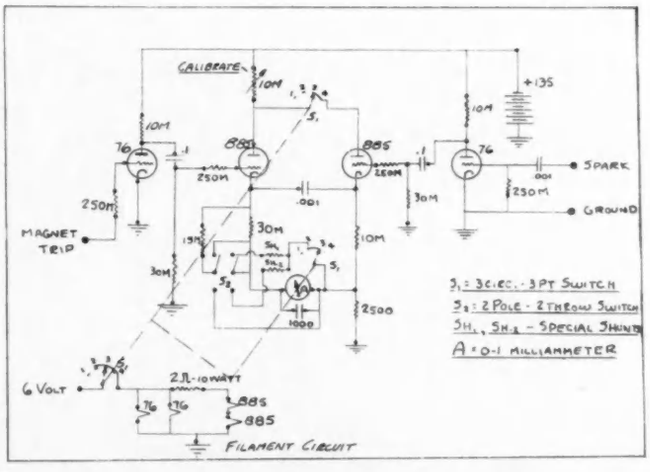


Fig. 1 – Revised Circuit for Spark-Advance Indicator

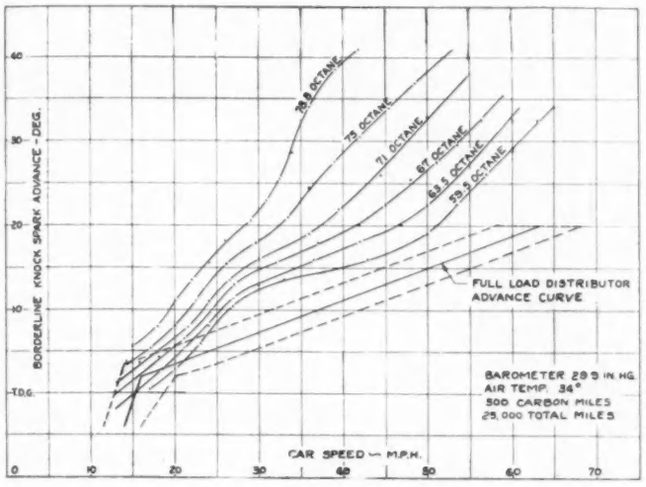


Fig. 2 – Borderline-Knock Spark-Advance Curves for Blends of C-10 and A-5

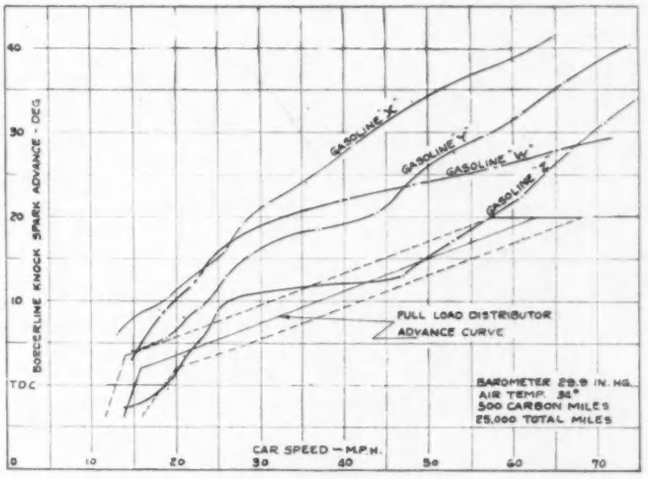


Fig. 3 – Borderline-Knock Spark-Advance Curves for Three Commercial Gasolines *X*, *Y*, and *Z* and Experimental Fuel *W*

tributor by removing the governor weights, locking the cam to the distributor shaft, and disconnecting the vacuum line to the engine.

2. A spark-advance indicator of the type described by MacGregor[2] or Oldberg.[3] The circuit of the instrument described in the latter paper has been modified slightly, and the revised circuit is given in Fig. 1.

3. Thermocouples suitably installed to indicate the necessary operating temperatures.

4. It is suggested, though not employed in these tests, that a more accurate and dead-beat means of indicating car speed than the car speedometer be used. A fifth-wheel electric tachometer might serve to advantage.

A driver and an observer constitute the test crew. The general test procedure consists in accelerating the car in high gear from approximately 10 m.p.h. and noting the speed and spark advance at which audible knock disappears. This procedure locates one point on the curve. The spark advance is then altered by about 2 deg. and the acceleration repeated to locate another point. This procedure is repeated until enough points are located to complete the curve. Several accelerations usually are made at each spark setting for the sake of accuracy. The driver calls the speed at the point of knock die-out, and the observer records the spark advance at the same point. This double-check on the point of knock disappearance very rarely leads to a disagreement between the observers of more than 2 m.p.h., and usually the agreement is exact. Operating temperatures are recorded at appropriate intervals. With the instrumentation used and a level course fairly free from traffic, data on one fuel can be obtained in about 15 min. The quantity of fuel required for a test on one of the smaller cars is 1 to 2 gal. – usually 1 gal. suffices.

The following notes on testing are offered:

1. When accelerating with high fixed spark advance, the low-speed portion of the acceleration should be made at part throttle to avoid producing abnormal combustion-chamber temperatures due to excessive detonation.

2. Determination of knock die-out point is more difficult on the "sensitive" type of fuel.

3. In case of an appreciable wind, it is best to run all tests in the same direction; otherwise both temperatures and rate of acceleration may alter enough to affect the test results.

4. These tests were all made under level road conditions. There seems to be no reason why hill tests on a uniform grade should not be satisfactory, though the reduced rate of acceleration would be expected to produce some shift in spark advance due to greater temperature build-up.

Examples of Data

Fig. 2 shows car speed-borderline knock curves for six blends of secondary reference fuels varying from 59.5 to 78.8 A.S.T.M. octane number. The data were obtained on a small six-cylinder car having 25,000 total service miles and 500 miles since carbon cleaning. The general curve pattern is typical of reference-fuel blends. Superimposed on these curves is the specified distributor advance curve for this model, with the recommended basic setting. The full line represents the desired curve, the dotted lines the allowable production tolerances.

Fig. 3 shows car speed-borderline knock curves for three commercial fuels and one experimental fuel on the same car, with the standard distributor advance curve superimposed. Gasoline specifications are shown in Table 1. Since a fuel will knock if the spark advance exceeds the value for borderline knock, these curves may be used to predict fuel knock in this car.

Correlation of Borderline With Knock-Intensity Data

Suspecting that the use of a fixed spark advance might affect the knock die-out point due to excessive detonation prior to the disappearance of knock, the following data were obtained on one car:

1. Borderline knock curves of three commercial fuels: *A*, a premium grade; *B*, a regular grade; and *C*, a third-grade fuel.

2. Actual *road* spark-advance curves of the standard distributor used on the test car.

3. Knock intensity-car speed data on these fuels using the standard distributor at three initial advance settings.

Fig. 4 shows the borderline knock curves for the three fuels, with the distributor curves at the three settings superimposed. From Fig. 4 we can obtain data for a composite curve in which car speed is plotted against excess spark advance above borderline knock for each fuel at each initial distributor setting. These curves are shown in Figs. 5*a*, 6*a*, and 7*a*.

Fig. 5, for Fuel *A*, shows at the top the spark advance beyond borderline knock plotted against car speed and, in the

[2] See "Spark-Advance Indicator," by J. R. MacGregor and K. R. Eldredge, presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 7, 1937.

[3] See S.A.E. TRANSACTIONS, November, 1937, pp. 521-526: "A Spark-Advance Indicator and Knock-Rating Observations," by Sidney Oldberg, Gilbert Way, and J. B. Macauley, Jr.

Table 1 – Gasoline Specifications

Fuel	Gravity	Distillation Temperature					Octane Number		Tetraethyl Lead, cc. per gal.
		5 Per Cent	10 Per Cent	20 Per Cent	90 Per Cent	End Point	A. S. T. M.	Research	
A	....	114	127	157	350	395	76.8	....	....
B	....	117	130	156	356	401	69.0	....	....
C	....	172	187	210	359	422	52.4	....	None
D	63.1	122	137	164	354	390	70.9	71.0	2.2
E	61.5	108	121	149	348	390	69.2	75.0	None
X	....	111	120	137	321	386	75.5	....	....
Y	....	108	117	269	339	395	70.1	....	....
Z	....	130	147	176	359	403	55.7	....	....
W	59.3	107	120	152	359	396	69.1	78.8	None
A-5	61.5	...	191	...	341	...	40.4	....	None
C-10	67.5	...	165	...	240	...	78.8	....	None

lower curve, the observed knock intensity against the same abscissa. Considering that the two curves represent the results of quite independent tests, the agreement seems quite good. Similar curves are presented for Fuel *B* in Fig. 6, and again the agreement is good. In Fig. 7 the same curves are given for Fuel *C*. Here we have a case of high over-advance. There is a general agreement in shape between the two curves, but the die-out point shows the effect of over-advance, hence our note under "Procedure" concerning acceleration at high spark advances. The curves given are typical – much more data of a confirmatory nature were collected than can be presented here.

Data Applied to Car-Fuel Relationship

Octane Requirement – Currently, most car manufacturers recommend a basic initial distributor setting, with the further

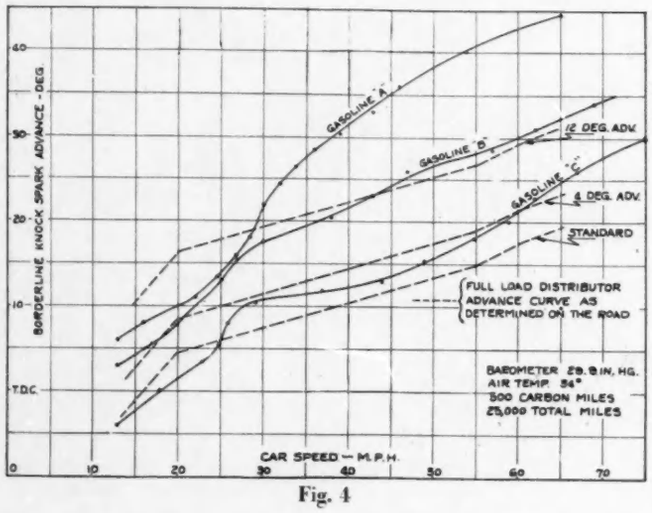


Fig. 4

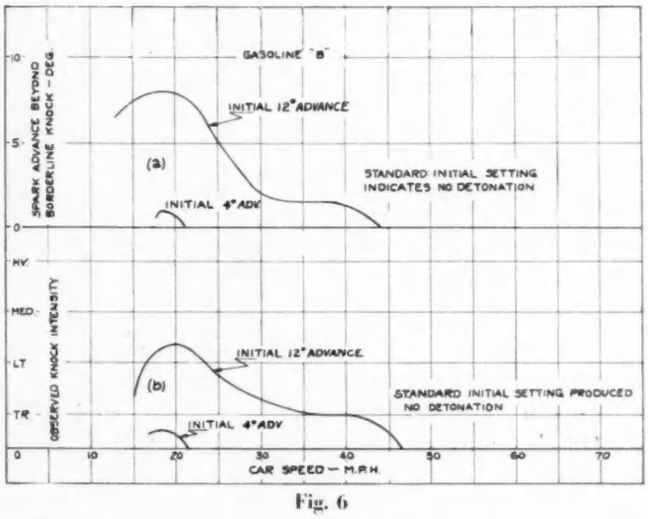


Fig. 6

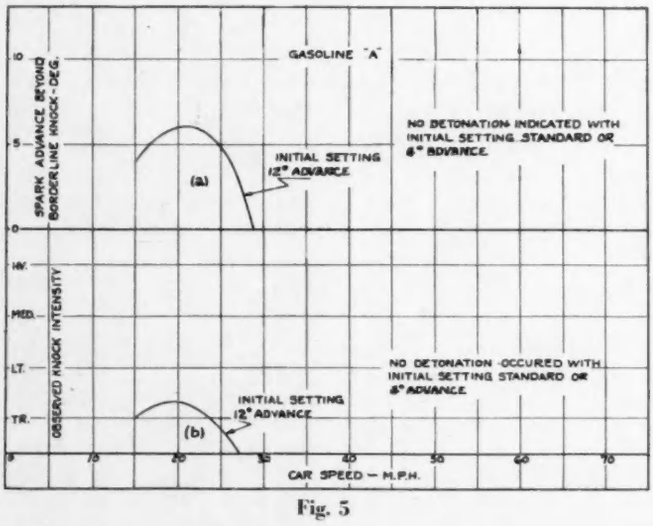


Fig. 5

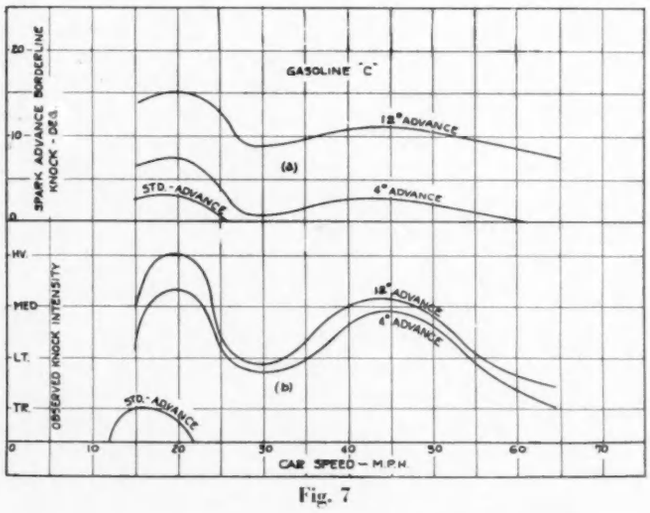


Fig. 7

Fig. 4 – Borderline-Knock Spark-Advance Curves for Three Commercial Gasolines *A*, *B*, and *C*

Fig. 5 – Composite Vs. Observed Knock-Intensity Data for Gasoline *A*

Fig. 6 – Composite Vs. Observed Knock-Intensity Data for Gasoline *B*

Fig. 7 – Composite Vs. Observed Knock-Intensity Data for Gasoline *C*

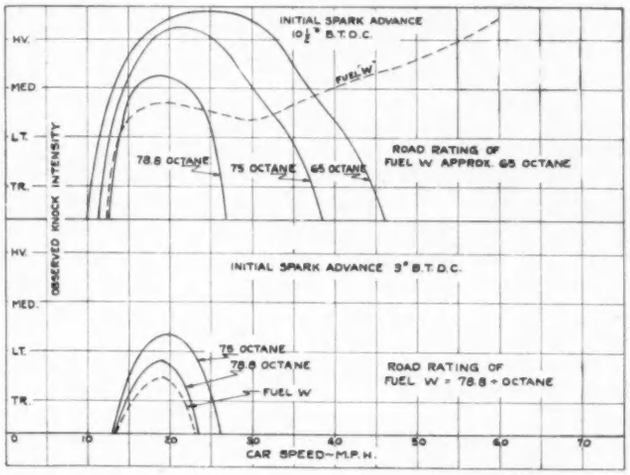


Fig. 8 – Effect of Excessive Spark Advance on the Road Rating of One Fuel

instruction that this setting should be altered according to fuel and operating conditions such that a slight "ping" is heard on full-throttle acceleration. This is a rational and necessary procedure, since some accommodation must be made for differences in temperature, altitude, fuel, and engine condition.

If we refer back to Fig. 2, the effect of production distributor tolerances may be predicted. The solid spark-advance line is the desired curve. The dotted lines represent the necessary production tolerances. On the car in question, octane requirement from lower than 59.5 to about 72 could be obtained by using various distributors, all within production tolerances and with the same initial setting.

The effect of this situation upon a survey of car octane requirements might prove peculiar. Thus, if a number of cars of similar model with identical engines, distributor characteristics, and initial distributor settings were checked throughout the country, a considerable spread would result due simply to weather conditions, altitude, and engine condition. On the other hand, if the distributors of these cars had been adjusted by the owners to produce slight "ping" and the cars were checked in this condition, the "octane requirement" would be nothing more than a road rating of the fuel that the owner had been accustomed to using, in terms of reference fuels. Any actual survey of cars *as found* must include some of both of these classes plus a large third class where the distributor setting is a function of changing car condition and owner inertia.

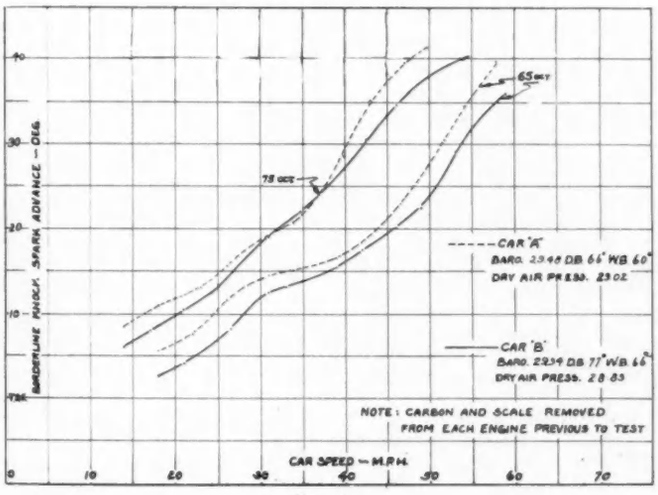


Fig. 9

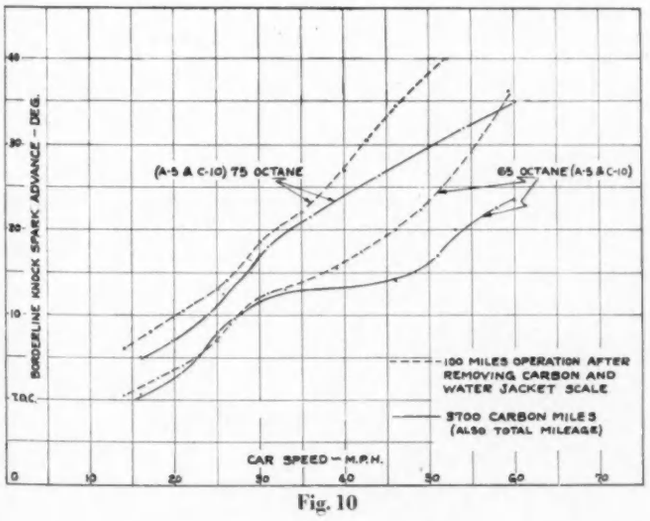


Fig. 10

Fig. 9 – Differences in Knocking Tendency Between Two Similar Cars

Fig. 10 – Effect of Engine Carbon and Water-Jacket Scale on Detonation in One Car

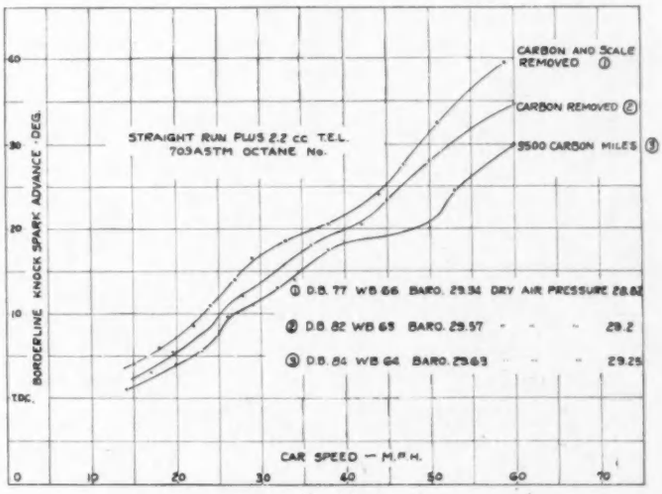


Fig. 11

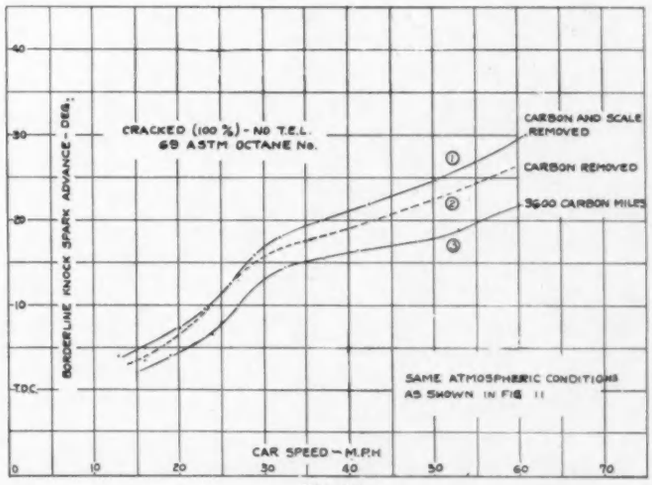


Fig. 12

Fig. 11 – Effect of Engine Carbon and Water-Jacket Scale on Detonation in Same Car – Using Gasoline *D*

Fig. 12 – Effect of Engine Carbon and Water-Jacket Scale on Detonation in Same Car – Using Gasoline *E*

Confronted with such considerations as these, grave doubts as to the practical value to the car manufacturer of such average or statistical studies are bound to arise. The position of the car manufacturer and the petroleum refiner differ on this point. The car manufacturer is interested in the performance of fuels in properly maintained vehicles, and quite properly disclaims responsibility for poor performance in cars not so maintained. The refiner, on the other hand, is interested in the performance of his product in all vehicles, regardless of their maintenance.

Road Rating of Fuels – Road rating of a gasoline in terms of reference-fuel blends is at present based upon maximum knock intensity regardless of the car speed at which maximum knock occurs. This method is rational since knock intensity is the objectionable feature. It is doubtful that a more satisfactory alternate method using borderline knock data could be devised, although one laboratory is using a variant of the method to obtain closer discrimination between fuels than can be obtained conveniently by the bracketing method.

From time to time the validity of fuel ratings at abnormally high spark advances has been questioned. The tendency of the slope of the curve of borderline spark advance against speed to decrease at the higher speeds seems to be related to the "sensitivity" of the fuel. This tendency also seems to increase with engine carbon deposit. In a case of this sort an increase in spark advance might bring about an approach to the conditions shown in Fig. 8, in which the conditions were made extreme. Here at 3 deg. initial advance the Fuel *W* rates 78.8 whereas, with 10½ deg. initial advance, it would rate about 65 in terms of A-5 and C-10 blends.

Applications to Car Variables

Preliminary studies indicate the usefulness of this technique to the automotive engineer in evaluating:

1. The influence of engine design changes upon fuel requirement.
2. Effect of degenerative changes due to operation, such as combustion-chamber deposits, water-jacket scale, and so on.
3. Effect of altitude and climatic conditions.

Differences between road and dynamometer conditions have been recognized generally. Bartholomew, Chalk, and Brewster[4] have called to our attention the differences in fuel behavior in a steady running engine and in one which is accelerating. This technique makes it possible to study under normal accelerating conditions the spread in knocking tendency of similar engines of the same model, free from the errors introduced by the distributor. Fig. 9 shows the variation in borderline knock on two cars of the same model, running on two blends of A-5 and C-10 (65 and 75 A.S.T.M. octane number). Although weather conditions were not the same, the spread is in the opposite direction to what might be expected from the atmospheric data. Both cars had been run less than 7000 total miles and had had carbon and water-jacket scale removed just prior to the test. Fig. 10 shows the effect of removing both the carbon and water-jacket scale which had accumulated in 3700 miles of operation. The fuels were the same reference-fuel blends just referred to – namely 65 and 75 octane.

The effect of such engine deposits upon knocking tendency of different fuels may be studied quite completely by this method.

In this series of tests a six-cylinder car having approximately 3500 carbon miles was rated with two types of fuel:

1. A straight-run gasoline plus 2.2 cc. tetraethyl lead to give 70.9 A.S.T.M. octane number with zero "sensitivity" (Gasoline *D*).
2. A cracked unleaded gasoline of 69 A.S.T.M. octane number and a "sensitivity" of nearly 6 (Gasoline *E*).

Borderline knock curves on each fuel were run on the car (3) as found, (2) after cleaning carbon, and (1) after removal of water-jacket scale by an acid bath. Fig. 11 shows the results on the straight-run fuel, and Fig. 12 on the cracked fuel.

Conclusion

In general this technique shows possibilities which should warrant giving it a more extensive trial. In these preliminary tests, it has proved reasonably rapid, and the results seem reasonably reproducible. It appears to offer a very useful tool for an attack on the "severity" problem, and we hope that it helps in the effort to produce cars of more stable detonation characteristics with respect to mileage.

[4] See S.A.E. TRANSACTIONS, April, 1938, pp. 141-156: "Carburetion, Manifolding, and Fuel Antiknock Value," by Earl Bartholomew, Harold Chalk, and Benjamin Brewster.

Discussion

Contrasts Viewpoints of Oil and Automobile Industries

– J. O. Eisinger
Standard Oil Co. (Ind.)

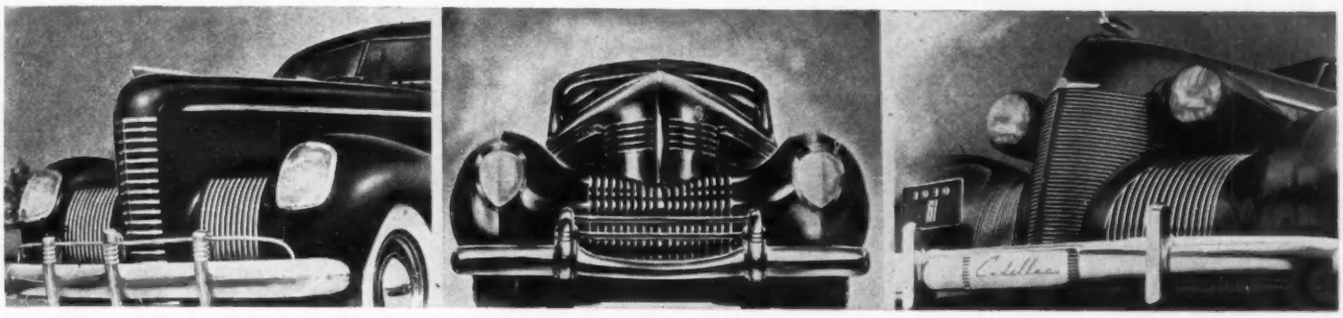
THE instrumentation and method suggested by the authors as a road-test technique merit further study in applications which will establish both their range of usefulness and their limitations. It is our thought that such a tool will simplify the problem of designing engines so that, at all speeds and under all operating conditions, every advantage is taken of fuel antiknock quality consistent with an adequate margin of knock protection. The authors' preliminary studies relating to combustion-chamber deposits and other degenerative changes are of interest to both the oil and automotive industries as well as indicative of the type of information such a test procedure makes available.

We want to amplify this statement of the authors which appears under the heading, "Octane Requirement":

"Confronted with such considerations as these, grave doubts as to the practical value to the car manufacturer of such average or statistical studies are bound to arise."

We appreciate, as do the authors, that there is considerable difference between the viewpoints of the automotive and oil industries as to what represents the most desirable octane-requirement information. For example, we are all interested in the octane requirements of new cars as received from the factory. We are interested also in the octane requirements of the cars which drive into gasoline service stations day after day. Whereas the former requirements perhaps will be considered the most valuable by those who manufacture and sell automobiles, the latter requirements may be deemed of first importance by those who produce and market motor fuels. A statistical survey of the octane requirements of cars *as found*, despite certain shortcomings, appears to be an expedient means for indicating to the oil industry what the needs of their fuel customers really are. Although results are influenced by many of the variables mentioned by the authors, such as the effect of owner inertia and changing car conditions on distributor setting, we feel that it is only by the inclusion of these variables in a reasonably comprehensive statistical study that customer needs are appraised intelligently.

We do not wish to infer that we are not interested in the fuel requirements of new cars, or in the influence of design changes on these requirements, and so on. On the other hand, we feel that such information more vitally and directly concerns engineers in the automotive industry who may use it to advantage for improving engine design and operation. In this connection we note that the authors' Figs. 2, 3, and 4 show that, for any given octane-number fuel, a much more ample margin of knock protection is provided in the high-speed range than for low-speed operation. We would appreciate a few comments from the authors as to the difficulties involved in providing a full-load distributor advance curve which imposes uniform full-throttle requirements on fuel quality throughout the speed range of a car.



Nash Chrysler Cadillac

Trends in Design of 1939 Cars

By Austin M. Wolf
Consulting Engineer

THE many improvements in the new models can be classified under safety, comfort, economy and styling – with accent on safety. Improved vision, lower center of gravity, more effective controls, instruments and indicators that truly impart a message, are a few of the results of the industry's major interest.

Roomier bodies, sunshine tops, improved suspension, easier controls, and better ventilation assure comfort. Economical operation is brought about by weight reduction, improved carburetion, further adoption of the improved overdrive, and the special finish and treatment of engine parts. Advanced styling shows a smooth unity of form initiated at the tip of the front, in its unbroken sweep of the rear, concealed door hinges, the removal or skirting of running boards, and a balanced ornamentation of metal trim.

Of special interest are the Chrysler speedometer, Buick inbuilt directional signal, Oldsmobile clutch and rear suspension, Warner Gear overdrive, the numerous steering-column shifts, Borg and Beck clutch plates, Chrysler Custom Imperial fluid flywheel, crankshaft-mounted fans, Cadillac and Pontiac variable-rate rear springs, Cadillac Hi-Plane Hotchkiss drive, Chevrolet front suspension, Packard rear cross-link shock absorber, Studebaker and Nash ventilating systems, new Diesels, and the Super-Twin coach.

General

Among the new models are the Ford "Mercury" 95-hp. V-8 on 116-in. wheelbase, the Hupmobiles with revised Cord bodies and four- and six-cylinder engines and the Pontiac "Quality 6." The Oldsmobile "60," which uses the same body as the Pontiac Quality 6, is also a new model. which weighs 225 lb. less than the 1938 six-cylinder model. Cadillac has discontinued the "65" series because of its competition with the "60" Special. The previous "60" series has been changed to "61" to agree with the Cadillac factory numbering system. The Packard-8 resumes its previous designation, the "120." The "Super-8" now incorporates the "120" chassis and utilizes the same body shell further reinforced and trimmed more luxuriously. The Overland, Plymouth, and Dodge wheelbases have been increased to 102, 114, and 117 in., respectively. The Buick "40" wheelbase has been dropped 2 in. to 120. The Pontiac "25" and Oldsmobile "60" have a 115-in. wheelbase; the wheelbase of their remaining models is 120 in. Chevrolet is introducing two new body types, a four-passenger coupe on the Master DeLuxe chassis and a station wagon for both its chassis.

Weight reductions are numerous, some typical examples follow: Pontiac "25" and "26" four-door touring sedans have curb weights of 3125 and 3300 lb. respectively as compared with 3430 lb. for last year's 6. The "28" weighs 3380 lb., a saving of 120 lb. over the 8 of 1938. With a 50-lb. body increase, the Oldsmobile "70" is still 75 lb. lighter than the previous 6, and the "80," with a curb weight of 3525 lb., is 100 lb. lighter than the old 8.

Engines

The Oldsmobile "60" six-cylinder engine differs from the "70" series only in the stroke, being 3⅞ in. instead of 4⅛ in., giving a displacement of 216 cu. in. and developing 90 hp. at 3200 r.p.m. Compression ratio is 6.2:1 as against 6.1:1 in the "70." At 1600 r.p.m. the torque outputs are 170 and 180 lb-ft., respectively. The Chrysler Royal engine develops 100 hp. at 3600 r.p.m. as a result of boosting the compression ratio to 6.5:1. In changing the stroke from 4½ to 4⅞ in., and incidentally applying full-length water jackets, the Imperial displacement has been increased to 323.5 cu. in. and the compression ratio boosted to 6.8:1; 135 hp. is developed at 3400 r.p.m. as against 110 hp. with the shorter stroke. This engine now powers both the Imperial and Custom Imperial. An 8-hp. increase is obtainable with an optional aluminum head in the latter. Packard compression ratios, in general, have been decreased slightly. Overland has increased the compression ratio to 6.35:1 (6.81:1 optional) and, with Aluminum Co. of America T-slot, cam-ground, tin-plated,

heat-dam pistons, has boosted the output to 61 hp. at 3600 r.p.m. with 106 lb-ft. torque at 2200 r.p.m. The Hudson "112" has increased its horsepower from 83 to 86 by using the 1¼-in. carburetor in place of the 1⅛-in. size.

Cylinders and Valves

All Packard engines other than the 12 have cast-iron heads. Last year only the 6 used them. The "120" head has been redesigned entirely with resulting greater turbulence. The 6 retains a compression ratio of 6.52:1. The "120" has a compression ratio of 6.41:1 as against 6.6:1 in 1938. The Super-8 has a 6.45:1 as against 6.5:1. Whereas optional ratios of 7.05:1 prevailed in all three engines, the new optional heads are designed for ratios of 6.85:1. The Graham head is of cast iron except when the supercharger is used, in which case it is of aluminum. The six-cylinder Oldsmobile head was altered in the area directly over the piston to minimize "ping." Its weight has been decreased through the use of lighter sections, but strength has been maintained by incorporating stiffener braces at various effective points.

The Graham supercharged engine uses 1½-in. Thompson XCR steel exhaust valves, whereas the Model 96 uses 1¼-in. silchrome. With the use of austenitic steel in the exhaust valves and proper seat inserts, ACF has had repeated cases where engines have run over 100,000 miles without the valves being touched. The Wilcox-Rich self-locking adjusting screw is now used by all Chrysler products. The Wilcox-Rich hydraulic lifter is used by Cadillac; LaSalle (Cadillac "50"); Lincoln; Lincoln Zephyr; White Models 24A, 20A, and 30A; Franklin aircraft and truck engines; Seagraves; Continental A-50 aircraft engine; and Lycoming industrial. To decrease the possibility of camshaft and tappet-base scoring, all Cadillac tappet-valve bodies are coated with ferrous oxide. By the use of new tools, a finer surface is imparted to the cam end of the Chevrolet tappets which are now of the solid cast-and-drilled type. The lower ends of the push-rods have spherical surfaces instead of the flat plane conical type previously used. The bushings are inserted in the valve rocker arms, the valve-stem pad ends are subsequently hardened and then, locating from the hardened ends, the bushings are diamond-bored. Buick uses malleable hardened rocker arms on all series and without bushings. In all the Cadillac V-8s, the fuel-pump eccentric has been relocated to oppose the camshaft unbalance and a counterbalance weight has been added to the camshaft sprocket to balance the assembly. Packard is using a Tocco-hardened malleable-iron camshaft. Chrysler products incorporate the "Superfinish" as applied to cylinder-bores, pistons, piston-pins, crankshaft main and pin bearings, camshaft bearings and contours, valve tappets and stems, and the flywheel face. A non-whipping chain with special-shaped joint pins is produced by Morse Chain Co. and used on Packard, Cadillac, LaSalle and Pontiac. Morse also has developed a heat-treated cast-iron camshaft sprocket having more than twice the hardness of standard iron sprockets.

Pistons and Rings

The two top rings in most American passenger-car engines have been coated with ferrous oxide. In all Cadillac V-8s the piston top land is now wider, removing the top ring farther from the heat of combustion. The top compression ring is 3/32 in. wide and deeper than S.A.E. standard. All four rings are ferrous-oxide coated. Material has been added to the piston bosses and head of the Cadillac-16 to increase piston strength. Cadillac, Chrysler, and all Packard compression rings are made of Perfect-Circle M alloy, primarily developed for Diesel and aircraft use. The Pontiac ⅛-in. compression rings have been replaced by 3/32-in. rings. Oldsmobile has done likewise and increased the radial thickness. The top land is 1/32 in. thicker (7/32 in.) and, to facilitate break-in, the compression rings are plated with a phosphate coating. Sealed Power has introduced the Hi-F oil ring in single- and double-slot types. Increased life of liners, pistons, and rings has been obtained by ACF in adopting a cam-ground T-slot Lo-Ex alloy piston. With the wet-type heat-treated iron cylinder liner of 450 to 500 Brinell, ACF has had instances of wear of only 0.002 to 0.003 in., mileages varying from 150,000 to 200,000. The Buick piston dome is undercut 1/32 in. for relief, all ring lands are reduced in diameter, and the upper end of the skirt is beveled.

Crankshaft and Ventilation

The Pontiac six-cylinder engine used in Models "25" and "26" has a crankshaft in which the diameters of the main and connecting-rod bearings have been increased ⅛ in. A packing has been added to the rear main bearing between the oil throw and the flywheel flange and also at the front end within the timing-chain cover. Heavier front and rear enclosing plates on the balancer weight have replaced the outer crimped ring and inner ferrule formerly used. Chevrolet has replaced the flat leaf springs in the harmonic balancer by a pair of annular soft-rubber rings each having six raised bosses on one face. When the two rings are placed over the six studs riveted to the hub, the six bosses of rubber form a bushing around each stud, thus floating the balancer weight and insulating it from the hub. The Chevrolet front-end oil seal of the spring-loaded rawhide type replaces the previous plain cork seal. The previous oil slinger behind the harmonic balancer has been omitted. The Oldsmobile "80" engine utilizes the weight of the cooling fan which it mounts on the front end of the crankshaft as an inertia member in the form of five unsymmetrically spaced aluminum blades riveted to a steel spider. The unit has formed tubes which are hydrogen-brazed into the spider hub to provide the seats for the customary balancer tuning springs. The square pins located in the crankshaft pulley driving the water pump pass through the balancer tuning springs in the usual manner.

All Cadillac V-8s now are provided with a velocity suction system. Air enters the oil filler through an air cleaner as heretofore and, after filling the crankcase, passes upward into the valve-tappet compartments. Instead of continuing into the engine intake system, the gases are led rearwardly through a conduit in the vee alley into a passage way cored behind the engine rear bulkhead and ahead of the flywheel. This passage way leads downward to the bottom of the engine at which point an outlet fitting discharges the gases beneath the car. The louvre at the rear of the Buick rocker-arm cover is omitted, and a breather-type oil-filler cap with copper gauze is used.

Lubrication

The floating type of screened oil inlet has been adopted for the Packard-6, "120," and Super-8. The Pontiac oil-pan baffles are now integral with the oil-pan body. In the Cadillac V-8s the oil pan is shorter and deeper than before, concentrating the supply and reducing aeration of the oil under all conditions, particularly when the oil pan is overfilled. The AC Kleer-Kleen oil filters (S type and L-1 type) utilize a solid one-piece filtering element called Igneonite. They are built with drawn-steel cases which are zinc-plated on the inside. An internal ball check at the outlet prevents reverse oil flow through the filter should the inlet and outlet connections be reversed through error. Motor Improvements has developed a type-DP final-stage fuel-oil Purolator for Diesels being capable of removing all particles down to and including those which are only three to four millionths of an inch in diameter. In the Chevrolet engine, oil for the valve rocker-

shaft lubrication is now taken from the high-pressure side, assuring a constant supply under all normal conditions. Proper oil pressure is maintained by means of a metered bleed hole in the distributor body through which excess pressure is relieved to the low-pressure side. The distributor is provided with a pipe plug for connection in the low-pressure side to allow oil-filter installation.

Cooling System

The physical sizes and core constants of most radiators remain approximately the same. The changed grilles provide an adequate air supply with no drastic radiator changes. All Cadillac V-8s have lower and wider radiators to suit the new hood contours. Increased ruggedness permits raising the filler-cap relief-valve pressure to 5½ lb. per sq. in., raising the boiling temperature to 229 deg. fahr. at sea level. The tube-and-fin core construction supersedes the former cellular type with 3⅜-in. cores having 9½ tubes per in. The cooling efficiency is the same as with the previous type. The header is of double thickness at each end, and there are reinforcing struts in the header plate. Lock seams secure both top and bottom tanks to the headers, and struts tie together the front and back of the top tank. The latter no longer overhangs the core at the rear and its shape reduces aeration of the water and decreases expansion. New fan brackets and belts are used with the lower cooling-fan location.

The lower grille opening in the Lincoln Zephyr has promoted the cooling efficiency by better registering with the fan contour. The Mercury-8 crankshaft-mounted six-bladed fan has three blades with a 7⅜-in. tip radius and three with an 8⅜-in. radius. The spider to which the blades are riveted has a centering flange giving a ¼-in. bearing on a hub member and driven therefrom by frictional contact with a rubber ring at each side. Maximum static and running unbalance is kept within 0.5 oz-in. Balance is obtained by riveting a washer on the spider side of a blade as required. The Ford DeLuxe fan is also placed on the crankshaft. In the Oldsmobile "80" engine, the crankshaft-mounted fan draws air through a 3½-in. core having a frontal area of 405 sq. in. The rear face of the engine is in the same location as in the "70," thus pushing the core forward and necessitating one that is wider than its height due to the hood ledges at this point. The frontal area of the "60" and "70" cores is 425 sq. in. with a 2-in. depth. The cell size is different in each case.

In the Pontiacs, last year's eight-cylinder fan is now used on both the 6 and 8. The down-flow cellular type of copper core has a depth of 2 in. on the "25" and "26," and 3 in. on the "28." The frontal area in each instance is 324 sq. in. A pressure cooling valve to replace a radiator cap permits 5 lb. per sq. in. pressure before the vent opens, this valve being standard on the "28" and optional on the "25" and "26." The Packard-6 and "120" fans have corrugations on the trailing edges of alternate blades to reduce fan noise. The Super-8 fan is 1-in. smaller. The gravity flow of the radiator is 38 gal. per min., an increase of 2.7 gal. per min. over 1938. The water pump is shorter; the pressure vent in the 12 increases the normal boiling point by 12 deg. fahr. Superseding the porous-metal bushings, the new Chevrolet pump is equipped with the double-row New Departure "sealed-for-life" bearing. Supplanting the original chevron packings is a self-contained spring-loaded molded-rubber composition seal. To provide adequate cooling of the Master "85" with the 3.727:1 axle ratio as compared to 4.222:1 on the Master DeLuxe, it has a larger fan and different size core. For removal of the under-hood air and supplementing the louvres in the fender skirt, a tunnel is formed by bulging out the skirt at the dash where the air is exhausted into the heel of the fender. "Thiokol" synthetic rubber is used by L. H. Gilmer Co. for V-belts. It also is being used in timing cover gaskets, Okonite ignition cable, gasoline-tank grommets, and Chrysler products' filler-neck hose.

Fuel System

The Carter WD-0 dual carburetor with climatic control is used on the Buick "40" and Oldsmobile "80," whereas the model WA-1 is used on Pontiacs and Oldsmobile "70." The Stromberg carburetor on the Cadillac V-8s has been simplified by eliminating a number of small drilled passageways to reduce the hazard of dirt or gum obstruction, and the former electric choke has been superseded by one actuated by exhaust-manifold temperature. The automatic choke on the large Buick engine is built into a housing integral with the Stromberg AAV-26, utilizing manifold vacuum, a thermostat spring, and an offset choke butterfly. The Packard 6 and "120" with 1¼-in. Chandler Grove and 1-in. Stromberg Duplex carburetors respectively, have the automatic-choke mechanism built in. The six- and eight-cylinder Hupmobile engines are Carter equipped. With the increased engine angle, the Buick intake-manifold main branches are horizontal, the short branches being offset to suit.

To suit the changed hood compartment, the LaSalle and Buick intake silencers have been redesigned. The Packard 6 and "120" have deeper resonance chambers in their oil-bath-type air cleaners. The De Soto fuel pump is of the inverted type for cooler operation. The Chrysler-built cars other than Plymouth have an 18-gal. tank instead of one of 16 gal., with the two steel sections seam-welded together. The Chrysler Imperial tank has been increased to 21-gal. capacity. In the gasoline gages of all G.M. cars, the contact which is moved over the resistance coil is now carried directly by the tank float arm instead of being driven indirectly through gears. More economical operation has been obtained by enlarging the carburetor balance vent hole in the Chevrolet from ⅛ to 3/16 in. To eliminate the necessity of increasing the tension of the choke-valve closure spring and to assure positive closing of the valve, the throttle shaft is set slightly off-center in the air horn. For manufacturing and assembly reasons, the riser sleeve in the inlet manifold is molded in the carburetor heat insulator. Ribbing of the top and bottom pans of the gasoline tank has increased its resistance to internal as well as external pressure. The filler-cap tightening cam on the end of the filler neck is now pressed in place inside the neck instead of being flanged over, spot-welded, and soldered. The Oldsmobile filler neck, instead of being clamped to the frame, is supported from the tank by a top- and bottom-welded brace and is now on the left side as a result of a canvass for preference. The carburetor choke "hot-box" is located directly opposite the exhaust port outlet of cylinders 4 and 5 on the 6. The new shielded location results in quicker operation of the choke valve and the fast idle. A small step on the inside bend in the intake-manifold branches supplying cylinders 2 and 5 results in the liquid particles being thrown out into the mixture stream with more uniform fuel distribution to each of the two end cylinders.

Exhaust System

The Graham supercharged engine is equipped with dual exhaust manifolds and dual mufflers. The Oldsmobile muffler shells are made of terne plate and the muffler mounting is by means of flat clamps in place of the previous flange supports. All Cadillac mufflers are of the three-pass type with a 40 per cent reduction in back pressure at high speed and the double-wrapped terne-plate shell is of 0.050-in. thickness as compared to 0.036 in. For further resistance, the shell is secured to the headers by spun-over lock seam joints instead of being welded. Each end of the muffler is supported by

new-type rubber-bonded insulators, whereas the tail-pipe is held by a clamp supported by two-ply friction duck encompassing a rubber grommet. The inlet and outlet elbows are of seamless tubing and leave the mufflers through the ends instead of through the sides. The Chevrolet shell end and rim also are spun over and the head thickness is reduced from 0.050 to 0.037 in. The Pontiac exhaust manifold has been redesigned to save weight, a one-piece design replacing the former two-piece design.

Electrical System

All Cadillac engines are now equipped with 10-mm. sparkplugs. Sterling Cable has introduced a seven-strand conductor of stainless steel to replace the conventional 19-strand copper high-tension cable. The Chrysler Imperial has additional spark-timing control located on the instrument panel within reach of the operator. The Buick distributor shaft has a tang which drives the oil pump and is pinned to the distributor gear. This part replaces the previous second pin that drove a slot in the oil-pump shaft, and assures complete engine stoppage with a frozen oil pump. In the Buick steering-gear-ignition lock, the latter portion is located high enough to prevent striking and causing injury to the knee or shutting off the ignition.

The gross maximum charging rate of the Cadillac "75" and "90" generators is 30 amp. Buick generators deliver up to 36 amp. The Studebaker President and Commander generators have an output of 29 and 28 amp. respectively. The Electric Auto-Lite 4½-in. third-brush ventilated generator has a maximum of 35 amp., and the cut-in speed has been reduced to 800 generator r.p.m. To increase generator belt life the Cadillac V-8 generator pulleys have been redesigned so that the belt pulley contact is on the inside surface of the belt instead of along the V-shaped sides. The 16 generator is now mounted high above the engine vee at the front and is driven by a short belt from a pulley just behind the fan-driven pulley.

An increase in the Chevrolet distributor vacuum-control travel from 14 to 16 deg. contributes toward greater fuel economy at part-throttle operation. The distributor vacuum-control pipe has been increased from ⅛ to 3/16 in. and, with a stiffer spring back of the vacuum-control diaphragm, causes the distributor to follow more accurately the required timing change. A single bolt secures the distributor and control assembly to the block and, at the same time, acts as the pivot for the octane selector setting.

The General Motors cars and the Studebaker President are equipped with a new Delco-Remy starting-motor overrunning clutch. A minimum mass is subject to overrunning and the heavy outer shell, formerly integral with the pinion, is now a part of the sleeve and collar assembly splined to the armature shaft. The wedging cam angles are located in the outside collar so that centrifugal force tends to disengage the rollers as soon as the load upon them is released. This arrangement results in quieter operation and lessens the possibility of clutch seizure and a consequent throwing out of the armature windings. The Buick starter ring-gear teeth have a 20-deg. pressure angle. The "40" vacuum starter switch is mounted on the carburetor and comprises a stainless-steel ball that can be cammed upward at an angle by rotating the throttle shaft, raising a plunger with a W-shaped contact spring to close the solenoid relay circuit. Upon starting, the engine vacuum raises the ball and allows the plunger to return and open the circuit.

The Studebaker 15-plate, 95-amp-hr. battery is located under the hood. By placing the filler hole cap over the vent, thus closing it, an air-lock condition occurs when the solution level reaches the bottom of the air vent extension thus preventing over-filling. Delco-Remy has developed a line of farm-tractor batteries and electrical equipment in which the design guards against penetration of dust or moisture into the working parts, and with the internal construction suitable for sealed, unventilated type of operation.

Engine Mounting

The Packard Super-8 engine has a five-point rubber mounting. At the front there is a center mounting below the vibration damper to carry the weight and an inclined mounting at either side to absorb any rocking force. The rear mounting consists of the customary inclined pads at each side of the transmission case. The front motor supports in the Cadillac "50," "60" Special, and "61" have been moved inward and lower on the "50" and "61," reducing the torque reaction speed from 8 m.p.h. to less than 5 m.p.h. The new rear support as well as the front ones are non-adjustable. Buick mountings are redesigned completely and are of the same shape and size for all models, the "40" and "60" rubber being softer.

Clutch

The Chevrolet diaphragm spring clutch, built by the Inland Division, has been adopted by Pontiac. The Buick "40" has a similar-acting one-piece conical spring, excepting that it has a corrugated surface. It bears directly at its center against the rounded face throw-out bearing. All Buick linings are now die-grooved. Angular radial vanes cast on the rear face of the Chevrolet pressure plate circulate air outward through openings in the cover adjacent to each slot. The clutch pilot is of the roller type.

Borg and Beck has developed a driven plate in which there is a large reduction in the polar moment of inertia, materially improving gear-shifting, which is particularly important in the prevalent remote-control type of gearshift. Instead of the usual disc between the facings, there are only the cushion members which are riveted to a small-diameter disc carrying the hub springs. The cushion members or springs are riveted to extend radially outward from the center disc between the facings and the wave-type springs are riveted to each facing on alternate waves in the spring. In the case of the Buick "40," Olds-8, Dodge, De Soto, and Chrysler, the cushion spring is practically symmetrical with its riveted base. In the Oldsmobile-6 an unsymmetrical spring segment is used. The average deflection rate of the cushion is more uniform if the ends are nearly symmetrical, but this uniformity is obtained at the expense of a somewhat higher deflection rate during the initial compression of the disc. The best cushion design depends upon the particular car characteristics. Some cars show a greater tendency to chatter under light torque than others, in which case a very soft initial cushion is desirable, which is obtained most easily by the use of high and low crowns with non-symmetrical ends. On the other hand, a car may exhibit its worst chatter tendencies with heavy torque and moderately high plate load, in which event it is desirable to have the deflection rate as low as possible with the disc heavily loaded, and the more symmetrical type of cushion meets this condition, since both ends stop work at nearly the same time. Since the initial deflection rate is somewhat stiffer, the final deflection rate will be somewhat softer for the same total cushion and the same total carrying capacity as the spring with non-symmetrical ends. In the new Cadillac driven disc, eight waved cushions are integral with the center plate and they are symmetrical, with the friction discs riveted to alternate waves. The greater uniformity of contact permits the LaSalle clutch diameter to be reduced to 10 in.

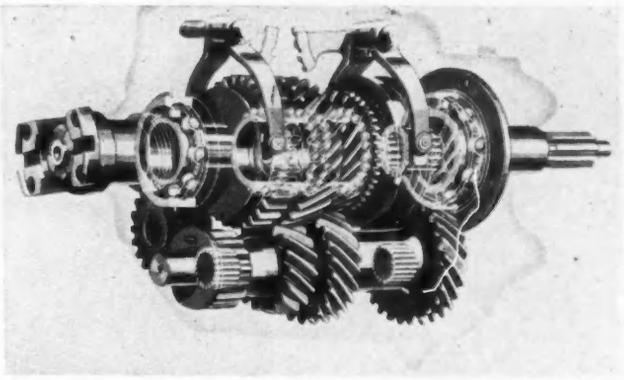
The hub springs in the Oldsmobile driven disc are re-

tained by turned-up ears in the center hub. By altering the shape of the fingers on the throw-out levers and considerably increasing the finger-face contact area, they now ride directly on the graphite ring. The line of contact of the curved surface is sufficiently broad to cover its full annular width. The improved ventilating characteristics decrease operating temperatures. The size of the graphite ring and the lubricating reservoir assembly has been increased considerably. A baffle has been added through the center of the clutch to prevent oil vapor reaching the driven disc. The new pedal linkage decreases the pedal pressure from 34 lb. to 29 lb. With the use of rubber pads on the Chevrolet pedals, the push-rod pads are plain stampings welded directly to the push-rods. The latter are fastened to the pedal levers with a nut and bolt, the proper position being maintained by serrations cut into both members at the joint. The previous pedal chain is now made a single link. Over-center clutch tension springs have been adopted by the Chrysler line. A metal closure between the clutch fork and flywheel housing replaces the previous Pontiac fabric design. The pedal effort is passed through an auxiliary shaft pivoted between the frame and the flywheel housing, preventing engine movement from affecting the clutch pedal. Buick pedals are frame-mounted.

The Olds-6 driven plate is held to a friction lag (friction built into the assembly between the hub member and the side-plate members to assist in damping transmission rattle) of 0.010 to 0.025 in. on a 7½-in. radius at 70 ft-lb. torque. The Buicks, Dodge, Chrysler, and De Soto are held to 0.030 to 0.050 in. Borg and Beck has a production fixture as well as a laboratory apparatus for determining the foregoing. If the up-and-down load curves are plotted against deflection, the resultant chart is very much like a hysteresis curve of the magnetic flux in an electric motor. The Chrysler Custom Imperial is equipped with a fluid flywheel.

Transmission

Gearshift control under the steering wheel is practically universal, either as standard or optional equipment. In the Chrysler products with the exception of the Plymouth Road King, the shifting shaft is in the form of a tube within the steering mast. Pontiac, Nash, Oldsmobile, Hudson, Chevrolet, Packard, Graham, Studebaker, and Buick mount it parallel to the column. Pontiac, Oldsmobile, Buick, and the Chrysler products use a flexible cable for selecting. Studebaker provides a bell-crank at the foot of the shifting shaft and rod control. In the Packard design, selection is made at the base of the shifting tube with a rod running from each of the two levers to two oscillating shafts in the transmission cover. The Pontiac system is retained in principle and is also used by Buick and Oldsmobile. The selector shaft within the Pontiac shifting tube is bent at right angles and terminates in an integral ball, forming a ball joint within the hollow tubular shift lever into which it snugly fits. Movement of the shift lever in neutral raises or lowers the selector rod. A threaded fulcrum bushing provides a height adjustment of the shift lever in the neutral position. In the Buick control the selector rod is also offset, but is secured to, and within, the channel-section shift lever by a pin. At the base of the tube the motion is conveyed to the cable by means of a bell-crank assembly. In the Oldsmobile control the selector cable is actuated directly by the selector rod by means of a coupling below a bracket supporting these two members. The cable returns upwardly along the column for a very short distance. In the Chrysler products the transmission shift lever slides transversely for selecting one or the other of the shift rails. A spring normally presses it toward the second-high rail but is overcome by the flexible cable when under compression by the hand control. The bottom shift lever is



Packard-6 and "120" Transmission

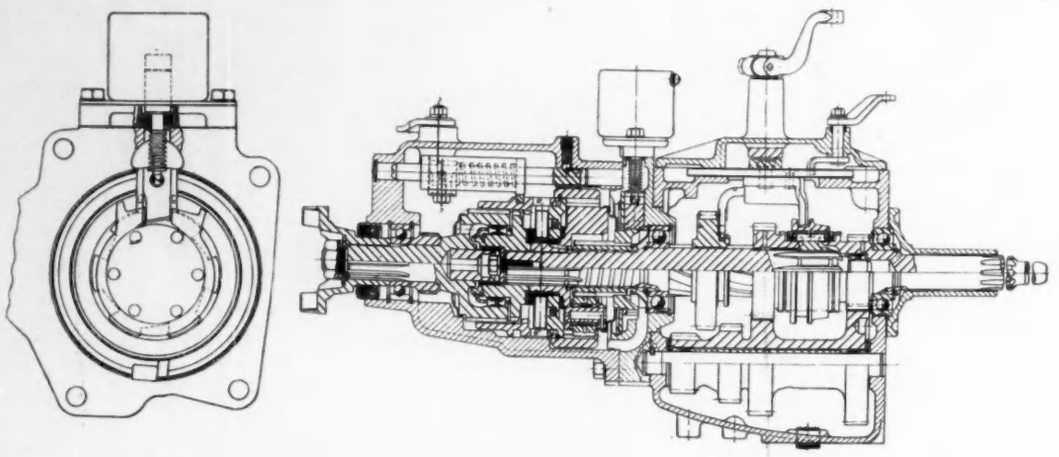
in the form of plate incorporating a segment always covering a pin at the column end of the cable.

Chevrolet provides an optional vacuum shift developed by Bragg-Kliesrath in conjunction with the Chevrolet engineers. The control tube centers a selector rod, and a bell crank at the base is rod-connected to a selector bell crank on the top of the transmission. A lever on the base of the shifting tube is rod-connected to a reactionary lever system on the transmission whereby the gear shifting is done by the piston of a vacuum cylinder of the air-suspended type. The lever system enables the driver to "feel" the shifting action. Ports in the tubular piston-rod each side of the piston are controlled by a sliding piston valve therein with atmospheric and vacuum passage ways and actuated by the reactionary lever system.

A Warner Gear shifting mechanism in the transmission side cover is such as to permit the use of any type of remote control. In the Studebaker application the selector lever carries a bent arm on its inner end and the tip engages a notch in either of the shift rails. When in the notch, that particular rail is held fixed and the other is free to slide. The shifting lever acts through a walking-beam type of linkage which is laid directly on the flat shift rails. Whichever rail is free to move will be shifted, the other one acting as a fulcrum for the walking beam while being held fixed by the selector finger.

The Pontiac type of transmission with the centrally located sliding low gear, built by the Buick Division, has been shortended by 9/16 in. A larger cageless roller bearing is used for the main shaft, and lubrication of the countershaft bronze bushings is assisted by taking advantage of the natural flow of lubricant from front to rear outside the countershaft cluster and returning through the spirally grooved bushings on its flow forward. A light pressed-steel cover caps the case top, and the entire shifting mechanism is within the case with a round rail and its fork at each side. A selector bar mounted crosswise has two cams which are so spaced that, when one engages its shifter fork, the other one clears its fork. In the meantime, the selector bar engages a notch in the shift rail which is at rest, thus locking it in its neutral position. The transmission height is considerably reduced, allowing the lowering of the floor at this point. This transmission is used on the Buick "40," Pontiacs, and Oldsmobiles. All these transmissions utilize a booster spring to ease the pressure required to bring the gears into mesh. The Buick and Pontiac gear-shifting control mechanism can be adjustably swung around the steering column to permit the shifting lever to be brought within the operator's desirable reach.

All Packard transmissions now simulate that of the 12 in which all gears are in mesh and are of the helical type except for the reverse spur train with its sliding gear on the main shaft. The first-speed main-shaft gear is mounted on two ball bearings similar to the previous second gear mounting.



Studebaker Transmission and Overdrive Unit with Solenoid Control

Lubricant is supplied under pressure to these bearings which also acts as a silencing damper. A new interlock on all Cadillac transmissions is of the positive type and permits a reduction in the stiffness of the ball detent spring. A total travel of the transmission lever has been reduced from 8 in. to 6 in., and the neutral position has been brought 1 in. closer to the operator. To reduce friction in reverse, a steel-back babbitt thrust washer and bearings supersede the previous bronze bushings and washer used for the reverse idler gears.

The Warner Gear overdrive used by Studebaker, Chrysler, De Soto, Nash, Graham, Hupmobile, and Packard now provides a means for cutting out the overdrive and returning to direct at the will of the operator. The forward end of the sun gear is carried in a circular member having large notches in its periphery. A pawl engages one of these notches and can be withdrawn by means of a solenoid built by Electric Auto-Lite to release the sun gear in order to drop back into direct. The accelerator linkage incorporates a switch for energizing the solenoid when the accelerator is pushed beyond the wide-open position. The solenoid has two coils, one powerful enough to withdraw the pawl, and the second one is merely a holding coil serving to keep the pawl out against the comparatively light return spring. To relieve the load on the sun-gear pawl, momentary torque reversal is obtained by breaking the ignition circuit for two or three explosions. As soon as the solenoid plunger carrying the pawl reaches the outer end of its stroke, it automatically opens the main solenoid circuit leaving only the holding coil circuit on. When the pawl reaches the end of its outward stroke, ignition is reestablished. The cutting-in speed for the overdrive ranges between 25 and 30 m.p.h. in the various cars.

Warner Gear has developed a synchronizer requiring a minimum amount of shift effort in view of remote controls. The new synchronizers employ a blocking principle using floating rings either die-cast or drop-forged in a single piece from a special hard-bronze alloy. The rings carry chamfered clutch teeth on their outer circumference, the chamfers forming the ramps to give the blocking action. The hub member is fixed axially on the transmission main shaft. To start the synchronizer rings into frictional engagement and also to limit their rotation in respect to the hub member within the limits of the blocking ramps, three light-weight thrust members are provided in slots around the circumference of the hub member. Surrounding the latter is the usual internally toothed sliding sleeve to engage one of the synchronizer rings followed by engagement of the positive clutch teeth of the selected gear. Actuation is through very light ball spring poppets or a snap ring. Initial action is light followed by a powerful blocking action and, after synchronization, the positive engagement is obtained by mere finger-tip effort on the shift lever.

Oldsmobile alone retains the automatic transmission which has been improved by softening the downward shift from third to first as the car approaches a stop. This softening is accomplished by an hydraulic accumulator that permits a slow drop in oil pressure, resulting in a gradual rather than abrupt engagement of the planetary gear sets. The oil pump has a single instead of double driving gear.

Yellow Coach and Spicer have developed jointly a Ljungstrom type of hydraulic converter for bus use in which there is a friction clutch between the flywheel and the converter, a reverse gear back of the converter, a roller clutch to disconnect the hydraulic system when in direct drive, a direct-drive positive clutch, and a fluid cooling system.

Propeller Shaft

Buick provides a transmission extension in view of the torque ball being mounted just back of the frame X-member center. A Hyatt bearing supports the front end of the propeller shaft. Mechanics universals are found on the "40" and "60," and Spicer on the "80" and "90." The propeller shaft is normally horizontal but, under heavy load, slopes slightly upward from the universal to the differential due to the low-hung layout. A transmission extension also is provided on the Oldsmobile "70" and "80" and the Pontiac "26" and "28." These assemblies extend approximately 18 in. beyond the rear face of the transmission case, permitting a shorter propeller shaft and a lower body floor. A bushing is provided at the end of the extension and it is lubricated, together with the splined portion of the main shaft immediately behind it, from the transmission. An oil seal bears on the universal joint hub. Any shortening or elongation of the propeller shaft is taken up at this splined joint. In the Oldsmobile "60" and the Pontiac "25," a similar construction is used except that, with a short end-cap, no bushing is required for the main shaft that extends approximately 5 in. from the rear of the transmission. The Oldsmobile torque arm front anchorage divides the universal joint angularity equally between the joints, insuring constant velocity at all times. Furthermore, the sliding action has been reduced from ¾ to 3/16 in.

Rear Axles

Hypoid gears with 1⅜-in. drop are used by Oldsmobile and Pontiac. The axle-shaft oil seal brings the felt washer in direct contact with a ground surface on the shaft instead

of against the bearing retainer sleeve. Cadillac uses one size of axle for the "50," "60S" and "61"; the larger design is used for the "75" and "90." These axles incorporate a compressible steel spacer between the two pinion bearings, eliminating the former shim adjustment. Bower bearings are used on Studebaker and Packard axles for pinion and differential mounting, rear wheels and front wheels. On the Packard Super-8 a needle-bearing-mounted roller contacts with the back of the ring gear if deflection reaches 0.005 in. The Buick New-Departure pinion bearings have 1/16 in. larger balls, ⅛ in. greater width, and 1/16 in. smaller inner diameter. Hudson ring gears are attached to the differential case by seven 7/16-in. bolts instead of six ⅜-in. bolts.

Brakes

Of particular interest is the adoption of hydraulic brakes on all Ford products as well as on Overland. Buick and Packard offer the NoRol unit as optional equipment. On the Cadillac "50" and "61" the master cylinder is now mounted ahead of the pedal while the front brakes are 2 in. wide.

Pontiac has changed from 12- to 11-in. brakes in the endeavor to combat fading. With the combination of Multibestos DR on the primary shoe and Multibestos L8 on the secondary, both of the molded type, it has been possible to obtain 30 to 40 per cent more high-speed stops before serious fading occurs. Recovery to normal pedal pressure after several normal high-speed stops is also accelerated. The cast chrome-nickel alloy iron brake drum is machined to 0.004-0.005-in. limits for concentricity. Before the finishing cut is taken, the drums are inspected by the magnaflux process. With the change in the Chevrolet front suspension, the brake anchor plate is attached to the steering knuckle. The hand brake lever now is located under the dash at the left side with a pull-rod running to a frame-mounted bell crank pivoted on an electroplated pin. A pull-rod extends back to an idler lever on the second cross-member from which two pull-rods run to the rear brake cables. To prevent any possibility of rattling in the rigging, tension springs are used and a slight twist is put in the pull-rods when they are installed in the car. The Cadillac hand-brake-lever ratchet has been superseded by a roller clutch which is quiet in operation and locks the lever in any position.

Wheel Rims and Tires

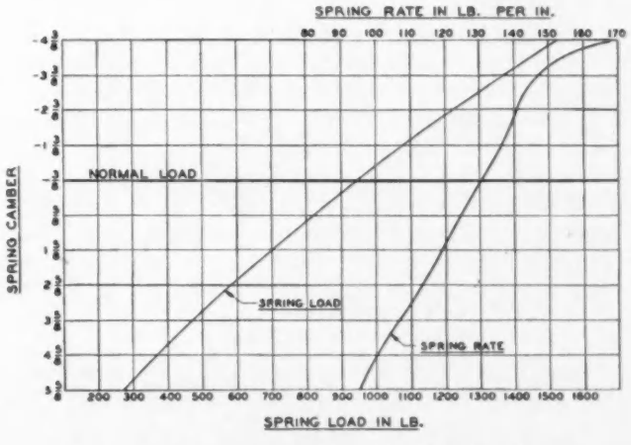
There have been few changes in wheels, sizes remaining the same. The Oldsmobile wheels have four relief portions under the rim in place of six. In the Chrysler products and Hudson, the discs are recessed under the rim in four places instead of three for the use of chain straps. On the Cadillac "50," "60S" and "61," four slots have been added to each disc. A forged-steel front hub supersedes the former malleable-iron type. Large hub caps predominate, and ribbed-side-wall tires are popular. The Packard 12 continues on 8.25-16 tires, the largest used in production passenger cars. Pontiac offers 6.50-16 oversize tires as special equipment on either the "26" or "28." Seiberling is offering a heat-vented truck tire in which there are small breathing cavities in each side wall adjacent to the tread. Spurred on by the developments in rayon, a new heat-resistant cotton cord has been developed. The yarn of the Bibb HR cord is saturated to soften the natural gums and waxes in the fibres, to fuse these all together. The saturated plied yarns are twisted under tension while wet to just within the breaking point.

Rear Suspension and Torque Members

Cadillac has introduced the Hi-Plane Hotchkiss drive. The rear end of the spring has been raised considerably by the use of a tension shackle which, incidentally, imparts a variable rate; the front end has been raised a small amount, the camber of the spring increased, and the main leaf is placed as close as possible to the axle housing. The car roll center has thus been raised, passing substantially through the axle centerline. In this way, if one wheel goes up and the other down, the axle rotates about its own center and no side force is exerted against the springs and body shock is minimized. In this way axle tramping and hopping have been reduced greatly, making the use of a cross-link stabilizer bar unnecessary. All spring eyes and shackles are now rubber-bushed. Pontiac's variable-rate rear spring utilizes a three-leaf auxiliary spring below the main spring. Its oscillation rate falls between 70 and 76 per min. as compared to the previous 86 to 96 per min., the variance being the difference between one passenger or five. A tight-fitting metal spring cover seals in the lubricant, and center clips between the main and auxiliary portions are not needed. In the individual spring leaves the lower corners on the compression side have been eliminated to prevent the customary cracking of the top corners. A 7 per cent weight saving also results.

Coil springs are utilized in the Oldsmobile rear suspension, and two channel-section torque arms approximately 58 in. long are attached to the axle housing on 36¼-in. centers. The arms extend back of the axle housing, being clipped to the under side, forming seats for the rear springs. The forward ends of the arms are rubber-bushing-supported from the frame X-member and are spaced approximately 10 in. apart. The forward position of the rubber-bushing supports was chosen carefully to give the most freedom from over-steer and under-steer characteristics and, being located midway between the universal joint and on a centerline coincident with the propeller shaft in its normal position, a constant velocity of the joints is obtained. Steel "seats" are welded to the axle housing which are encompassed by rubber insulators vulcanized to a steel backing, and the assembly is clipped to the torque arms. For sidewise control of the axle, a 1 1/16-in. by 0.144-in. gas-welded tubular stabilizer bar is used. End fittings are swedged and riveted into the tube and encompass rubber bushings.

The Buick rear torque tube has been supplemented by channel-section strut members which are clipped to the bottom side of the axle housing and extend to the rear to form the lower coil spring seat. At their forward end they converge and are bolted on a fitting welded below the torque tube near the ball. It is mounted in a rubber bushing behind the X-member center section by adjustable studs for centering. The torque ball has a rubber coating vulcanized to it and is treated to make it self-lubricating. The friction created by the clamping action of the inner and outer retainers affords a rear-axle damping control. Buick rear spring rates are lower,



Characteristics of Pontiac Variable-Rate Rear Spring

the "41" being 129 against 150 per min. in 1938. Nine active coils of 0.587-in. wire with 5 1/16-in. inside diameter replace 8½ coils of 0.615-in. wire with 5-in. inside diameter.

The Packard inserts of rubber and lead-antimony[1] introduced last year are now used on all chassis except the 12. The antimony inserts are now protected by a surrounding rubber washer. The frame anchorage of the Packard stabilizer bar consists of a Houde two-way shock absorber. On the Chevrolet rear spring with tapered leaves, the deflection rate on the five-passenger types has been reduced from 129 to 120 lb. per in., and four of its eight leaves are reduced in thickness. The rear-spring geometry has been changed to enhance steering characteristics by lowering the front eye 1⅜ in. and raising the rear eye. The spring seat is redesigned to raise the spring 3/16 in., and the angle of mounting is changed from 5 deg. 21 min. to 8 deg. 45 min. The Cadillac and LaSalle rear springs are mounted inside the frame rails. A sway eliminator is used by Chevrolet ahead of the knee-action unit on the Master DeLuxe and is mounted in rubber blocks attached to the under side of the frame rails. Each end of the bar is connected by a link rod to the lower spring seat of the suspension system. On the "85" a ¾-in. diameter sway bar is connected to the frame at each side by a rubber-bushed link and terminates in a rubber-bushed hole in the spring retainer block.

Front Suspension

All Chrysler products incorporate the parallel-link type of individual front suspension, utilizing Amola steel in the coil springs. The Studebaker President Planar suspension spring has been increased 1½ in. to 49½ in. The shock-absorber body forms part of, and oscillates with, the upper link member and the vane shaft is held stationary by an arm with a rubber grommet connection to the frame. Cadillac, Pontiac, and Oldsmobile are using stampings in place of forgings for the lower link. The threaded bearings at the outer ends and also at the inner end of the Oldsmobile and Pontiac are provided with extruded bosses. Spring seats are riveted to the assembly. Increased ground clearance and reduction in unsprung weight are attained. Rubber bushings are used at the inner end of the Cadillac and Buick lower links, the latter retaining the forged construction. New geometry provides a higher roll center for Buick, and wheel camber increases on a turn. Chevrolet has dropped the Dubonnet type and gone to the parallel-link method, the whole assembly being mounted on the front cross-member as a unit assembly. A weight saving of 31½ lb. is obtained in the new system. The shafts of the double-acting end-to-end discharge shock absorbers which are encompassed by, and secured to, the upper link, are parallel with the centerline of the car as well as with the lower link shafts. Threaded bearings and synthetic-rubber seals are used at all points. The deflection rate of the new coil springs is 286 lb. per in. or 106 at the wheels. These values compare with 1520 and 120 lb. per in. in the Dubonnet system. Two bosses on the front of each knuckle support have aligning surfaces for checking the caster angle.

Hudson's "Auto-Poise" control comprises a front sway-eliminator bar ahead of the axle with the ends connected to the wheel spindles by inclined connecting links, swinging outwardly to the spindle anchorages. Self-righting is claimed at all times.

Frame

The A. O. Smith I-beam type X-member frames continue in 1939, the section itself having been changed somewhat to reduce weight. More economical distribution of metal usually is obtained by the narrower but thicker upper and lower flanges and as deep an overall height as the chassis will permit. The Buick "40" and "60" side rails are of double construction from front to rear cross-members, this construction being necessary because of the reduced depth of the side rail caused by lowering the body floor. The frame does not extend beyond the rear spring seats, the extension of the car body beyond this point being supported by body reinforcement, taking the gasoline tank and bumper mounting. The Oldsmobile "70" and "80" have 2 in. lower floors and the running-board hanger passes through the side rail near its neutral axis in order to preserve ground clearance. When a sill panel replaces the running board, a shorter hanger is used or the conversion can be made in the field by cutting off the hangers and drilling new holes. The hanger, bolted in place, acts as a structural member between the frame X-member and the side rails. The "60" frame, although only 2 in. less in wheelbase, is approximately 25 lb. lighter than the 1938 six, whereas the series "70" and "80" frames are 2 in. longer and 10 lb. lighter. The anchoring of the series "80" radiator core and engine was designed in conjunction with the "70," the latter's radiator core being the location of the "80" front engine mounting and an extra frame cross-member supports the "80" radiator. This member is shaped so as to protect the crankshaft-mounted fan from front-wheel stone throw or any obstructions.

Cadillac "50" and "61" frames are 2½ in. lower or 13¼ in. from the ground. The frames are lighter, and further weight elimination has been obtained in a reduction in the number of body brackets. The body is secured to the frame at seven points on each side, but the bolts pass directly through the side-rail flanges at all points except the front and intermediate anchorages. The Chevrolet front cross-member is attached to the under side of the side rails by four bolts at the outer flange and three at the inner. The side-rail bottom plates have been changed from 1/16 to ⅛ in. thickness. The rear cross-member is attached to the side rails by heavier gussets. The Lincoln Zephyr front cross-member, supporting the engine and mounting the front spring, is insulated from the main frame structure. Chrysler products' frames are all deeper, especially at the center of the X-member.

Control

Improved steering geometry giving more accurate control and minimizing steering-wheel whip on roads has been incorporated in the Cadillac, Pontiac, and Oldsmobile layouts. A cross drag-link actuated by the steering-gear pitman arm is supported at the right side by an idler lever symmetrically located with respect to the pitman arm. A threaded bearing is used to support the idler lever. The drag-link to each wheel steering arm is ball-jointed to the cross drag-link at a point inside the Pitman-arm and idler-lever ball sockets. The Chevrolet Master DeLuxe gear, centralized with the lower suspension link frame shaft, uses a short tubular adjustable drag-link at the left side, whereas the right unit is a solid ⅝-in. diameter steel rod with a ball-socket housing forged at each side. Overall ratio is 21.6:1 as compared to the previous 18.2:1. The pitman arm is rubber-insulated from a plate mounted above it in which the drag-link ball studs are anchored. Two studs are riveted into the plate, each passing through a rubber bushing of two-piece conical section in the pitman arm. Excessive relative movement between the arm and plate is prevented by projecting lugs on the former. With the 12-in. further forward location of the gear housing, the steering-shaft length is increased from 40 7/32 in. to 53 5/16 in. The Master "85" with the conventional fore-and-aft drag-link provides a two-piece axially aligned rubber

[1] See S.A.E. TRANSACTIONS, Vol. 43, August 1938, pp. 313-318, 324: "Ride Controls and Calibration," by C. R. Paton.

bushing in the end of the pitman arm in which the ball stud is mounted. The overall ratio is 19:1 as against 17.5:1. All Chrysler products have the steering gear rubber-mounted in the frame.

Plastic steering wheels are used almost universally. A better view of the instruments is obtained by a wider spacing of the upper spokes, such as the 156 deg. between them in the Cadillacs. The Packard Super-8 and 12 are provided with two arc-shaped spokes tangent to the hub with their centers on a normal horizontal axis. Several provide a lower vertical spoke and a horizontal one tangent to the hub top. The Mercury has two horizontal in-line spokes at the center of the wheel, while Nash has four of the flexible-rod type, two of each being in line and each pair tangent to the hub.

Equipment

Headlamps built into the front fender crowns are used by Graham, Lincoln Zephyr, Mercury, Ford DeLuxe, all Chrysler products, Studebaker, Nash, and Hudson 6 and 8. Squat fender apron mounting is used by Oldsmobile, Pontiac, Chevrolet and Buick. Overland places them on the fender crown. The Buick lighting switch has a closer operating current-limit thermostat. The terminals soldered to the switch leads protrude through the case, resting against the sliding switch blade, eliminating intermediate connections and providing a sliding contact. All Cadillac headlamps are much longer and retain their hood side panel mounting. Similar mounting is used in the Hudson "112" and the Hupmobile senior line. The mounting of the license plate on the center of the trunk lid is now universal, and the lamp and trunk-lid handle are, in many cases, unified into a compact combination. Buick's emblem on the trunk lid is in the form of a flashing directional signal operated by a small flip-switch on the gear-shift lever and with a pilot light at its left. A metal-encased Tung-Sol current interrupter provides a flashing rate of 85-90 per min., utilizing thermostatic wire. Hudson Country Club 8s have narrow, long parking lights on the hood at the front end of a decorative molding.

Special equipment on the Chevrolet Master DeLuxe comprises a pair of Delco-Remy horns connected to a switch on the instrument panel above the speedometer with two positions marked "city" and "country." In the latter position the matched horns are in the circuit, otherwise the regular single horn is cut in. With the Oldsmobile solid steering-gear shaft, the horn-button contact has been changed to a collector-ring construction with the lead-off wire running between the mast and shaft. Horn rings are increasing, and unconventional types are found on Nash and Buick.

The Lincoln Zephyr front bumper is provided with two spaced horizontal streamlined bars between the bumper guards, preventing blocking of air flow into the grilles. Pontiac has developed a new glossy finish on the bumper collars and gas-tank filler neck grommet. No lacquer is used and rubbing with a soft cloth will restore luster.

Plastics are used to a considerable extent in the new instrument panels with speedometer and instruments immediately in front of the driver, often combined under one glass. The safety signal speedometer, used in the Chrysler products and built by Electric Auto-Lite, provides a Lucite button on the pointer disc and glows green up to 30 m.p.h., yellow between 30 and 50 m.p.h., and red thereabove. The Electric Auto-Lite gasoline, temperature, and oil pressure gages are built on the potentiometer principle and are of the dead-beat type. They are all substantially identical except for calibration, and the contacts are located in the dash instruments and not in the transmitting elements, to insure greater reliability. The Chrysler products place the headlamp-beam indicator above the instrument panel while, at the corresponding right point, is the glove-compartment latch button.

On the Packard 6 and "120" the instrument panel is finished in French burl walnut with wide chrome bands having vertical fluting. Speedometer numerals have been increased considerably in size. Olds has returned to a conventional-type panel. The ash receiver in the "60" is on the upper portion of the panel, whereas, in the "70" and "80," it is in the center of the control-button group. The Chevrolet speedometer numerals are twice as high as last year and range across the dial in a horizontal line. At each ten-mile increment there is a round red dot. The headlamp-beam indicator is located immediately above the 50-m.p.h. reading. The headlamp switch, formerly located with the choke and throttle controls, is now isolated at the left of the instrument group so there can be no possibility of the operator accidentally switching off the lights when seeking a control button or the cigar lighter. A number of cars are equipped with Casco automatic lighters. Automatic tuning of the radio receiver is the vogue. Six buttons usually are provided, and they actuate a solenoid which, in turn, performs the actual tuning process.

Studebaker is providing a distinctly new heating unit. It is located beneath the front floor panel below the left front seat and consists of a hot-water core unit beneath which is located a fan with a vertical motor. The sealed metal housing surrounding this unit obtains air from outside through a passage way located above the running board and, after passing through the air filter, the heated and cleaned air is discharged beneath the front seat where suitable apertures distribute it between the front and rear compartments. The capacity is in excess of 230 cu. ft. per min., and it is claimed that an interior temperature of 70 deg. fahr. at head level can be maintained in zero weather. In warm weather, when the hot-water supply is shut off, the unit acts as a ventilating system. A small motor and fan as a separate unit, available for defrosting the windshield, is located beneath the instrument panel and discharges warm air from the central heating system to the usual windshield slots. The Nash "weather eye" provides up to 800 cu. ft. of outside air per min. and is improved in principle over last-year's unit by the addition of a temperature-control dial and a fan switch just below the instrument panel, and an automatic thermostat controlling the water flow through the heating core. A rain-and-sleet shedder is added to the air filter. Pontiac is providing as an accessory a die-cast scoop attached to the right side of the cowl forcing fresh air into the heater through fabric tubing connected to the cowl ventilator. All Cadillacs introduce outside air into the body through the front heater, it being taken in through a small rectangular inlet located on the side of the cowl just above the front fender. The air is filtered and rain is separated from the inlet air prior to reaching the heater. The latter has a reversible motor so that heat can be delivered from the rear of the heater to give diffused warmth. A rear heater with a grilled register, mounted under the rear floor, is provided on the "75" and "90." A Cadillac accessory consists of a telescopic antenna mounted on the inside of the cowl at the left which can be extended or retracted about 20 in. by intake-manifold vacuum. A further 20-in. extension can be obtained manually. Olds provides a compensating condenser which automatically corrects for temperature changes of the set to assure constantly accurate tuning.

The Chevrolet shock absorbers of the horizontal parallel-cylinder type have the compression and rebound (at top) cylinders, one above the other, the pistons therein being actuated by a vertical arm with a cam at each end. Each cylinder is positively filled from the pressure discharge of the other. At the rear, a new link rod replaces the welded two-piece pressed-steel unit. The Master "85" has single-acting

rear shocks, whereas the fronts are of the direct-acting type. Their angular fore-and-aft mounting makes them more effective on the rebound than in compression. The larger-diameter direct-acting rear shocks of the Pontiac are placed at the rear of the axle housing with the lower ends spread out toward the wheels and inclined inwardly to the frame mounting at a considerable angle. A dual piston-rod seal is now used, while a baffle in the reservoir keeps the fluid in constant communication with the valves at all times, which are of the self-cleaning type. All these Delco units have chrome-plated piston-rods. In the Buick "40" and "60" front shocks the camshaft extends through the lower side of the body with the piston and cam above. The parallel-cylinder rear shocks are ahead of the axle with a link connection to the channel strut. On the "80" and "90" as well as on the Oldsmobile, the transfer of oil from one opposed cylinder to the other is through holes in the shock-absorber body instead of through piston-mounted valve chambers. The Oldsmobile units are mounted directly below the axle housing with the control arms to the rear. In the Chrysler products, the direct-acting front shock absorber is between the top link tie bolt and a boss near the outer end of the bottom link rear arm. The upper link of the Packard 6 and "120" is provided with a tie bolt under spring tension which brings pressure on a friction-material liner between bosses on the two arms. This construction aids in eliminating transverse vibration.

Grilles and Sheet Metal

The Lincoln Zephyr type of grille with louvreless hood has been adopted by Mercury, Ford DeLuxe, Studebaker, and Buick. Buick stands preeminent in the smooth, sleek fairing of hood and fenders. Retaining the center grille, though of lower and restricted opening and supplemented by side grilles in the fender apron are Cadillac, Chevrolet, Chrysler products, Hudson (except the "112"), Oldsmobile, Nash, and Pontiac. LaSalle now achieves its ultra-narrow front. The Nash center grille is also very slender. While center grille bars are always horizontal, there is no unanimity as to horizontal or vertical bars in the fender group. The lower portion of the apron is usually substantially vertical at this point for maximum effective grille area. Cadillac and LaSalle swing the side grilles back a considerable distance since the headlamps do not interfere. The Chryslers have widely spaced vertical bars extending across the center between the fender aprons and are supplemented by a few louvres in the hood portion above. Where side grilles are used, scoops in the fender direct the air from the high-pressure area to the radiator core. Though die castings are used in most instances, the Chevrolet side grilles are in the form of a stamped louvre group in fender color. Packard, with its usual worthy front end, paints each alternate radiator shutter of the Super-8 and 12 in body color.

Pontiac has adopted the alligator hood. All Hudson alligator hoods hinge at the front and have a latch control lever at the left side of the instrument panel. Cadillac has incorporated the safety catch as part of the hood-latching mechanism, and the spring-loaded hood support is mounted in the center of the dash, divorcing the supporting linkage and springs from the hinges to avoid excessive friction. The Oldsmobile alligator hood extending down at the sides to the fender aprons is of two-piece construction riveted together at the center and covered by a plated molding. The hood latch comprises a handle behind the front bumper which, when released, permits the hood to travel upwardly 2 in., after which further movement is possible upon release of the safety catch. Chevrolet continues with the side and top panels at each side of the hinge as a single sheet. Lincoln Zephyr has raised the hood nose while Mercury's is well rounded. The Overland hood has a rounded V-front, tapering forward with stamped grilles parallel to those in the front apron and bumper-bar center. The four-bar Buick hood louvre utilizes a pivoted portion of the two center bars for a handle and lock.

The right front fender in a number of instances carries a license bracket in order to prevent blocking of the air stream for cooling. Front fenders are generally more massive due to headlamp mounting and/or apron grilles. In those cars in which running boards have been eliminated, the lower portion of the rear fender is protected by a rubber cover, the appearance of which has been improved by a process to impart a smooth black finish. The Mercury rear-fender opening has a small radius and a deep valance. Chrysler incorporates a raised valance panel of streamlined form in both fenders.

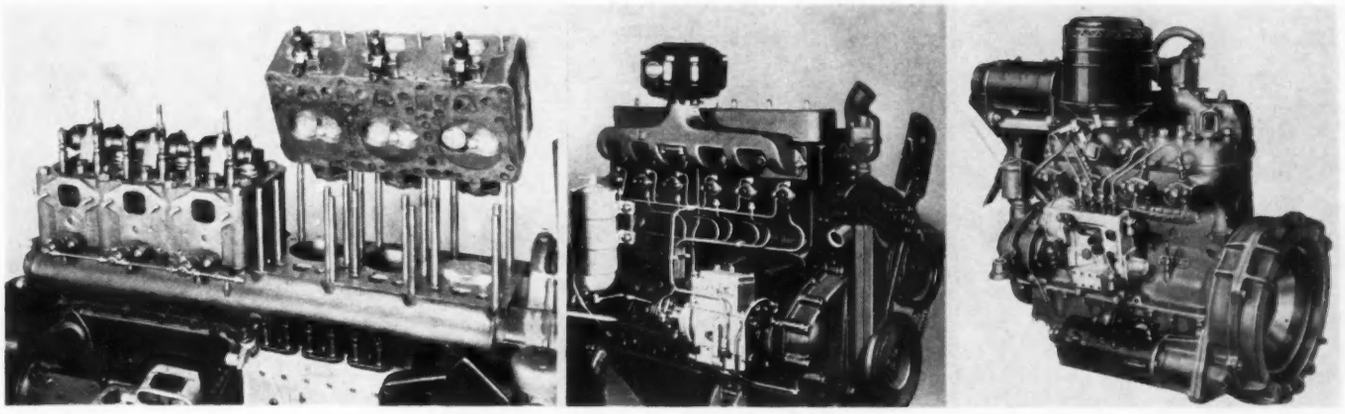
Body

Bodies are in general lower, wider, roomier and with better vision. The sunshine top, available on many of the General Motors cars, marks the part adoption of a popular foreign type. A 24 in. long by 40 in. wide opening is obtainable.

General Motors has developed a prevailing design based on last year's Cadillac "60" Special in which the side panels continue back into the trunk to emphasize length and apparent lowness. A sharp crease below the windows gives the effect of a distinct superstructure. The Chrysler products, other than Plymouth, utilize a prevailing design with V-windshield and rain trap in the cowl ventilator, and in which the formerly projecting trunk is merged into a smooth rounded, tapering rear quarter. Hudson does likewise, and Nash offers both types.

Windshield posts have been narrowed considerably, and the corner radii reduced as well as in windows. In the body shell utilized by LaSalle, Cadillac, Pontiac, and Oldsmobile the maximum windshield pillar width, inscribed in a circle encompassing the remotest parts, has been reduced from 4¾ in. to 4⅛ in. With greater width and height, the windshield glass area has been increased from 4.05 sq. ft. to 5.33 sq. ft., or 32 per cent. Front-door windows (both sides), rear-door windows, and rear-quarter windows have been increased 46, 3, and 24 per cent respectively, or a total increase, including the windshield, of 27 per cent. Front-entrance room has been increased by placing the front corner post further forward. The Chrysler products are equipped with Electric Auto-Lite windshield wipers operating twin blades and powered by a compact motor of power sufficient to be practically stall-proof. Should the blades be frozen to the windshield, a clutch prevents the motor from being left in a stalled condition. Automatic parking insures the blades of always coming to rest at the same point at the bottom and out of the driver's vision. The motor must start under load at 4 volts. Many instances of longer windshield wipers are to be seen. The Chevrolet windshield-wiper control knob is of streamlined shape, affording a handle for easy manipulation and to indicate the "off" position. On the convertible G. M. bodies the sun shades also can be raised above the normal position to act as a draft deflector when the top is down. A sliding-bolt lock is attached to the ventipane window to engage the vertical window guide, preventing the window from being forced open from the outside. The rear-quarter windows no longer pivot but slide back into the rear-quarter panel under movement of a sliding handle at the lower front corner. The window is self-locking as the spring-loaded lower end of the handle moves longitudinally into slots along a concealed track. Many of the cars are equipped with narrow chrome beading around all windows.

Optional running boards are now available on Graham and all G. M. products other than Chevrolet. This provision is possible since the fenders are separate from the running



Mack-Lanova, Dodge, and Hercules Ford-Replacement Diesels

boards. The lower edge of the body in all Chrysler products, other than Plymouth, has a flair extending out over the running boards. When the doors are opened, additional step width is available. In the Lincoln Zephyr body a skirt is secured to the base of the doors, entirely covering the running board and keeping it clean. Length is emphasized further by continuity of color and contour. The fender ends are faired into the skirt contour. In the General Motors bodies the body panel at the sides of the trunk flairs outwardly to meet the narrow rear fenders with a fillet effect, avoiding the usual design with square corner. Center pillars have been moved forward to increase shoulder room when entering the rear door. Bantam and Hudson are using the Goodyear "Airfoam" latex rubber seat cushion over the coil-spring unit. The Cadillac "75" and "90" bodies have mineral wool on the cowl sides behind the side kick-pads. An asphaltum sealer binds the wool and holds it in position. The Cadillac "61" dash insulating pad consists of four double thicknesses of asphalt-impregnated slaters felt, each double thickness consisting of one plain and one waffled sheet of material. The Buick "40" and "60" rear body mounting is at the center of the rear cross-member. There are four mountings at each side, the front being on the side-rail top flange and the remainder on brackets.

In the new Chrysler body line and with all G. M. bodies, the front-door upper hinges are concealed and, in the G. M. line, the rear-door hinges also are concealed. Hancock rotary door latches are used on the new Chrysler line, and the door lock buttons are mounted in the window garnish moldings. On G. M. bodies, outside door locks are provided on both front doors, having spring covers located below the handles. A pivoting safety latch is held in position by a coil spring recessed in back of the striker plate, permitting the latch to deflect when the lock bolt passes over it, resulting in easier and quieter operation.

In most instances ash trays in the rear compartment are located at the front of the arm rests. In the De Soto four-door sedans, an ash tray is built into the back of the front seat. Plastic knobs and handles are used throughout interiors to an increasing extent. The rumble seat is disappearing, most coupes being provided with opera seats within the body. Graham provides a combination coupe to take six adults on two full-width seats under one roof. Pontiac and Cadillac front-seat back frames are now constructed of steel tubing. All G. M. arm rests have leather tops. Automatic raising and lowering of the Plymouth runabout top is achieved by a vacuum cylinder at each side in the body.

Trunk capacities have increased. Carpets are now standard on all Cadillacs, and Packard uses a sprayed-on finish for the shelf and sides. The new Chrysler bodies eliminate the bulging trunk and the new G. M. design has dropped the "flat-back" type of body. More luggage compartments are provided with the spare tire placed vertically. The trunk lid on the Nash streamline bodies is counterbalanced and will remain at any angle.

Heavy Vehicles

A replacement four-cylinder Diesel for Ford V-8 trucks has been developed by Hercules particularly for export use. Cylinders are 4- by 4½-in.; displacement, 226.2 cu. in.; compression ratio, 16:1; and it develops 70 hp. at 2600 r.p.m. and a maximum torque of 162 ft-lb. at 1400 r.p.m. Hercules is now working on a six-cylinder, 5- by 6-in. flat Diesel. Dodge has available for its 3-ton truck a six-cylinder, 3¾- by 5-in., 331 cu. in. displacement Diesel equipped with Ex-Cello pump, pintle-type nozzles and automatic injection timed according to engine speed. The Mack-Lanova is a six, 4⅜- by 5¾-in., 519 cu. in. displacement, developing 131 hp. at 2000 r.p.m., a maximum torque of 381 lb-ft. between 1200 and 1450 r.p.m., and a b.m.e.p. of 110 lb. per sq. in. Bosch pump and pintle-type nozzles are used. The cylinder-block and crankcase are integral; dry-type sleeves are of high nickel-chromium semi-steel. Solid copper gaskets are used between the twin heads and the block. Connecting-rod caps are split at 35 deg. from the horizontal and have a tongue-and-groove joint with the rod. Shimless copper-lead steel-backed interchangeable precision bearings are used on the main and connecting-rod pins. High-pressure lines are of high-tensile steel and have an equal length which was determined scientifically to be non-resonant with the pump pulsations. Waukesha has produced an Overland series of six-cylinder Hesselman engines of 5¾- and 6¼-in. bore and 6½-in. stroke developing 172 and 202 hp. respectively at 1850 r.p.m. The wet-type cylinder sleeves are of Waukesha Molychrome iron and a Proferall cast-alloy crankshaft is used. The engine is stopped through interruption of the ignition system by cut-off of the fuel feed.

Hercules has added an RXL series of six-cylinder gasoline engines with bores of 4⅜ in. and 4¾ in. with a 5¼-in. stroke. The Mack Thermodyne engine is a six-cylinder, 4⅜-in. by 5¾-in., developing 146 hp. at 2200 r.p.m. and 380 lb-ft. torque between 1000 and 1400 r.p.m. Valves are in the head and the combustion-chamber is D-shaped. Each of the twin heads is secured by 20 hold-down studs. Valve time provides a 20-deg. overlap. The lower portion of the bell housing is provided with a screened opening for clutch ventilation.

Chevrolet trucks have a horizontal bar grille, and the front-end appearance is entirely different from that of the passenger car. Cabs are 1½ in. longer and wheelbases have been increased a similar amount to retain the same CF dimensions. COE trucks, giving a gain of 28¼ in. from the center of the

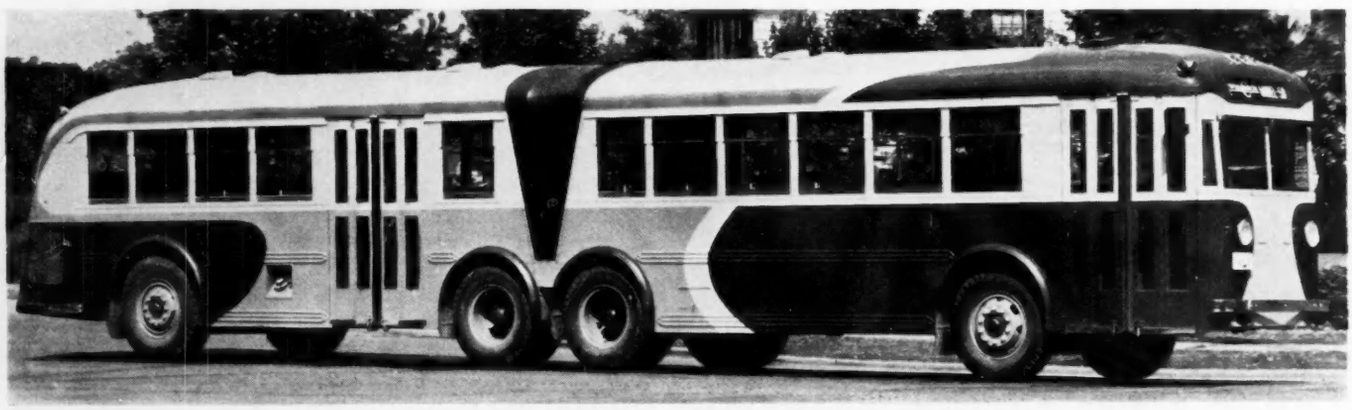
front axle to the back of the cab, have wheelbases of 107⅜ and 131⅛ in., and take the standard bodies that are mounted on the conventional 133 and 158½-in. wheelbase trucks. Chevrolet and Dodge are offering two-speed rear axles as optional equipment. The Mack composite rear axle comprises a pressed-steel banjo, tube sleeves and spindle tubes of seamless-steel tubing which are fused together electrically, using shielded welding rods.

Twin Coach Co. has developed a Super-Twin consisting of two 35-passenger sized bodies joined together back-to-back with large rubber hinges just below the floor line. Floor movement in going over ramps or depressions is absorbed by a collapsible rubber seam which contracts and expands only to the extent of 7/16 in. At the roof line the ends of the two bodies have an 11 in. maximum movement, and this section is bridged by a rubber diaphragm made especially to stretch in the longitudinal direction, this stretch amounting to only a few inches over ordinary city streets. Two dual-tired driving axles are located in the center and can be powered either by Diesel-electric or trolley bus equipment. The electric driving motors in either case are located approximately amidship of each body unit. The Diesel bus is powered with a Hercules model HXB, 5- by 6-in. engine. The steering layout of the front unit is conventional with mechanical connections extending to the rear-end axle, the wheels of which turn simultaneously with those of the front axle but in the opposite direction. The turning radius is equivalent to a 35-passenger vehicle with a 195-in. wheelbase. Although manual power is sufficient in ordinary driving traffic, an air servo supplements the manual effort.

The new Reo coaches, ranging from 25- to 34-passenger capacity in transit and parlor-car type, are built of aluminum and steel around a monocoque type of construction. The seamless frame tubes of Yoloy steel tie the cross-members together beneath the floor and extend in 12 ft. from each end, tying together at least three cross sills. They, in turn, are bolted to a ¾-in. plywood floor with bolts spaced every 5 in. These tubes in connection with the floor give a high compressive value to distribute endwise shocks into the body skin without possible localized crumbling of the shell. Tubular mounting of the springs overcomes the torsional weakness of the conventional channel rail. All panels are united by 3/16-in. dural rivets on 1½-in. centers and, where two rows of rivets are required, they are staggered to prevent any possibility of parallel fracture of the panels. Floor edges are of the coved type, the floor covering material running into a coved molding of approximately 3-in. radius where it joins the side wall. The engine is placed transversely under the rear seat, and the drive is at right angles from a transfer case. The air openings, radiator, and fan are at the side of the body. Monocoque body and frame construction is incorporated in the 103½-in. wheelbase dairy car with 44-deg. cramp angle of the front wheels and 120-in. wheelbase parcel car with a 275 cu. ft. body. The engine is placed crosswise directly behind the driver with a right-angle drive from the engine to the transmission. Four-cylinder engines are used with displacements ranging from 124 to 162 cu. in., giving 16 to 17½ miles per gal. with a 1½-ton load and 10 stops per mile. Weights are approximately 3680 and 3790 lb. for the dairy and parcel cars respectively.

ACF is producing a model 26-S with the front door ahead of the wheels for city operation and the 26-U coach with the door behind the front wheels for suburban operation. Both are equipped with the Hall-Scott six-cylinder horizontal, 4- by 5-in., 337 cu. in. flat engine developing 106 hp. at 2600 r.p.m. In conjunction with the Carrier Corp., ACF has developed an air-conditioning system which is factory-installed and operating in many fleets in the Southern and Western territory. A four-cylinder auxiliary engine drives the air-conditioning compressor and these two units, together with the condensers, are mounted under the floor to the rear of the entrance door. This apparatus is arranged so that it may be removed, if desired, during seasons when cooling is not required. The evaporator and two motor-driven fans are mounted above the operator in the front of the bus, and the cool air is blown from this compartment back into the bus at ceiling level. The interior baggage racks serve to distribute the cool air throughout the interior. The equipment is capable of maintaining an inside temperature of 80 deg. fahr. when the atmospheric temperature is 100 deg. fahr. The fans have a capacity of 1100 cu. ft. per min. and make possible a complete recirculation of the inside air once every minute. In addition, 370 cu. ft. of filtered fresh air are introduced into the coach every minute, making a complete air change every 3 min. All windows are stationary and sealed with the exception of the extreme front rear sash which are of the swing type, and the body of the coach is insulated with 1½ in. thick mineral-wool blanket.

Utilizing the basic principles of the ice engine for railway service, Waukesha in conjunction with Tropic-Aire, has developed an air-conditioning system used by Greyhound. The central unit occupies a space 17 by 24 by 44½ in. and weighs 390 lb. The output is 3 tons of refrigeration at 1800 r.p.m. and 4 tons at 2400 r.p.m. with 90 deg. fahr. condenser entering air, 40 deg. fahr. evaporator temperature, a water consumption of 4 to 7 gal. per hr., and an average gasoline consumption of ¼ gal. per hr. The blower-type air-circulating fan, direct-driven from the engine crankshaft, has a capacity of 900 cu. ft. per min. and furnishes condenser and radiator cooling air and cabinet ventilation. The unit is removable for inspection by means of a six-roller mounting that travels in channel tracks.



Super-Twin Coach

Proposed

Method of Test for Ignition Quality of Diesel Fuels[1]

THIS is a proposed method and is published as information only. Comments are solicited and should be addressed to the Headquarters of the American Society for Testing Materials, 260 South Broad St., Philadelphia, Pa.

1. Scope – This method of test is intended for determining the ignition quality of Diesel fuels in terms of an arbitrary scale of cetane numbers.

2. Cetane Number – The cetane number of a Diesel fuel is defined by and is numerically equal to the percentage by volume of cetane in a mixture of cetane and α-methylnaphthalene which the fuel matches in ignition quality when compared by the procedure specified herein. Thus, by definition, α-methylnaphthalene has a cetane-number rating of zero and cetane, of 100. Cetane number is reported to the nearest whole number.

Apparatus

3. Apparatus – The ignition-quality testing unit described in this section shall be used without modification. It shall be known as the "C.F.R. Diesel-Fuel Testing Unit."[2] The apparatus shall consist of a continuously-variable-compression engine, together with instrument panel, base, and suitable loading and accessory equipment, as follows:

a. Engine – Continuously-variable-compression, one-cylinder, with dimensions as follows:

Bore, in.	3.25
Stroke, in.	4.50
Displacement, cu. in.	37.4
Turbulence combustion-chamber:	
bore, in.	1.625
length, in., adjustable	0.375 to 2.750
Valve-port diameter, in.	1.187
Connecting-rod bearing:	
diameter, in.	2.25
length, in.	1.625
Front main bearing:	
diameter, in.	2.25
length, in.	2.00
Rear main bearing:	
diameter, in.	2.25
length, in.	4.25
Piston-pin, floating, diameter, in.	1.25
Connecting-rod, length, center to center, in.	10.00
Timing-gear face, in.	1.00
Piston rings, number	5
Exhaust pipe, diameter, in.	1.25
Weight of engine, lb. (approximate)	475
Weight of complete unit, lb. (approximate)	1375

b. Crankshaft – Fully machined, heat-treated, and counterbalanced.

c. Crankcase – Cast iron.

d. Connecting-Rod – Rifle-drilled, S.A.E. No. 1045 steel, heat-treated, bearing alloy cast directly into big end.

e. Main Bearings – Renewable-sleeve bushings, babbitt-lined.

f. Valves – Silcrome.

g. Push-Rods – Mushroom type with lock-nut adjustment.

h. Cylinder – Cast-iron alloy, bored and honed, Brinell hardness 220 ± 20.

i. Cylinder-Head – Detachable variable-compression high-turbulence.

j. Cooling System – Evaporative.

k. Lubrication – Pressure feed to main, connecting-rod, piston-pin and camshaft bearings, to idler-gear stud and gears, and to valve rocker arms.

l. Oil Heater and Thermometer – Electric heater in base to bring oil to operating temperature quickly, and a thermometer on the instrument panel to indicate when equilibrium temperature has been reached.

m. Air Heater and Thermometer – Electric heater mounted on intake silencer to maintain constant air-inlet temperature, and a thermometer at base of silencer to indicate this temperature.

n. Water Heater and Thermometer – Electric heater in cooling-water line between outlet of condenser and inlet to cylinder jacket to bring water to operating temperature quickly, and a thermometer between condenser and heater to indicate this temperature.

o. Injection System – This system shall consist of the following parts as furnished with the engine:

(*1*) Injector pump, Bosch specification PE1B50A 302/3S97, port closing at 0.075 to 0.090 in. lift from base circle.

(*2*) Injector, Bosch specification DN – 30S3.

(*3*) Fuel line (tank to pump), ⅜-in. copper tubing, bottom of fuel tank to be 25 ± 1 in. above pump inlet.

THIS method has been developed in the Automotive Diesel Fuels Division of the Cooperative Fuels Research Committee and was released for publication at the Sept. 16 Meeting of the C.F.R. Committee following a demonstration meeting at Wood River, Ill., and a Division Meeting at the 1938 Summer Meeting of the Society. The method has been under development for some years by a Volunteer Group under the leadership of T. B. Rendel. ("Reports of the Volunteer Group for Compression-Ignition Research" were published in S.A.E. TRANSACTIONS, June 1935, pp. 206-209; June 1936, pp. 225-233; January 1938, pp. 27-36; the first two by T. B. Rendel and the third by C. H. Baxley and T. B. Rendel.) This Group some time ago was taken up by the C.F.R. Committee as its Automotive Diesel Fuels Division.

An Exchange Group to test the proposed method already has been joined by 21 laboratories. The proposed method also is published as an appendix to the 1938 Report of Committee D-2 on Petroleum Products and Lubricants of the A.S.T.M. *for information only*, or can be obtained from the A.S.T.M. in pamphlet form.

[1] Under the standardization procedure of the American Society for Testing Materials, this proposed method is under the jurisdiction of the A.S.T.M. Committee D-2 on Petroleum Products and Lubricants.

[2] At present the sole authorized manufacturer of the C.F.R. engine is the Waukesha Motor Co., Waukesha, Wis. Other manufacturers may be approved in the future, but testing laboratories should not purchase testing units, except from the Waukesha Motor Co., without ascertaining whether such units have been approved. Inquiries in this connection should be directed to R. P. Anderson, Secretary of Committee D-2 on Petroleum Products and Lubricants of the A.S.T.M., 50 West 50th St., New York, N. Y.

(4) The injection line, ¼ in. O.D., ⅛ in. I.D., and 36 in. in length.

(5) Vertical open riser with measuring burette located on pump sump outlet.

p. Instrumentation – The ignition quality is measured by the coincident-flash fixed-delay method using neon lights which are located on the periphery of the flywheel and are operated by contact points on the combustion and injection indicators. The necessary instrumentation consists of the following parts:

(1) The combustion indicator which mounts in a tapped hole (⅞ in. by 18 threads per in.) in the combustion-chamber and is equipped with a spring-steel indicator diaphragm 0.543 ± 0.003 in. in diameter and 0.015 ± 0.0005 in. in thickness. The contact points are mounted on bronze leaf springs 0.021 ± 0.0005 in. in thickness. Graduated spring-tension adjusting screws are provided. In addition, a graduated gap-adjusting screw is provided with a spring cushion plunger to protect the contact points against overloading as the result of combustion pressures.

(2) The injection indicator, which utilizes contact-point assembly parts similar to the combustion indicator, with brackets for attachment to permit operation of the contact points by movement of the injector valve.

(3) The neon-light mount which carries two ¼-watt, 110-volt, neon bulbs located behind an aperture plate with 13 deg. spacing between openings.

(4) The sight tube and top dead-center reference mark with brackets.

(5) Flywheel slip-rings and brush holders with necessary brackets and wiring for the neon-light support.

(6) A 110-volt direct-current generator to supply current for the injection and combustion indicators. This generator is belt-driven from the power-absorbing unit described under Paragraph q. The voltage is controlled to 115 ± 5 volts by means of a field rheostat located on the instrument panel.

q. Power-Absorbing Unit – The engine is connected by V-type belts to an electric generator which preferably should be an induction motor with synchronous characteristics, but may be any electric generator capable of maintaining proper operating conditions. In most cases the electric generator will act as a starting motor to crank the engine, but if a direct-current generator is used and no outside source of current is available, the engine may have to be cranked by hand.

r. Panel – A control panel with starter switches, heater switches, and oil-temperature and pressure gages is included as part of the unit.

s. Base – The complete unit is mounted on a cast-iron base.

4. Foundation – The foundation shall be of concrete, at least 15 in. in height above the floor level, resting, if possible, directly on the ground. Otherwise, special provision must be made to eliminate vibrations that may influence ratings, and the manufacturer should be consulted. For convenience in operation, the foundation should be located at least 2 ft. from any wall. Dimensions are furnished with the apparatus.

5. Reference Fuels – The primary reference fuels[3] shall be normal cetane and α-methylnaphthalene. Secondary reference fuels,[4] suitably calibrated against these primary reference fuels, are normally used in actual testing.

Procedure

6. Standard Operating Conditions – The engine shall be run under the following standard conditions:

a. Engine Speed – Constant, 900 ± 9 r.p.m.

b. Jacket Temperature – Constant within ± 1 deg. fahr. (0.6 deg. cent.), and at a temperature between the limits of 209 and 215 deg. fahr. (98 and 102 deg. cent.).

c. Cooling Liquid – Distilled water or rain water, or ethylene-glycol solution when necessary at high altitude.

d. Crankcase Lubricating Oil – S.A.E. 30.

Note 1 – The viscosity range of crankcase lubricating oil, S.A.E. 30, is from 185 to 255 sec. when determined on the Saybolt Universal viscosimeter at 130 deg. fahr. (54.4 deg. cent.), in accordance with the Standard Method of Test for Viscosity by Means of the Saybolt Viscosimeter (A.S.T.M. Designation: D 88) of the American Society for Testing Materials.[5]

e. Oil Pressure – 25 to 30 lb. per sq. in. under operating conditions.

f. Oil Temperature – The electric oil heater shall be used only to raise the oil temperature rapidly to the equilibrium operating temperature, 120 to 150 deg. fahr. (49 to 65 deg. cent.), normal operation maintaining the equilibrium thereafter. By "equilibrium" is meant that steady temperature attained by the oil upon continuous operation of the engine without the use of the heater.

g. Valve Clearances, Cold – Intake 0.008 in., exhaust 0.010 in.

h. Injection Advance – 13 deg. b.t.d.c. for rating and 5 deg. (± 2 deg.) b.t.d.c. for combustion contact setting.

i. Injection Pressure (injector opening pressure) – 1500 ± 50 lb. per sq. in. (106 ± 4 kg. per sq. cm.).

j. Injection Quantity – 13.0 ± 0.5 ml. per min., to be checked on each test sample and reference-fuel blend.

k. Injection-Pump Setting – See Paragraph 3 (o) for injection-system specifications.

l. Injector Cooling-Water Jacket Temperature – 100 ± 5 deg. fahr. (38 ± 3 deg. cent.).

m. Exhaust Pipe – A separate exhaust pipe shall be used for each engine. It shall be made of pipe of not less than 1¼ in. in inside diameter having a maximum of two ells and a total length not to exceed 20 ft. The use of a short straight-through muffler with passage 1¼ in. in diameter is permissible.

n. Air-Inlet Temperature – The air-inlet temperature shall be maintained at 150 ± 2 deg. fahr. (66 ± 1 deg. cent.).

o. Clearance Volume – The clearance volume measured to the top face of the combustion indicator hole shall be 72 ± 0.5 ml. with the engine at top dead-center. At this volume the micrometer should be set to read 2.000 in. The compression ratio is given by the equation:

$$C.R. = \frac{18}{L} + 1$$

where L = the length of the combustion-chamber in inches as measured by the micrometer.

7. Starting and Stopping the Engine – While the engine is being turned over by electric motor, the fuel bypass valve on the injector shall be closed and the compression ratio increased until the engine begins to fire. To stop the engine, the fuel bypass valve on the injector shall be opened and the electric motor then switched off.

8. Checking Injection Pump for Port Closing – The pump-plunger port should close when the plunger has traveled up 0.075 to 0.090 in. from the base circle of the cam. This setting is important as it influences the injection rate. This adjustment is made in the factory and should not require resetting unless it has been tampered with. To check the port closing the following operations shall be made:

a. Remove the discharge valve and spring from the top of the pump and replace the discharge-valve clamp nut with injection tube arranged conveniently for blowing into.

b. Submerge the end of the fuel drain line in oil.

c. Remove the pump cover, open the fuel pump rack to its wide-open position, and turn the engine until the pump plunger starts up. Then blow on the tube and, at the same time, turn the engine slowly until bubbles disappear at the end of the fuel drain line which is submerged in oil. When bubbles cease, it indicates that the port has just closed. Determine this point carefully by several trials and mark the flywheel.

d. Remove the tube used for blowing into the pump and insert a pin in the discharge-valve hole and resting on the top of the plunger. Arrange a dial indicator resting on the pin to indicate the pump-plunger lift. With the plunger at the bottom of its stroke and the dial gage set on zero, turn the engine until the reference mark previously placed on the flywheel to indicate point of port closing, registers. Then read the dial gage which will indicate the pump-plunger lift from the base circle when the port closes. This lift should be 0.075 to 0.090 in. Any deviation from this value can be adjusted by turning the pump-plunger adjusting screw. After making an adjustment, a new reference mark must be established on the flywheel as outlined in Paragraph c.

9. Injection Pressure Setting – The injection-pump cover shall be removed and, with injection-line pressure gage connected and injector arranged to spray into the air, the pump plunger shall be operated with a screw-driver used as a lever. With the pressure gage set at 1500 lb. per sq. in. (106 kg. per sq. cm.), the pressure on the injector spring shall be adjusted until equal quantities of fuel spray from the gage and the injector. The opening pressure of the injector will then be the same as indicated on the gage.

(*Continued on page* 464)

[3] Normal cetane is available from the E. I. du Pont de Nemours & Co., Inc., Wilmington, Del., and α-methylnaphthalene, from the Reilly Tar and Chemical Co., Indianapolis, Ind.

[4] High-cetane secondary reference fuel known as "Shell high-cetane reference fuel," is available from the Shell Petroleum Corp., Wood River, Ill. Low-cetane secondary reference fuel, a mixture of α- and β-methylnaphthalenes, is available from the Reilly Tar and Chemical Co., Indianapolis, Ind.

[5] See 1938 Supplement to Book of A.S.T.M. Standards.

Experimental Procedure of Testing and Organization

By J. M. Crawford and P. A. Collins

Chevrolet Motor Division, General Motors Corp.

JUST what goes on in the development of a new model at Chevrolet from the time requirements are set until it goes to the dealer, is told by the authors. They explain that the engineering department is divided into three major groups – designing, production, and testing, and they outline the jobs in the department from that of the chief engineer down to those of young men without specific titles who are receiving special training which will qualify them for advancement in engineering.

This discussion is made specific as the authors "high-spot" some of the problems of car design and development from a technical as well as a timing point of view, in bringing out the 1937 model which, they say, represented the greatest change in any year to date.

They note that every unit of this model, except the front suspension wheel unit, was redesigned completely. Decision to produce it in its final form, they report, was made about the time that its predecessor, the 1936 model, was started in production, adding that this decision could be made only because it was based upon designs of various major units that had been on the drawing boards and on test since early in 1934.

The part played by the various engineering sections and their personnel receives particular attention, as does the cooperation between the sections. The authors also tell how G.M.C. facilities, such as the General Motors Research Division, and the Proving Grounds at Milford, Mich., and Phoenix, Ariz., tie up with this development work.

BEFORE describing the engineering organization and test procedure of Chevrolet, it might be well to cover briefly its physical set-up as well as its products, for both have an important bearing on the Engineering Department approach to its own organization problems.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 16, 1938.]

In many respects, Chevrolet differs from other large producers. Compared to one, it is a Division of a large Corporation, with its major unit manufacturing plants spread over a large area – in contrast to concentrated manufacturing activities. Compared to others, it has world-wide assembly-plant facilities and a wider range of products.

Chevrolet Manufacturing Plants

The map of Michigan and adjoining states, Fig. 1, shows the location of its principal activities in this area. The General Offices are located in the General Motors Building Annex in Detroit. Manufacturing plants located in Detroit consist of the Axle, Forge, Spring, and Bumper Divisions. The Main Engine Plant and Pressed Metal Division are located in Flint, Mich. The Bay City Plant supplies carburetors, die-casting, upset parts, and screw-machine products. The foundry operations for crankcase, cylinder-heads, transmission cases, and so on, are concentrated in Saginaw, Mich. Here, also, is located the main Transmission Plant, an auxiliary Bumper Plant, and the Service Parts Manufacturing Division.

Transmission Plants also are located in Toledo, O., and Muncie, Ind.

All commercial and truck bodies are produced in our own plant at Indianapolis, Ind.

An auxiliary Engine and Axle Plant recently has started production in Buffalo, N. Y.

These unit manufacturing plants have a capacity of 7000 units per day in two eight-hour shifts.

Chevrolet Assembly Plants

In addition to these main plants are 11 assembly plants located throughout the United States, as shown in Fig. 2, Flint being the F.O.B. point.

General Motors of Canada and the Export Division assemble Chevrolet cars and trucks in 20 plants outside of the United States, as shown in Figs. 3 and 4.

The products consist of two models of passenger cars (differing mainly in the type of suspension), and three commercial models of different chassis design. Altogether, there are 60 different models released to production when body types and wheelbases are taken into account on passenger cars and trucks. This number does not include variations due to wheel and tire, and other special equipment in the commercial line.

The physical set-up of the Engineering Department is spread over a considerable area, somewhat in the manner of the Manufacturing Department (see Fig. 1); and its own

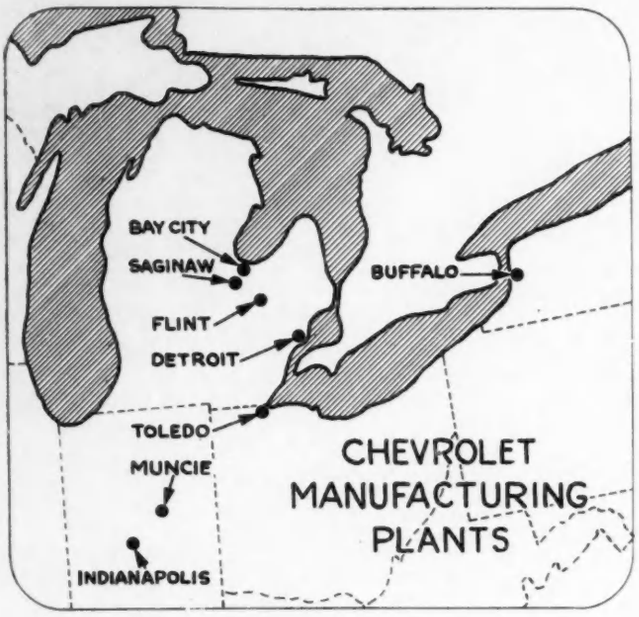


Fig. 1

facilities for product development are augmented by the facilities of the Corporation, which further spreads its activities.

The Engineering Executive Offices and Design Departments are located on the ground floor of the General Offices.

The Experimental Laboratory and building operations are adjacent to the Axle Plant property – a 15-min. drive from the main offices.

The Plant or Production Engineers responsible to the Engineering Department in Detroit are located at Flint, Detroit, Buffalo, Saginaw, Muncie, Toledo, and Indianapolis – handling the engine, axle, transmission, and commercial body problems.

The Corporation facilities available consist of the General Motors Research Division located across the street from our General Offices. Fisher Body Engineering and the Styling Section also are located in this same building. Proving Grounds are located in Milford, Mich., and Phoenix, Ariz.

Chevrolet Engineering Organization

The Engineering Organization is covered in a 29-page book describing the functions and contacts of its various Divisions, and in another Book of Procedure describing them in detail. Definite responsibility is placed on each individual, as shown in the Organization Chart of Fig. 5. This chart shows the Engineering Department divided into three major groups – Designing, Production, and Testing (Experimental).

The Chief Engineer establishes all policies and contacts with the Management and Departments within Chevrolet, and with Fisher Body, the Styling Section, and other Corporation Divisions. These contacts are handled direct, as well as through contact men in various capacities in the Engineering Department. The engineers engaged in such contact work report back to the Assistant Chief Engineer whose duty it is to coordinate their work with the activities of the internal organization which is under his direct supervision.

In this capacity, the Assistant Chief Engineer is responsible for the coordination of all activities headed up by the Design, Production, and Experimental Engineers, as well as the Drafting Room and Records Departments.

The Design Engineering Division develops all new designs on paper and is responsible until these designs are released to the Production Engineering Division. This statement does not mean that its interest ceases at this point, as it is still charged with the responsibility of following up its designs until the model is discontinued, to see that no changes are made from a production standpoint to alter the original requirements of the design. Nor does it mean that this Division is in absolute control of the design, as the Plant Production Engineers are brought into the problem through the Production Engineer as early and as often as his judgment dictates.

In actual practice, the Production Engineer is called in on the problem as soon as the basic principles of the design are established. His recommendations are given consideration, and the second design incorporates all changes necessary from a manufacturing and cost standpoint.

In some cases where the manufacturing problems outweigh the technical ones, both engineers go to work on the problems simultaneously, and no samples are built until the manufacturing problems are satisfied.

In this way designs go into production without last-minute changes due to the shop's inability to meet the specifications. Changes of this type are never made in time to re-test and, unless they are extremely minor in nature, they generally result either in field trouble or increased cost and delays.

Naturally, the Plant Engineer looks at the problem from the standpoint of equipment, either that in the plant or that available, whichever produces the best result for the dollar and, where new equipment is needed, enough time is provided by letting the shop get in on the ground floor to check thoroughly the result before large commitments for equipment are made.

It is largely due to the fact that the Production Engineer has a dual responsibility that the Engineering and Production Departments are brought so closely together. He is responsible to his Plant Manager for the production result, and to the Central Office Engineering Staff for the technical result.

Quality checking of production parts is another function that brings the two departments together. Reports of quality, handled by a staff under the Plant Engineer, go to both the Plant Manager and the Central Engineering Department in Detroit. The value of these reports to the Inspection Department is obvious, and the Designing Engineers benefit by having their attention called to the design the specifications of which are difficult to control. And, just as the Engineering Department has the privilege of checking up Production, so

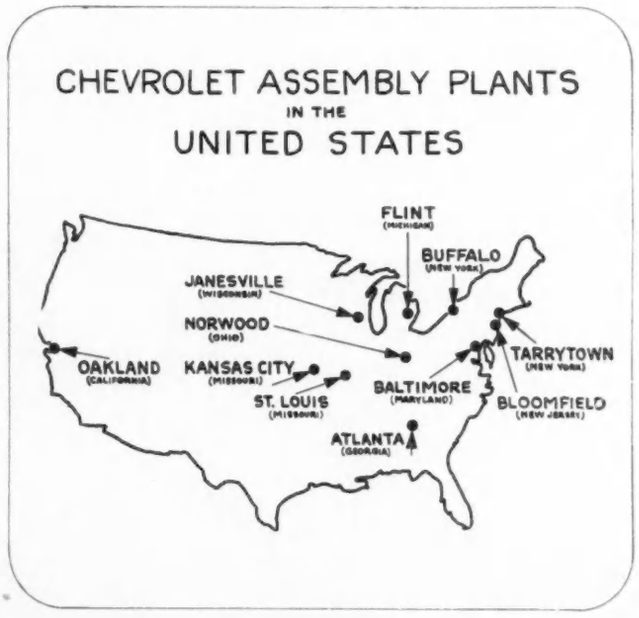


Fig. 2

the Production Department receives reports of Central Office Engineering developments through the Plant Engineer, which are in any way likely to affect their future program.

The point to be made here is that there is no sharp line of demarcation where the Engineering Department turns its development over to the shop to clean up with the hope that all will be well, for their interests are all pooled from the very beginning to produce the best possible results in big volume for the money expended.

In addition to its advantages in developing products, the effect on the development of men must be recognized. Plant Engineers are strengthened by being on their own to a large extent, and cooperative efforts between departments always give the management the opportunity of observing manpower development through the opinions of many, rather than through only department heads.

The Experimental Engineering Division comprises the necessary group to build experimental material – to test and report performance of experimental and production material as well as of competitive products – to supervise the design-checking of production material – to contact with the Design Engineering Division as to corrections and suggestions – and to prepare and tabulate necessary weight data and analyses.

The Experimental Engineer is the contact between this Division and Design Engineering, and all exchange of information is through this engineer.

It is through the work order and reporting system of this Department that the development of a model is controlled by the engineering executives. It is the "watch dog" of the Engineering Organization, as well as of the follow-up system in many respects.

It enables all those responsible to take direct action to further the progress of the development by keeping them in almost daily touch with all the details. The distribution of orders and reports is such as to encourage the cooperation of engineers other than those actually responsible for the design.

The procedure in this Department is rigid and uncompromising in nature – as it has the dual responsibility of preventing defective design getting into production, as well as of assisting in development by fact-finding experimentation.

Before going into the details of this Department which will be covered later in this paper, I thought it might be well to "high-spot" some of the problems of car design and development from a technical as well as the timing standpoint to explain why ours must be an evolutionary development rather than a revolutionary one. Inasmuch as the 1937 Model represented the greatest change in any year to date, we will take that Model as an example.

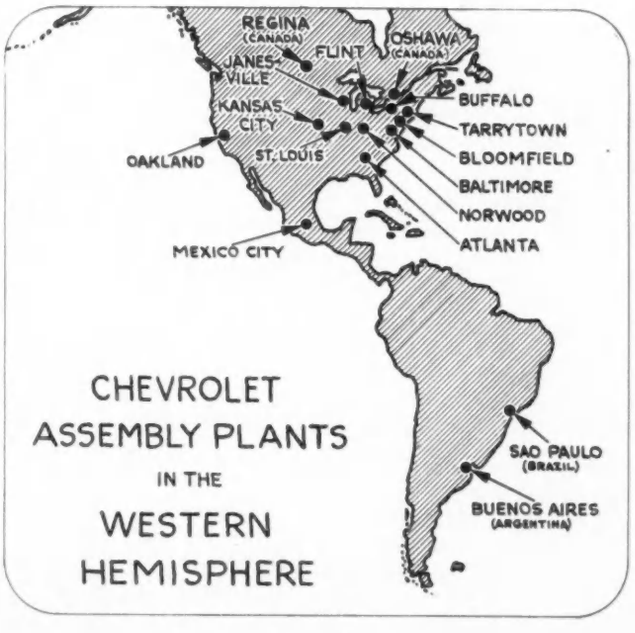


Fig. 3



Fig. 4

Th center-line cross-section of the car shown in Fig. 6 will serve to illustrate how the development of these units is tied together. I might say here that the objective was to reduce the weight approximately 200 lb. with an increase in performance of 15 per cent – without a sacrifice in economy of operation or durability, and without a reduction in passenger room or comfort, but with increased visibility.

The fundamental characteristics of the box-section frame which, of necessity, must be considered in working out a new body design, were determined in 1934 and built into our Standard Model of 1936. Thus, 40 per cent of passenger-car volume was used to work out the production problems in the frame. Drafting-room work actually started Mar. 7, 1934.

The hypoid axle design, upon which the floor height and rear-seat position were based, was started July 9, 1935, at the Central Office and Mar. 9, 1935, at the Gear and Axle Plant, and put on test April 21, 1935.

This unit, being one where the manufacturing problems outweighed the technical ones, called for a simultaneous development by the Engineering and the Production Departments.

The transmission, upon which the front floor height was based, was likewise a cooperative design – where new as well as old plants were given consideration, and the amount of weight reduction in this unit, with the increased torque handled, could only have been accomplished safely by starting the design July 31, 1934, and putting it on test November, 1934.

The engine development will be taken up later on as an

example of how the Experimental Division functions. Its development was undertaken for the sole purpose of increasing performance and smoothness at a reduction in weight and length. It was adopted in this Model to permit the maximum passenger capacity for a given wheelbase and proper load distribution. The latter two features were determined only after the powerplant development had reached a stage which determined the length of the unit and the placement of the front suspension. The work started on this powerplant April 3, 1934, and the decision to use this engine in the 1937 Model was based on the progress made by Oct. 18, 1935.

Engineering Test Facilities

The testing and development facilities used in connection with the foregoing design problems are, like the balance of our Organization, widely spread, as shown in Fig. 7. The General Motors Research Laboratories are used for gear-testing; cold-room work; brake and clutch-lining tests; study of dynamics with respect to balance, vibration, and stress; general chemistry problems such as lubricants, paints, plating and rubber research; organic chemistry as applied to fuels, and so on; physics, as applied to special instrumentation.

Harrison Radiator conducts cooling tests in its wind tunnel.

Dynamometer and other test equipment, to determine fully the characteristics of their respective products, are under the supervision of the Plant Production Engineers at the Engine, Transmission, and Axle Plants.

The Production Chemical and Physical Test Laboratories are also at their disposal.

The Central Office Engineers are provided with an Experimental Shop and Laboratory of about 61,000 sq. ft. (all on one floor) with facilities for making and servicing complete cars and trucks including the body. Its own office force is set up to handle purchasing and all other problems in connection with flow of material.

The Laboratory, which has just been expanded, covers an area of 13,150 sq. ft., containing 17 individual rooms in which are housed dynamometer, chassis rolls, carburetor flow bench, stroking machines, and so on.

The Proving Ground at Milford has, in addition to its speed loop and hill route, a complete chassis dynamometer and, in a separate building, a modern laboratory for sound determination. On the grounds, Chevrolet has its own quarters and staff reporting to the Experimental Engineer in Detroit.

It is the responsibility of the Experimental Engineer to schedule and coordinate the work handled in these various Organizations and, for this purpose, his staff is set up in the Central Office in Detroit.

We have told you how the Experimental Division fits into the operation of the Chevrolet Engineering Department. To explain the Experimental Organization and operation, we have decided to talk about three things: first, the general organization; second, the method of control; and third, the method of reporting and disposition of information. Taking the first of these items (the general organization), we will refer to Fig. 8.

Experimental Engineering Organization

There are three principal divisions, these divisions being controlled through the Experimental Central Office located in the Central Office Engineering Department in Detroit. The first of these divisions is the Shop and Material Control. This Division is under the Superintendent and his assistants who are responsible for the various departments and the work being carried on in each one of them.

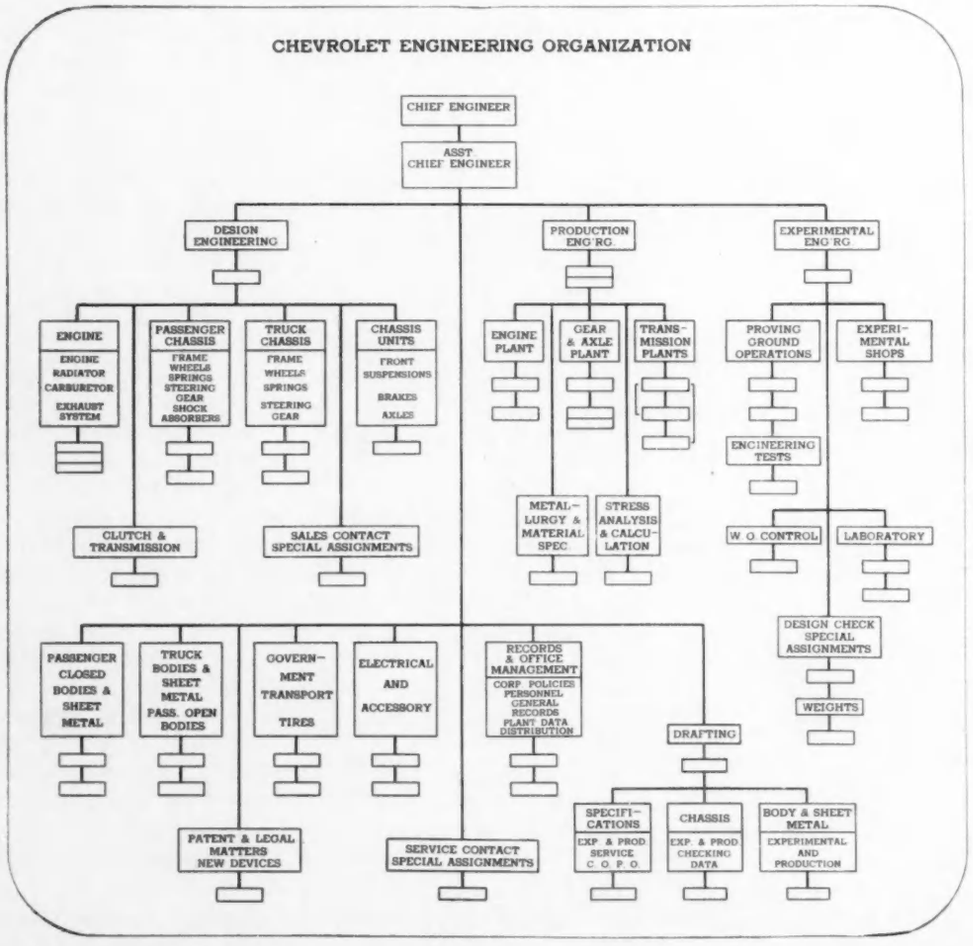


Fig. 5

By referring to the chart, it will be seen that the Experimental Shop is divided into units, such as Machine Shop, Pattern Shop, and Sheet Metal. The purchasing, follow-up, and disposition of material is also handled under this department, the material for test finally being assembled in the Engine Assembly, Final Assembly, and Experimental Garage. This department produces or procures all material for test and design check.

When the construction of material or purchase of necessary parts has been completed, the material is turned over to the Laboratory or to the Field Test Division. These two divisions are set up separately, each one of them reporting to the Experimental Central Office.

The Field Test Division, under the Superintendent of Field Tests, is located at the General Motors Proving Grounds at

Milford and is divided into two major divisions. These divisions are Development Testing and Durability Testing.

For example, at certain times each year production cars are run on durability test. For this purpose, complete cars are taken as delivered to the dealer, and run for 20,000 to 25,000 miles on a fixed schedule, using three shifts 24 hr. per day, rotating drivers daily and allowing ½ hr. on each shift for inspection by trained observers. The Superintendent of Tests and his Assistant also ride the cars periodically to report operating conditions.

This schedule calls for a circuit of approximately 14 miles, consisting of concrete speed loop, gravel and Tarvia roads over a hill course, and one mile of Belgian block road. These schedules also call for a predetermined number of stops to check brakes, and starts, for clutch operation, and all the other items usual to road testing.

At the conclusion of these tests the cars are disassembled completely in the Detroit Experimental Department, and laid out for inspection by the Chief Engineer, Assistant Chief Engineer, Design Engineer, and Project and Production Engineers. The units, such as engine, transmission, and axle, then are shipped to the Plant Production Engineers for their inspection and reports.

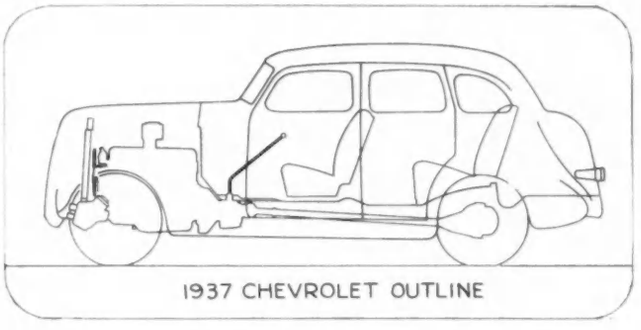


Fig. 6

In comparing the results of these accelerated tests with cars in normal service, a ratio of 3:1 has been established. Hence, parts going through 20,000 miles O.K. are considered satisfactory.

Other durability tests on experimental material are conducted in much the same manner, although various schedules are used. These schedules vary according to the type of material being tested. However, all schedules are based upon previous test experience and are covered in detail in a 200-page book, which contains complete procedure for the Experimental Division.

In the Development Section, under the supervision of the Development Test Engineer, all special tests are handled. This work covers all testing when mileage is not a particular factor and various tests such as brake deceleration; car performance and economy; steering effort; handling and roadability; high-speed bearing tests; governor tests; cooling tests; and numerous other tests which require instrumentation.

All parts the performance of which either can be measured or judged must be O.K.'d by the Development Section before durability testing is undertaken. For the purpose of comparison, regular production cars in standard condition are used. These cars are set aside for this purpose alone and, under no condition, are they used for durability testing.

By referring to the chart, you will notice that special tests by the General Motors Division at the Proving Grounds are sometimes run in cooperation with our Chevrolet Test Crew. This condition is also true at the General Motors Laboratory

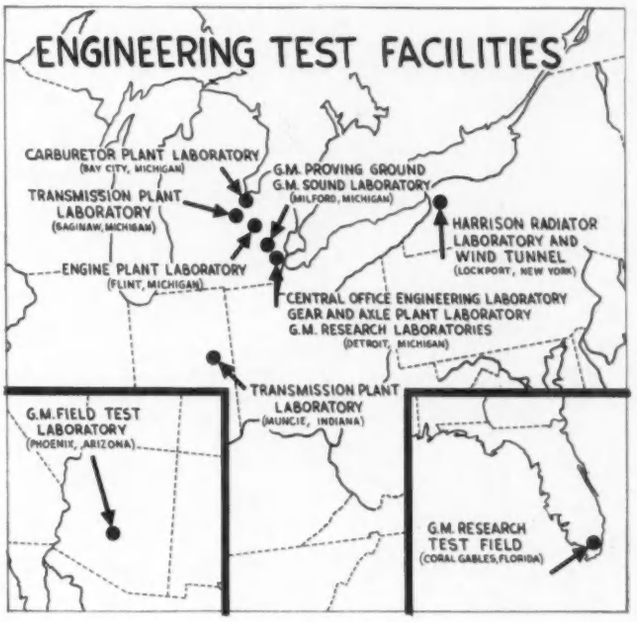


Fig. 7

at Phoenix. In these cases, the reports are handled under the supervision of our own test organization.

Some interesting figures covering development and durability tests over a ten-year period are given:

Over the past ten years the Chevrolet Division at the Proving Grounds has averaged 1,440,735 miles per year; has used an average of 134,712 gal. of gasoline per year; and an average of 10,232 qt. of oil per year. The average cost per mile is approximately $0.07. This figure covers the complete cost of operation, including salaries, wages, overhead, parts, gasoline, oil, tires, maintenance, repairs, and changes of material.

The personnel of the Development and Durability Sections under the Test Superintendent averages 60 men including drivers, mechanics, inspectors, and special test and office men.

It will be noted that the results of all tests are covered in the end by Final Reports, about which we will tell you more later on.

The other testing division, represented by the Laboratory on the chart, has been referred to previously and consists of very complete equipment for dynamic and physical testing. This department is under the Laboratory Superintendent who reports to the Experimental Engineer at Central Office.

Most of the test work on motors is carried out in accordance with "The General Motors Test Code" with which some of you are familiar. This Test Code very definitely specifies how and under what conditions all motor testing shall be done. It would be too complicated to discuss at this time, but it can be said that all tests are run to a definite base line and, for that reason, any comparison can be made within a reasonable period of time. The numerous other tests are run according to characteristics involved and material in question, with all results being reported as Final Reports when all data have been completed.

At this point it might be of interest to call your attention to the method of tying together results found in Field Tests and those obtained in the Laboratory.

Due to the time and cost element involved when failures occur on durability tests, no improved design is re-tested at the Proving Ground until an improved Laboratory result is obtained.

Immediately after a failure, steps are taken to reproduce the results in the Laboratory by special test so that a base line

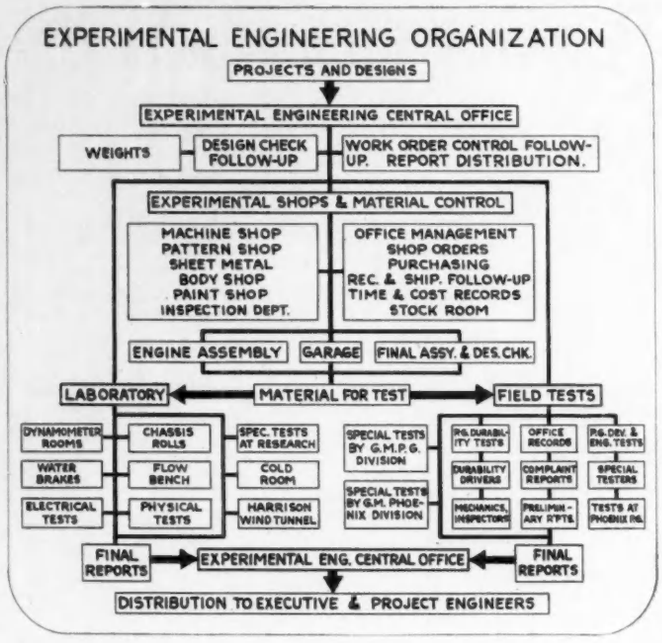


Fig. 8

can be set up. Dynamometer or water brakes are used on units where power absorption is required, and stroking machines are used where deflection of structural parts is the cause of failure.

This method of development *after* a given result on the road is produced greatly accelerates the work and reduces it to a scientific procedure. When the life of the unit or part is brought up to that of successful designs in actual production, it is then re-tested at the Proving Grounds.

It is true, of course, that a great deal of the development work can and is done in the Laboratory before road tests are attempted, but road tests are necessary to establish the base line before the job is cleaned up.

We have explained these three major divisions of the Experimental Organization and their functions and have mentioned several times that they are controlled by the Experimental Engineer at Central Office located in the Central Office in Detroit.

In this office we find the Central Office Engineer who is responsible for coordinating the work done by the Department, such as Work Order control; maintaining proper files; following up building projects and progress of all tests and field reports. He is also responsible for the distribution of all information obtained to the interested parties.

The Contact Engineer is responsible for all Design Check and follow-up of same. He also has supervision over the Weight Analysis Engineer who handles all weight records of Experimental and Production details, as well as of competitive cars as furnished by the General Motors Proving Ground Staff.

Method of Control

Our second subject is the method of control. Before explaining the system used, it is necessary to call your attention to the location of our test facilities as explained earlier in this paper. This set-up shows a decentralized organization, and the control system has been developed to operate under these conditions.

Projects handled by the Experimental Department originate with the Executive Staff and Project Engineers who are responsible for new design or changes to parts that are controlled by the Central Office; and with Production Engineers who are responsible for changes to production material found necessary to facilitate manufacturing, or for the checking of material from various production sources.

To start a project through the Experimental Department, it is necessary that the Project Engineer first write up a Design Request to the Chief Draftsman, stating the changes to be made and the reason, or explaining the idea back of a new design. See Fig. 9. This request (represented by a line on the chart of Fig. 9) must be approved by the Assistant Chief Engineer, after which it is sent to the Records Department for proper numbering, typing, and distribution. The Drafting Room then proceeds to make up the necessary drawings in collaboration with the Project Engineer who instigated the work.

If it is then decided to build the parts covered by the design, a Building Order Request is written by the Project Engineer, giving a description, part numbers, number of parts, and use for which the material is intended. Another line on the chart represents the route followed by these orders, which are sent to the Experimental Engineer's Office for approval. This office is in close touch with the work being carried on in the Experimental Department, thus preventing overlapping of projects and planning the work to obtain the best results from the facilities available. A copy of the approved Building Work Order is sent to the Drafting Room where then is issued the necessary drawings, releases, parts list, and bill of material to the Experimental Department for its use. Another line represents these releases.

When test work is required of this Department, a request is submitted to the Experimental Engineer by the Project or Production Engineer responsible for the particular unit or model. The Experimental Engineer's Office determines where the best results will be obtained, and the request is then sent to the Records Department for numbering, typing, and distributing, the basic number assigned to the Design or Building Order automatically being applied to the test order. Another line shows the route of the Test Orders.

In the case of the Production Engineer, a request to test material is all that is necessary as parts handled by this Department have gone through the design and building stage previous to production and, in most cases, are material from different sources for comparison with material which already has been tested and approved.

The Chief Engineer and other Executives, as well as the Project and Plant Engineers interested in the development, receive copies of the Test Orders.

The primary purpose of the foregoing distribution is to prevent duplication of effort in a widely scattered organization, and to keep the Engineering Executives fully informed of details in their *final* stage of development.

None of the divisions of the Experimental Department operates without authorization through the work-order control system due to the fact that verbal orders, in an organization of this type, would cause so much confusion that no definite results could be obtained.

Reporting and Distributing Information

The third subject is that of reporting and distribution of information. These reports might be termed the resulting product of the Experimental Organization and, as such, are written with care by men trained for the job.

Personal opinions are kept out of report conclusions at all times, and facts established by test and analysis are the only acceptable data. Fig. 10 shows the test report distribution.

The Report System used at the Proving Grounds consists of Daily Reports on Production and Competitive Cars, Production Complaint Reports, Experimental Complaints, Development Work Order Reports, and Final Reports.

The Daily Reports on Production and Competitive cars or trucks cover all work done during the previous 24 hr., that is, miles run, total miles to date, weather conditions, road conditions, and the schedule the car or truck is running on. This Report (as represented on the chart) receives an extensive distribution, starting with the President of the Company, with copies to the Plant Management, Chief Engineer, Assistant Chief Engineer, Production Engineers, and Project or Design Engineers. These reports are distributed from the Proving Grounds and sent directly to the person to whom they are addressed. There is no censorship possible, and each car tested gives a complete record of every item during the test.

The Production Complaint Reports cover individual production parts on which trouble has been experienced, and are a very definite method of bringing to the attention of the Production and Project Engineers the first indication of failure, or of excessive adjustment of production parts. A complaint is made out by the Superintendent of Field Tests and sent to the Experimental Engineer's Office for approval and distribution, which procedure immediately starts a process that cannot be stopped until the complaint is cleaned up. The complaint as issued covers the part, car, or truck involved; cause of the trouble; mileage run; schedule used; number of cars or trucks running under the same condition; and if the same trouble was experienced on previous models.

So that comparisons can be made quickly, a running average of all adjustments and failures on Durability Cars is maintained in a cross-index card filing system. This file is available to the Staff for filling out information on Complaint Reports.

The Complaint cannot be closed out until it is decided definitely that the trouble has been eliminated – the close-out being authorized by the Experimental Engineer, whose office has a follow-up on all complaints and issues a weekly memorandum showing the progress of all Complaint Reports. The Proving Ground Division issues a summary once a month, showing the status of all live complaints.

A record of the behavior of production parts on experimental models is reported on the same form as the complaints on Production Durability Cars, except the color of the report form is changed. These are considered as extra data on production parts, or show the effect of related experimental parts on production parts.

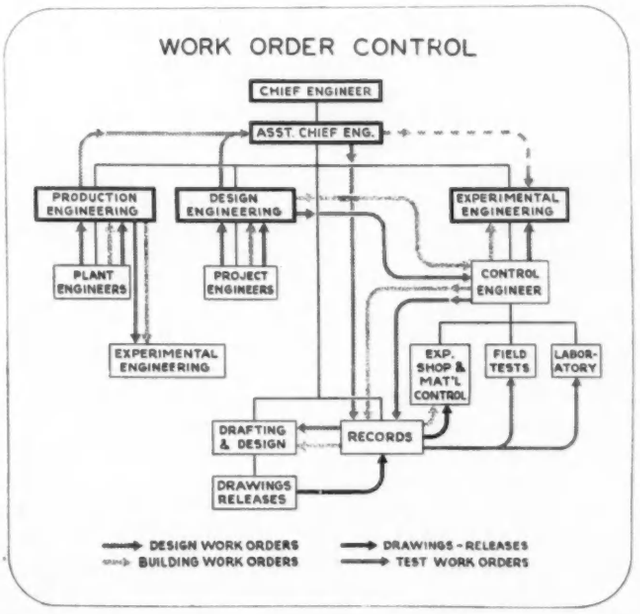


Fig. 9

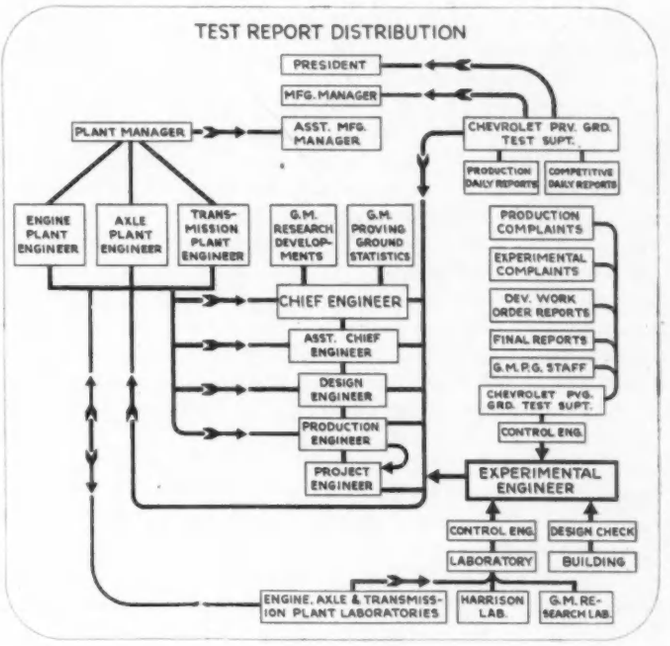


Fig. 10

The Development Work Order Report on progress of Experimental test material is written to the engineer requesting the test, and shows the cars involved, mileage on the material, and any work done or inspection made and the findings. Distribution of this report includes the Chief Engineer, Assistant Chief Engineer, Design and Production Engineers. These reports permit all parties interested to suggest ways and means of overcoming defects as shown by tests.

The Development Engineer also is responsible for compiling Final Technical Test Reports. (The form used for these reports is set up to show the results quickly without reading the complete report.) In this report, the first item shown is the Object of Test; next, the Conclusion Formed; then a description of the material being tested; method of test; and a complete log of all items that occurred during the test. In this Development Department, as well as in the Experimental Laboratory, are young engineers who actually are receiving special training which will qualify them for advancement in the Engineering Department when vacancies occur. These men have been drivers, mechanics, office men, and report writers. In some cases, they already have been in the Laboratory and Drafting Room, and are now following along in the Development and Special Test Department.

In addition to our own work at the Proving Grounds, the Corporation Staff in charge at the Proving Grounds furnish, through the Chief Engineer, a mass of statistical data as well as performance characteristics of all cars of domestic production, as well as of foreign cars of interest to any Division. Information of this type shows us our competitive position from the viewpoint of a disinterested party. As a matter of fact, the Corporation products never get the benefit of the doubt in comparisons set up by this Division.

The final or Technical Test Reports issued by the Laboratory follow the same form as those issued by the Proving Ground Staff and cover in detail the "Object of the Test," "Conclusions," "Description of Material," and all "supporting data." As shown on the chart, these reports have the same distribution as those originating at the Proving Grounds.

Reports from associated testing laboratories, where work is

being carried on at the request of our Division, are handled through the same channels and are subject to the same distribution as our own reports.

Reports on Design Check come from the Design Check and Final Assembly at the Experimental Shops to the Central Office, and are handled by a procedure similar to the Complaint Reports. The Central Office Contact Engineer, the Design Engineer and the Chief Draftsman, each receive a copy. This Design Check Complaint cannot be closed out until changes have been made on the drawings and material changed and re-checked by the Design Check Department. If the change eliminates the difficulty encountered with the original item, a close-out is issued by Design Check. This report, on a different color of the Complaint Form, follows the same distribution and indicates to each department that the trouble has been remedied.

Reports from the Plant Engineers, covering results obtained in their own laboratory investigation, are represented on the chart. These reports reach the Chief Engineer and Plant Managers by direct mail. They come to the Project Engineers through the Production Engineer to encourage discussion and cooperation between the two and reach the experimental files through the Detroit Laboratory and Plant Laboratories.

This completes the outline covering the general operation of the Experimental Department. To illustrate how the system works, we have outlined a certain development and test history.

Engine Taken as Example

The Laboratory probably is used in engine development more than for any other unit of the car, and the engine is taken as an example to show how the experimental procedure functions in car development. First, it presents by far the greatest *number* of problems and, secondly, in this particular instance, it brings out the necessity of planning well in advance when radical changes are contemplated, and large-scale manufacturing plants are involved.

When the engine design was started in April, 1934, no installation requirements were set up. It was merely to be a powerplant of minimum weight and length for a piston-displacement range of 207 to 224 cu. in. Increased smoothness and economy were also major requirements.

Inasmuch as our particular type of valve arrangement does not determine the length of the engine as it does in other designs, the crankshaft became the starting point, and so two engines of 209 cu. in. capacity, 3 7/16-in. bore, 3¾-in. stroke, were built and ready for dynamometer test in November, 1934. The length of the block itself was based on a 3 9/16-in. bore, but the smaller size was selected for test so that a direct comparison for efficiency could be made with our then-production engine of 207 cu. in. The crankpins were 2 5/16 in. in diameter with abnormally large No. 1 and No. 4 main bearings, 4 in. in diameter. This combination on a short-stroke shaft produced a large overlap at the No. 1 and No. 9 arms so that only a ¼-in. web was felt necessary to join the No. 1 and No. 6 crankpins to the end main bearings.

Other features of the engine not before used in Chevrolet production designs included full pressure lubrication, chain-driven camshaft, and four-ring pistons and, on account of the consideration being given to these new features, it was necessary that a great many production studies be made from layouts before the details were released for samples.

Provision also was made in the original design so that our regular method of handling the foregoing features could be incorporated at a later date without, in any way, compromising the results to be obtained from the new features. In other words, all our development is undertaken with a view of establishing direct comparisons between one construction and another, with as few as possible variations in related parts which may have some bearing on the final result.

The development test on the foregoing engine started in the Laboratory, November, 1934. Within a week, Proving-Ground tests of a duplicate engine were started. The dynamometer test immediately brought out the lack of durability of the main bearings, and the Proving-Ground tests were discontinued at the end of 17,000 miles.

These failures resulted in the building of two more engines with changes in the crankshaft designed to overcome the problems encountered. These tests were confined to the Laboratory, and the design was abandoned due to unsatisfactory results.

The third series of engines, retaining all the new features with the exception of the crankshaft, was put on test April 23, 1935. The crankshaft on these engines was of a more conventional design, employing arms at the front and rear main bearings and reducing the diameter of these bearings to approximate that of the intermediate bearing.

Inasmuch as the main bearings were still much larger than ever before employed and a great deal of development work was undertaken on the dynamometer tests to overcome the defects of the original design, this new crank was subjected to a large number of tests covering the studies of oil pressure and delivery requirements, the effects of various methods of grooving the bearings and drilling the crankshaft, as well as the effects of the crankcase ventilating system on the problems encountered.

Eighteen engines were built of the third design, so that tests on the Proving Ground could be run on durability, economy and performance, the Proving Ground results being constantly checked with the Laboratory. Engines also were installed in current production cars with various axle ratios, to put the engineers in a position to evaluate the factors of flexibility, smoothness, economy, and performance. (Three sizes, of 209, 216, and 224 cu. in. piston displacement engines were used for this purpose.) The noise level at various road speeds and with various exhaust systems was studied. The effect on these problems of the body, which was just getting into production, was studied by the engineers driving these cars.

As a result of these observations, the Engineering Department recommended the adoption of the 216 cu. in. engine for further development in connection with two different axle ratios. The Management approved these recommendations after trials at the Proving Ground on the basis of performance and smoothness compared to the 1936 production model. This, as stated before, was on Oct. 18, 1935.

Changes Suggested by Shop

After this decision a more intensive study was made by the Manufacturing Department, and recommendations were made as to changes in the crankshaft design to permit of new methods being used in the line-boring of cam and crankshaft bearings in the case.

In a desire to cooperate, the design was changed, and new sample engines were installed in Proving Ground and engineers' cars. Unfavorable results were reported by both the Proving Grounds and the engineers. In order to study the problems brought about by these changes, special instrumentation dealing with dynamics was developed by the General Motors Research Laboratory and used in our own Laboratory to find the source of the trouble.

The original design was not returned to, in spite of its successful demonstration, but a major change was made which affected not only the crankshaft and case but the wheelbase as well, in order to provide a greater factor of safety than existed in the original design.

This change was approved April 9, 1936, as a result of

rebuilding and testing several engines, and at no loss in time to production – due largely to the Engineering Department's method of release of tooling and equipment in accordance with the results of tests. These items are handled separately from the release of final designs.

The piston is another example of the cooperation between the shop and the Engineering Department and represents the extremes to which both go to secure an improved result.

The piston design started with a conventional four-ring, cast-iron type, 32-oz. weight. The wrist pin was 1 in. in diameter with a lock-screw in one pin boss and with full pressure feed to the pin. Dynamometer tests for power resulted in a detailed investigation which eliminated this piston design. The piston development then turned to an extremely light-weight steel casting with domed head, slipper skirt design, employing a wrist pin of ⅞-in. diameter.

Dynamometer tests for 50 hr., full load, maximum speed, were found to be satisfactory.

Manufacturing studies were started immediately, sample production machinery was purchased, and all machining problems were worked out.

In the meanwhile, detailed investigation was made of the structural and heat-dissipation characteristics of the design, and a number of engines were tested at the Proving Grounds in cars and trucks, and in the Laboratory, under various speeds, full load.

As a result of these investigations, the weight was increased gradually until it was thought possible to substitute a high-tensile cast iron for steel. The steel pistons for this experimental program were made with metal pattern equipment and were machined by the Production Department on a machine line-up especially purchased for this design.

The cast-iron development that followed resulted in a slight increase in weight, and the same production equipment was utilized. The weight was well under that of the original piston for which the connecting-rods and rod bearings were figured and tested so that satisfactory results could be guaranteed. In the meanwhile, however, the oiling system was changed which had its effect on the piston-pin boss design.

Finally, the piston development was concluded by the testing of 87 engines made with production tools to set the clearance at 0.00025 in. closer than the existing standard.

Along with the foregoing development, both aluminum and malleable-iron pistons were injected into the picture, the Research Laboratory playing an important part in this program as far as recommendations were concerned.

Various ring combinations of number, width, shape and tension were investigated by tests which determined their blowby, oil control, and wear characteristics.

In this investigation the laboratories of the sources of supply augmented our own investigations, and the separating of the effects of the various designs on other problems called for close cooperation on the part of the engineers responsible for the various details.

Foundry Work Interesting

Aside from the engineering problems of combustion-chamber shape and spark-plug location, the most interesting work on the cylinder-head consisted of the cooperative work with the foundry in the determination of core assembly, wiring of cores, and sufficient sand at the lean spots in the head to permit proper venting. These problems, of course, are peculiar to the valve-in-head design, and are handled from the *beginning* of the design through the Production Engineer.

All samples of heads, as well as of cylinder-blocks, are furnished by the Production Foundry at Saginaw, and an arrangement is in force with the Foundry Management that no samples will be furnished for tests unless duplicate castings can be furnished by production methods at minimum scrap loss.

In Chevrolet this construction does not result in excess weight due to the extreme precision used in core manufacture and molding methods.

The Plant Production Engineer is responsible for all tests in connection with cleaning up complaints arising from foundry variations and must, in advance of production, determine what these variations are and test against them. In this work, all the facilities of the Organization are available to him.

In this development, the desire to reduce weight to the minimum was responsible for the introduction of many deflection tests, as well as tests at elevated combustion temperatures not before attempted; and they were tied up to the hold-down bolt location, the plug location and exhaust-port shape, as well as the combustion-chamber roof.

The development of the induction and exhaust system represents the greatest amount of contact of engineers within the Engine Group. And, inasmuch as the Engine Design Group has the maximum contact with the Chassis and Body Divisions, it, therefore, will be taken as an example of function and contacts.

Job of Carburetor Engineer

In the cylinder-head just mentioned, the volume and shape of the combustion-chamber, the location of the spark-plug, and the size and shape of valves and ports, are the responsibility of the Carburetor Engineer in the Engine Design Group.

In direct charge of Carburetor design, he works closely with the Bay City Carburetor Division through the Production Engineer at Detroit, and the Engine Plant Engineer at Flint, on all production problems in connection with carburetor design.

From the carburetor itself his responsibility extends to the inlet and exhaust manifolds, as far as flow, distribution, and warm-up characteristics are concerned. The Design Engineer is responsible for the incorporation of the preceding elements into a practical design, and works through the Production Engineer to accomplish this result.

In the theoretical studies the Project Engineer has the benefit of consultation with General Motors Research men engaged in studies of fuel and air-flow experiments. In the practical application our own Laboratory Flow Bench is used.

On the electrical end, in addition to the spark-plug and its location, he has the responsibility of determining the proper spark-advance curve for both part-throttle and full-throttle operation. These curves are worked out to some extent in the Laboratory before Proving-Ground development is attempted. In writing up the Work Orders for tests of this type, the Project Engineer is privileged to guide the course of the tests along the lines which, in his opinion, will develop the necessary data upon which to base the final design.

However, the test procedure to check the final results is the responsibility of the Experimental Engineer. Reports of this nature record the facts in comparison to previously accepted results, and are put in final form for distribution to the Engineering Executives.

The Electrical Engineer is responsible for the electrical characteristics of the coil, distributor, wiring, and so on, used in connection with the foregoing ignition problems. The Design Engineer correlates the design work of these two Project Engineers, and the Assistant Chief Engineer establishes their contact with suppliers within the Corporation dealing with such units and schedules their design and test work to agree with the general program.

All of the foregoing functions are carried on without the necessity of scheduled conferences, the Work Order and Reporting System of the Experimental Division keeping the entire Engineering Organization informed of the progress in time to avoid the necessity of conferences to either straighten out affairs or plan future programs.

In addition to the Laboratory and Proving-Ground tests, Road Tests in various parts of the Southwest and the Phoenix area are undertaken at two stages of the development.

The first tests are for the purpose of recording actual temperatures existing in all units where the effect of temperature has an important bearing on the result. Other characteristics, such as ride and handling, are observed as well, and a complete record of the day's run is furnished the executive in charge of the trip. This executive must be the Chief or Assistant Chief Engineer, Experimental, Design, or Production Engineer.

The Technical Engineer in charge of reports has no other duty.

The second tests are made *after* the development work has been completed, and represent the way the model is released to production, except as to appearance features.

The same type of personnel is used in this final check.

To those unacquainted with the various uses to which a product in large volume and world-wide distribution is subjected, it would seem that the protection afforded by the foregoing procedure would be sufficient. And it would be if all the operating conditions and demands upon the product could be foreseen and tested against. Realizing that they cannot be (and this condition is particularly true in our Commercial and Truck Models), the Engineering Department keeps in close touch with the field situation through its Service Contact Man.

This individual is service-minded, and is directly under the Chief Engineer. I bring up this activity on account of its relation to our Development Program.

Our Field Sales and Service Organization is divided into 49 Zones. The Service Organization in these Zones is responsible for the credit allowed for material and labor claimed as being defective by the Dealer in adjusting owner complaints under the terms of our Service Policy. The Engineering Department has an arrangement with Service whereby we review all such claims after they are coded properly and indexed, after which material and information as to operating conditions, and so on, is turned over to us upon request.

This information flows through the Service Contact Man to the Design and Plant Engineers for their comments and action. Copies, of course, go to the Executive Engineers.

The findings and comments of these men are passed back to the Service Contact Man for his guidance in issuing recommendations to the Central Office Service Department upon the authority of the Chief Engineer.

By this method the Engineering Department is in a position to make use of factual information in revising its test procedures to agree more closely with actual field conditions. The decision to do so rests with the Chief Engineer.

The Design and Project Engineers thus are acquainted with the problems encountered in special types of operation, and the Plant Engineers are in a position to take action through their Quality Check-Up system, provided the responsibility lies with the Manufacturing Department.

The gathering of the data and reviewing our past performances which were necessary to support many of the foregoing statements were extremely beneficial to the engineers engaged in this work, and we, ourselves, benefited greatly in reviewing the progress made by our Organization since our present method of operation has been in effect.

Proposed Method of Test for Ignition Quality of Diesel Fuels

(*Continued from page* 454)

10. Injection Indicator Contact Setting – The injection indicator contact setting shall be made as follows:

a. Adjust injection contact mounting bracket until there is about 1/32-in. clearance between the spring leaf and the injector pin.

b. Screw out center adjusting screw.

c. Adjust the lower leaf until it just touches the end of the injector pin. Then apply tension by turning the screw down one more complete turn.

d. Adjust the upper leaf by turning down the upper-leaf adjusting screw until the contact points touch. Be sure that the points touch as a result of the leaf-spring tension and not because of force applied downward by the adjusting screw. If this should be the case, it is necessary to remove the spring and bend it slightly downward at the outer end. The adjusting screw should then be turned up one complete turn from this position.

e. Adjust the bumper-spring tension in the center adjusting screw by turning down the bumper-spring adjusting screw as far as it will go.

f. Adjust the contact-point gap, with the engine running normally and the neon indicating switch closed, by screwing down the center adjusting screw until the neon tube just makes a solid band of light around the flywheel. Then turn back the center adjusting screw two divisions (0.002 in.) and lock by means of clamping screws.

11. Combustion Indicator Contact Setting – The combustion indicator contact setting shall be made as follows:

a. Screw out center adjusting screw.

b. Adjust the lower leaf until it just touches the end of the combustion indicator pin. Then apply tension by turning the screw down one more complete turn.

c. Adjust the upper leaf by turning the upper-leaf adjusting screw down until the contact points just touch. Be sure that the points touch as a result of the leaf-spring tension and not because of force applied downward by the adjusting screw. If this should be the case, it is necessary to remove the spring and bend it slightly downward at the outer end. The adjusting screw should then be turned up one complete turn from this position.

d. Adjust the bumper-spring tension in the center adjusting screw by turning down the bumper-spring adjusting screw as far as it will go.

e. Make preliminary contact-point adjustment by screwing down the center adjusting screw until there is a gap of approximately 0.010 in.

12. Final Combustion Indicator Contact Setting – The final combustion indicator contact setting shall be made as follows:

a. Carry out this setting for each test fuel.

b. Adjust compression ratio until engine fires normally on sample fuel. This is about two ratios above the lowest operating ratio without misfire.

c. Close neon indicating switch.

d. Check injection angle to make sure it is 13 deg. b.t.d.c. with engine operating on sample fuel. The injection neon tube is displaced 13 deg. on the flywheel to make it flash at t.d.c. when injection actually occurs at 13 deg. b.t.d.c.

e. Make preliminary compression-ratio adjustment until combustion, as determined by the "hook" method, occurs at top dead-center. The "hook" method is as follows:

The combustion indicator pin is restrained to follow the diaphragm by means of a leaf spring. The spring pressure is such that, for accelerations up to the maximum rate of pressure rise due to compression pressure, the pin is held on the diaphragm. Thus the gap between the indicator pin and the upper contact is an accurate measure of the deflections of the diaphragm. The position of the diaphragm is a function of the crank angle of the piston and, thus, the position of the upper contact pin determines the angular position of the neon flash.

Near the flat top of the compression pressure diagram, therefore, a very slight contact-point movement indicates a considerable flywheel angular movement and the neon-tube flash position is very sensitive to the contact-gap adjustment. But when combustion occurs, the very steep pressure rise of the diagram makes the neon flash position very insensitive to contact-gap adjustment. The transition from the sensitive to insensitive gap adjustment range is called the "hook" point.

The contact adjustment procedure is to turn up the upper contact adjusting screw from a closed position until the rapid movement of the flash suddenly changes to a slow movement for a given constant angular motion of the contact adjusting screw. This is the "hook" point, and the position of the neon flash indicates the start of combustion.

f. For final-combustion contact-point adjustment retard injection timing to 5 deg. b.t.d.c.

g. Adjust upper contact until two flashes of the neon bulb on the

(*Concluded on page* 484)

Hi-Duty Spark-Plug Testing

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AS an engine under development is not a tool which can be used for the testing and rejection of questionable spark-plugs, some test is necessary to parallel the engine development so that suitable spark-plugs are available for it when it enters service. This paper describes a preliminary single-cylinder engine test method for spark-plugs which evaluates the quality not only most essential but the most difficult to secure, namely, resistance to preignition.

Tests were made on two types of engine—a liquid-cooled, single-cylinder test engine and an air-cooled, single-cylinder test engine. Operating conditions were selected with regard for satisfactory performance plus the ability to heat the spark-plug.

A large number of plugs were tested, some of identical design, and many of them were re-run several times as a check on the reproducibility of the test method.

In addition to the preignition test, a full-scale engine type test, a ground fouling test, a flight fouling test, and a starting-ability test are recommended, with bench tests at intervals and measurement of electrode erosion and wear as a check on the probable useful life and ability of the spark-plug to measure up to the requirements of flight service.

IN the development of engines of steadily increasing power output, improvement in essential accessories has gone hand-in-hand with engine improvements which have permitted increases in power production. An engine under development is not a tool which can be used for the testing and rejecting of questionable spark-plugs. Some test is necessary to parallel the engine development so that suitable spark-plugs are available for it when it enters service. This paper is intended to describe particularly a preliminary single-cylinder engine test method.

[This paper was presented at the Metropolitan Section Meeting of the Society, Paterson, N. J., April 12, 1938.]

[1] A mixture-control curve at constant throttle consists of approximately ten equally spaced increments of fuel flow in pounds per hour, plotted against (1) brake horsepower, (2) head and (3) barrel temperatures, and (4) specific fuel consumption, at constant r.p.m. The curve is started at full-rich mixture and, at this mixture strength and each succeeding mixture strength progressively leaner, the data are taken after engine conditions have stabilized.

Following the introduction of shielded spark-plugs, very little was done to improve their design until the advent of engines of high specific output forced the development of spark-plugs capable of operating over a wider range of combustion conditions. Resistance to fouling under idling is still essential for a spark-plug which must perform continuously in flight in an engine at substantially higher b.m.e.p. than previously employed.

A great number of spark-plug designs were evolved by the vendors to satisfy the need for a better plug, and qualifying engine tests, focused on the essential qualities demanded, became an urgent necessity. A single short test could not evaluate all of the merits of a spark-plug so it became necessary to employ a test which would evaluate the quality not only most essential but most difficult to secure, namely, resistance to preignition.

Independent of length of life, economy of maintenance, first cost, and acceptability of shape, there is the prime consideration that a spark-plug must not create or encourage preignition under any condition of operation encountered in an engine for which it is selected.

Preignition may range from an occasional after-fire when the switch is cut during run, to a condition resulting in combustion pressure of destructive magnitude ahead of top dead-center. An annoying demonstration of preignition in an advanced stage is a complete breakdown of spark-plug insulation during a mixture-control curve[1], or at lean mixture, resulting in no ignition when the spark-plugs are cooled by enriching the mixture. A demonstration of spark-plug inadequacy is to have a mixture-control run halted because the preignition of spark-plugs masters normal combustion at what should be a satisfactory mixture strength.

Selection of Test Engine and Method

Assuming for the sake of the obvious economy that a single-cylinder engine is the desirable choice of measuring equipment for determining preignition character of spark-plugs, there are several methods to be used, individually or collectively, to bring about the pressures and associated temperatures which will develop preignition. They are for a selected engine: very high b.m.e.p., very high r.p.m., and high operating temperatures.

It is believed that extraordinarily high b.m.e.p. or very high r.p.m., although competent to produce preignition quickly, may introduce variables extremely difficult to correlate with service requirements.

A test for preignition in an engine must produce at the spark-plug a local (hot-spot) temperature high enough to ignite the charge. In the interests of reproducibility and equipment life, it is not desired to bring about preignition so early in the cycle as to evolve destructive pressure opposed to the piston movements, but rather to produce conditions under

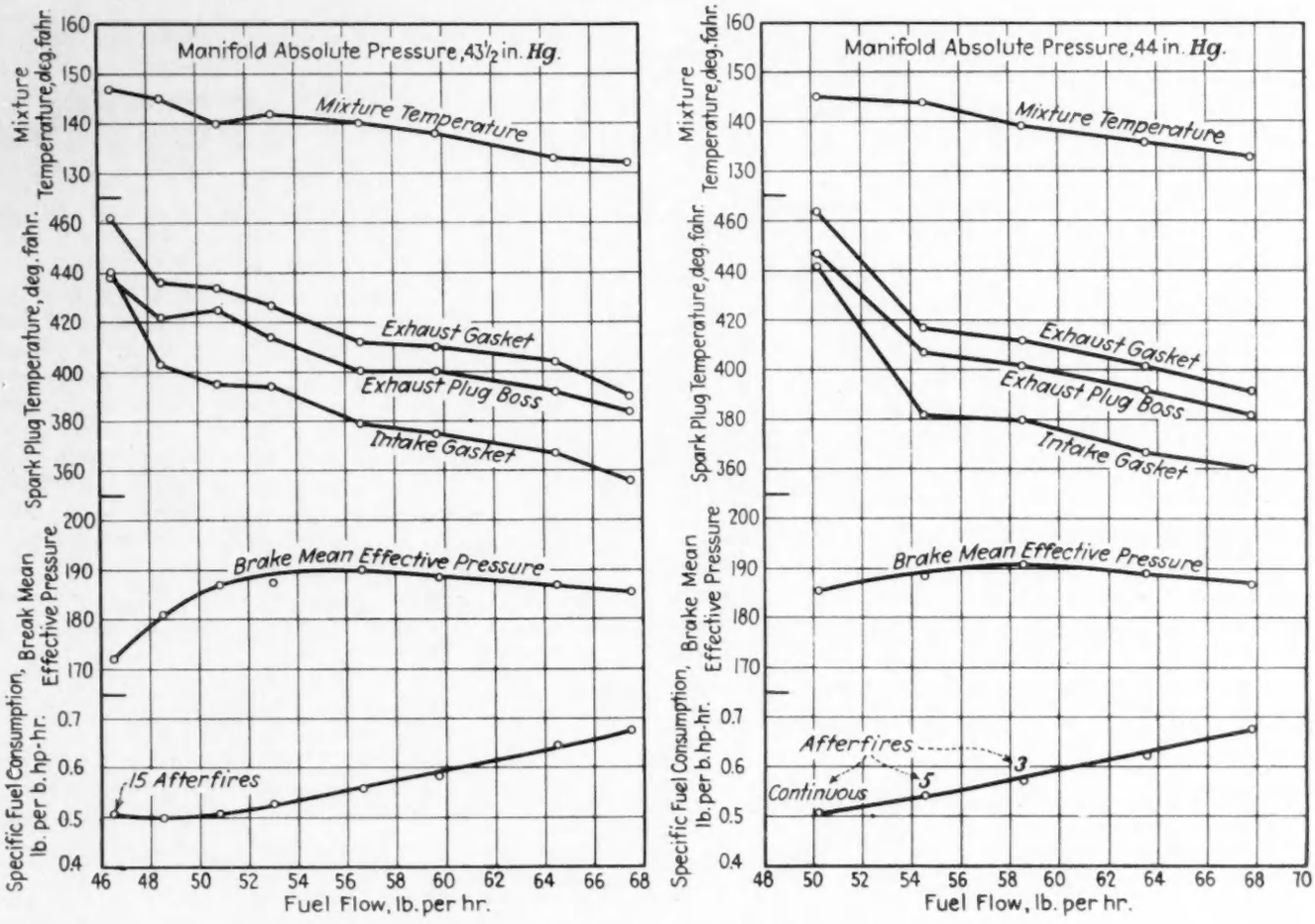


Fig. 1 – Spark-Plug Characteristics – Liquid-Cooled Single-Cylinder Test Engine
Compression Ratio, 6.5:1; Fuel, 87 Octane; 2400 R.P.M.; Carburetor Air, 150 Deg. Fahr.; Spark-Plug No. 14

which continuous steady after-firing will occur when the switch is cut.

Liquid-Cooled Single-Cylinder Engine

The engine formerly used for preignition tests of spark-plugs consists essentially of a liquid-cooled cylinder of 177 cu. in. displacement, 6:1 compression ratio (standard cylinder and head assembly from a full-scale engine), mounted on a large cast-iron crankcase incorporating a 6-in. stroke crankshaft. A conventional aluminum-alloy piston, 6⅛-in. diameter, was mounted on a special master rod fabricated for single-cylinder use.

Ethylene glycol was used as a coolant, circulated by a small pump driven from an accessory drive on the crankcase. The jacket liquid temperature was controlled accurately by means of a water-cooled heat interchanger. The supply tank was provided with a vent which afforded a means of detecting a leak in the cylinder barrel by appearance of vapor escaping from the vent.

A Stromberg carburetor, modified to work under pressure, was supplied with air previously compressed by means of a Roots-type supercharger, driven separately by a variable-speed electric motor. The temperature of the carburetor air was maintained at the desired level by means of a series of electric heating units in the duct between the supercharger and the carburetor. In addition to the electric units, a heat exchanger was mounted in the carburetor-air surge tank through which either water or steam could be passed as required. By this means accurate control of the carburetor-air temperature was possible. Fuel to the carburetor was supplied by a standard aircraft fuel pump with a bypass valve in the discharge line leading to the inlet for control of pressure.

Test Method

Inasmuch as the engine just described was to be used for preignition tests of spark-plugs, operating conditions were selected with regard to satisfactory performance plus the ability to heat the spark-plug. The following conditions finally were selected by elimination tests. These conditions are shown later compared with the air-cooled single-cylinder test-engine conditions.

1. R.P.M., 2400 ± 10.
2. Coolant outlet temperature, 250 $^{+10}_{-0}$ deg. fahr.
3. Temperature of previously compressed air entering the carburetor, 150 ± 2 deg. fahr.
4. Oil inlet temperature, 165 ± 5 deg. fahr.
5. Ignition advance, inlet 30 deg., exhaust 33 deg.
6. Fuel, 87 octane.

Item 1 was selected at the relatively high value of 2400 r.p.m. since the engine performed satisfactorily at this speed over a considerable range of power output and the time afforded the spark-plug for cooling is reduced with an increase in r.p.m.

Item 2 was maintained within close limits at the value of 250 deg. fahr. for satisfactory engine operation and held constant since the initial temperature of the spark-plug inserts and bosses was directly dependent on the temperature of the cooling liquid.

Items 3 and 4 were in the range of satisfactory engine operation although the former, inasmuch as it affected mixture temperature, had a direct bearing on the preignition rating.

Item 5 was selected as normal for satisfactory engine oper-

ation. Item 6 affects the preignition rating of a spark-plug since it affects the detonation characteristics throughout a mixture-control curve and, therefore, has some effect on the heating of the spark-plug.

It was necessary to maintain close limits on all operating conditions, since any variation in any condition that might affect head and spark-plug temperatures would affect the preignition rating of the spark-plug under test. Also, it is probable that a single-cylinder engine is inherently more sensitive to slight changes in operating conditions than a full-scale engine and, therefore, requires closer limits for all controlling conditions to insure comparable results.

It was considered desirable to run preignition tests by a series of complete fixed-throttle mixture-control curves of power output, measuring temperatures, and other related conditions at a series of fuel flows at constant r.p.m. Each succeeding mixture-control curve was run at higher manifold absolute pressure and b.m.e.p. Each curve was extended to a setting where the curve of specific fuel consumption vs. fuel flow hooked upward at the lean end, that is, where rapidly decreasing power resulted in increasing specific-fuel-consumption values. The final curve run would be limited before this point by violent detonation or constant preignition.

The first mixture-control curve was run at a manifold absolute pressure and b.m.e.p. which did not cause preignition or detonation with the plug under test. Each succeeding mixture-control run was made at a manifold absolute pressure higher by ½ in. hg. than the preceding curve. At each setting of the mixture control the switch was cut after the usual instrument readings were taken, and the number of after-fires counted. After a mixture-control run was continued until constant preignition or excessive detonation was noted, the manifold absolute pressure was reduced by ½ in. hg., and a mixture-control run was again made at this slightly lower output with the usual observations and counting of after-fires. The spark-plug was rated for preignition at the b.m.e.p. and manifold absolute pressure (expressed in these terms) representing a manifold absolute pressure ½ in. hg. below that at which constant preignition or potentially destructive detonation was encountered.

It was believed desirable to run under the described conditions in preference to constant specific fuel consumption at progressively increasing power output. This latter method involves maintaining a condition constant which is not directly obtainable without computation, and for which the fuel flow cannot be predetermined accurately for the given specific fuel consumption.

In practice, the engine operator must adjust both throttle and mixture control in raising the power output at constant specific fuel consumption. Opening the throttle alone usually produces a leaner mixture at the higher power output and adjustments of the mixture control must follow. If, when the throttle is opened, the mixture is leaned out, the temperature increases throughout the engine may be more rapid than normal. If the mixture is enriched at the same time the throttle is opened, excessive fuel cooling may result.

In either of the described cases an orderly progress of increased temperature with increased severity of operating conditions requires an extremely skilful engine operator. Any variation in equipment or conditions producing a leaner mixture for any period of time might very easily introduce an error in the preignition rating value due to the higher temperatures introduced by the lean mixture, before the condition was corrected.

Experience has shown that more consistent results can be obtained by running complete mixture-control curves although the period of time required is much greater than for a curve run to a selected minimum specific fuel consumption. It is believed that complete curves afford a more desirable test as regards the ability of the spark-plug to withstand operation at very lean mixtures, this condition being much more severe than operation on a relatively rich mixture as is implied by constant specific fuel consumption. The spark-plug in actual service operation will operate under conditions similar to the mixture-control-curve method although, under normal circumstances, conditions will never be as severe as during the test.

Validity of the Method

The described test conditions were established as a result of an extensive preliminary survey of available test methods. Each variable of the test perforce was evaluated separately, and the best compromise between the several variables selected.

Following a tentative establishment of the method, a large number of spark-plugs were tested, many of them re-run several times as a check on the reproducibility of the test for a given spark-plug. By the cooperation of some of the spark-plug manufacturers, it also was possible to test several spark-plugs of identical design, varying only in the thermal conductivity of the conductors employed. Tests of this type enabled us to verify that the method was sensitive to small but important changes in spark-plug materials of construction. With the qualifications with respect to precision described in the next paragraph, the validity of the method seemed to be determined adequately.

Precision of Controls and Stabilizing Conditions

The importance of proper design and competent stabilizing of conditions cannot be overemphasized in this type of testing where temperature-producing preignition determines the results of the test. Controls must be sensitive to provide a fine adjustment at all outputs likely to be encountered with due consideration for a wide range of ambient-temperature conditions. Lost motion should be reduced to a minimum with all backlash in controls in one direction so that engine conditions will not be affected by it.

With any type of engine testing, conditions must be stabilized at any one setting of the controls to insure accurate instrument readings and valid data. With single-cylinder testing this item becomes doubly important since very slight changes in certain variables, such as carburetor-air temperature or coolant flow or temperature, may promote appreciable changes in results. Any variable that affects any phase of the engine operation or results should be controlled within the closest practical limits.

It is probable that the time for a single test is lengthened

Engine	Shielded Yes	Shielded No	Reach Long	Reach Short	Single Cylinder MAP	Single Cylinder No.	Single Cylinder BMEP Lbs.	Full Scale Fouling Test Yes	Full Scale Fouling Test No	Full Scale Endurance Yes	Full Scale Endurance No	Approved Yes	Approved No	
1820 G	x		x		41		150	x		x				Required
	x		x		42		184	x		x		x		
	x		x		40		178		x		x		x	
	x		x		41		175		x		x		x	
	x		x		41		181	x		x		x		
	x		x		42		179		x		x		x	
	x		x		43½		191	x		x		x		
	x		x		42½		188		x	x			x	
	x		x		42½		186	x		x			x	
	x		x		42½		188		x	x			x	
	x		x		39½		161		x		x		x	
V 1570		x	x		39		161	x		x		x		Required
		x	x		39		161	x		x		x		
		x	x		38½		157		x	x			x	
		x	x		39½		170	x		x			x	
1820 F 60	x			x	39		161	x		x		x		Required
	x			x	42½		182	x		x		x		
	x			x	38		164	x		x		x		
	x			x	41½		177		x		x		x	Penalty
	x			x	40½		172		x	x			x	
	x			x	41½		179		x		x		x	Penalty
1820 F	x			x	37		150	x		x				Required
	x			x	37		150	x		x		x		
R975E	x	x		x	36		145	x		x		x		Required
	x			x	36		145	x		x		x		
		x		x	36½		148	x		x		x		
		x		x	34½		134.5		x	x			x	
	x		x		-		-	x		x		x		

Fig. 2 – Test Results on a Number of Spark-Plugs of Concealed Identity

NEW PLUGS				PLUGS AFTER RUNNING						
Plug	Capacity	Ionized Gap Test	Core-Gas Leakage	Capacity	Ionized Gap Test	Core-Gas Leakage	Average Electrical Erosion	Bomb After Recond., at 150 lb. per sq. in.	Total Time, Hr.	
B	55	O.K.	None	74	F½	None	0.035	O.K.	259:25	Spark-plug cores satisfactory for additional testing after refacing center electrode to original contour and shells satisfactory after replacing baffle (ground electrode).
"	53	F-1	"	71	F½	Trace	"	"	"	
"	50	F-1	"	73	F2½	0.4	"	"	"	
"	52	F-2	"	74	F½	None	"	"	"	
"	53	F-½	"	73	F½	"	"	"	"	
"	57	F-5½	"	75	F1	"	"	"	"	
"	52	O.K.	"	73	F½	"	"	"	"	
"	53	F-1	"	70	F1	"	"	"	"	
"	55	F-5	"	73	P02	"	"	"	"	
"	53	O.K.	"	74	O.K.	"	" (512:00 Hr.)	"	"	
A	54	O.K.	None	49	O.K.	Trace	0.023	O.K.	177:10	Electrodes eroded to such an extent that further testing was not considered desirable.
"	52	"	"	53	F5½	None	"	"	"	
"	62	"	"	51	O.K.	0.7	"	"	"	
"	60	"	"	45	"	Trace	"	"	"	
"	47	"	"	47	F6	0.2	"	"	"	
"	47	"	"	54	F6	None	"	"	"	
"	51	"	"	51	O.K.	None	"	"	"	
"	45	"	"	45	F6	Trace	"	"	"	
"	48	"	"	51	F5	1.5	"	"	"	
"	62	"	"	48	F5½	1.4	"	"	"	
"	51	"	"	52	F4½	0.9	"	"	"	
"	52	"	"	50	F5½	1.0	"	"	"	
"	63	"	"	45	F5	0.2	"	"	"	
"	47	"	"	54	F6	0.9	"	"	"	
"	48	"	"	44	O.K.	Trace	"	"	"	
"	69	"	"	50	O.K.	0.2	"	"	"	
"	50	"	"	50	F6	None	"	"	"	
"	56	"	"	54	O.K.	None	"	"	"	
A	45	O.K.	None	71	O.K.	None	0.028	O.K.	239:30	Further testing not recommended due to eroded shell baffle electrodes.
"	41	"	"	47	F4½	1.4	"	"	"	
"	44	"	"	65	F4½	0.5	"	"	"	
"	44	"	"	102	F4½	0.7	"	"	"	
"	43	"	"	100	O.K.	4.2	"	"	"	
"	43	"	"	67	F5	3.8	"	"	"	
"	41	"	"	110	F5	1.5	"	"	"	
C	-	O.K.	-	32	O.K.	Trace	0.030	O.K.	193:50	One spark-plug shell failure during test. Additional testing not recommended.
"	"	"	"	33	O½	None	"	"	"	
"	"	"	"	33	O.K.	0.7	"	"	"	
"	"	"	"	34	O.K.	0.5	"	"	"	
"	"	"	"	34	O.K.	0.7	"	"	"	
"	"	"	"	33	O.K.	0.7	"	"	"	
"	"	"	"	31	O.K.	Trace	"	"	"	
"	"	"	"	32	O.K.	0.7	"	"	"	
"	"	"	"	33	O.K.	None	"	"	"	
"	"	"	"	32	O.K.	1.4	"	"	"	

Fig. 3 – Spark-Plug Bench-Test Data – New and Used Plugs

considerably by running at a given setting until all temperatures, pressures, and so on, have attained equilibrium. This procedure is essential, however, as experience has shown that results will vary considerably if instruments are read and values recorded before equilibrium temperatures (particularly) are attained. It is also desirable to spot evenly spaced settings, approximately 10 in number, for each mixture-control curve completed, as an element of time affects the temperature levels at which equilibrium is reached. Data secured in this type of testing are shown in Fig. 1.

Incidental Inspection Data

Changes in the condition of the spark-plug as a result of running in the single-cylinder engine preignition test, which may qualify or explain the rating it develops, are best measured by bench tests. Several tests used which are not directly pertinent to the preignition test will not be discussed here.

Two essential tests, the bomb test and the core-leakage test, require mention. The bomb test consists essentially of a pressure-tight fixture into which the spark-plug may be screwed, and the spark gap (previously set to normal gap setting) subjected to 150 lb. per sq. in. gas pressure, using a comparatively inert gas, preferably nitrogen. Current, supplied by a service-type magneto delivering 750-1050 sparks per min., is impressed on the gap (under the gas pressure) and the firing, if any, is observed through a quartz window in the bomb.

Gas leakage is determined in another pressure fixture where the plug is submerged in water or kerosene above the gasket seat with gas pressure applied to the combustion-chamber side. Leakage in the joint between the core and the shell or through the core at 150 lb. per sq. in., as indicated by the water displaced, should be limited to ½ cc. per min. for a new plug and 2 cc. per min. for a plug after completion of the preignition test. The spark-plug, before and after the preignition test, must fire satisfactorily at 150 lb. per sq. in. in the bomb. Bomb tests, to be reproducible and reliable, must be conducted on spark-plugs previously dried in an oven for at least 2 hr. at 250 deg. fahr., although they also are tested invariably just as removed from the engine to determine if they are in operating condition.

Additional Testing for Acceptability

The single-cylinder preignition test is only a preliminary to a series of tests to determine the acceptability of Hi-Duty spark-plugs.

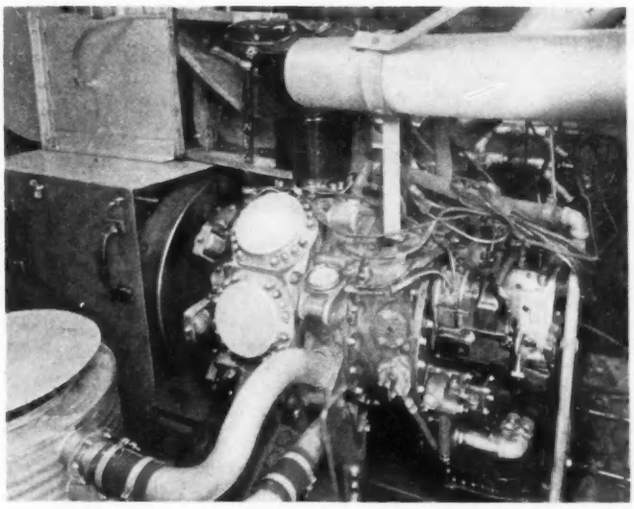


Fig. 4 – Air-Cooled Single-Cylinder Test Engine

A full-scale engine type test, a ground fouling test, and a flight fouling test must be completed successfully to determine the spark-plugs' acceptability, with bench tests at intervals and measurement of electrode erosion or wear as a check on the probable useful life and ability of the spark-plug to measure up to the requirements of flight service. A starting-ability test to determine the tendency of the plug to foul and fail to fire when starting the engine at low temperatures also seems essential to complete the measure of its acceptability. Such a test is contemplated for early adoption.

The single-cylinder engine preignition test determines, for a given spark-plug, the engine series for which it may be suitable when all other required tests are completed satisfactorily. Fouling tests, or subsequent failures, or excessive electrode erosion may qualify the selection based only on the preignition test. Full-scale engine tests under a wide range of conditions have shown, for each series engine, the minimum preignition rating for a suitable spark-plug for that series.

Fig. 2 is a tabulation of the test results on a number of spark-plugs, identity concealed, showing the preignition rating previously described, the engine series for which they are deemed suitable when satisfactory tests are completed, and their present status. Despite the large number of plugs shown here, they are few compared with the number actually subjected to the preliminary preignition test.

Shown in tabulated form in Fig. 3 are typical data for a set of spark-plugs before and after completing the full-scale engine type test. Failure of more than 10 per cent of the plugs in the engine or as indicated by bench tests after running is sufficient cause for rejection of the model under consideration.

Air-Cooled Single-Cylinder Test Engine

Partly as a means of utilizing test equipment readily available to any organization interested in single-cylinder engine spark-plug testing, the test engine has been supplanted by an air-cooled unit in which the power section is made up of conventional Series G Cyclone parts. Continued experience has emphasized the desirability of making one consideration paramount, namely, that the type of engine and engine conditions used for preignition testing approach as closely as possible to service under the most severe conditions which the spark-plug may ever be expected to encounter. The choice of the G Series Cyclone cylinder and piston, the most widely used of our production, satisfies the foregoing requirements with respect to the engine.

Simplicity of installation of the air-cooled cylinder makes possible a ready exchange of a cylinder with a short-reach spark-plug insert for one with a long-reach insert, or vice-versa, with a minimum of labor. Details of spark-plug insert, spark-plug, and electrode location follow the exact practice used in the same model engine in service. The method previously employed has been retained without change except for engine conditions.

The present air-cooled preignition test engine varies in several respects from the liquid-cooled engine previously described. It is illustrated in Fig. 4. In the present unit a full-scale engine steel crankcase is used, the cylinder being mounted on No. 1 cylinder pad. The remaining pads are covered by plates, and the remaining valve-cam followers are spring-loaded. All knuckle pins are installed in the master rod, and the multicylinder engine crankshaft is re-balanced for the actual weight. A standard piston and cylinder assembly is used in this installation giving a displacement of 202½ cu. in. and a compression ratio of 6.7:1.

Fuel injection is employed as a means of metering the fuel supplied for combustion. A United American Bosch fuel injector supplies fuel to a poppet-type valve located in the intake pipe. Induction-system air is supplied by a Roots-type

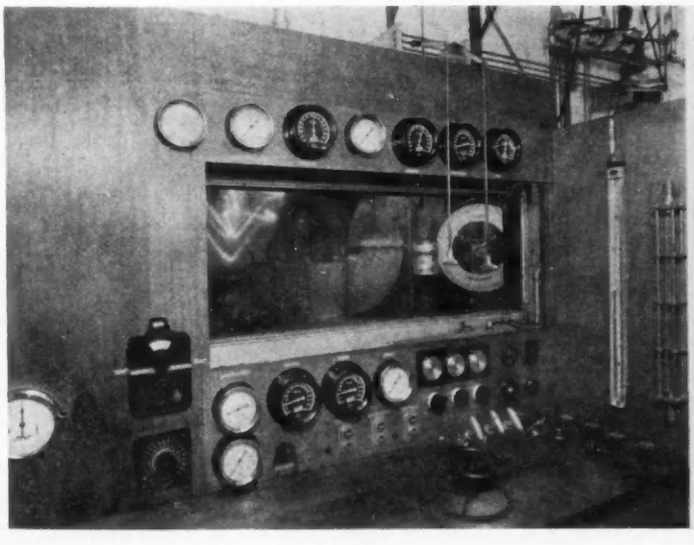


Fig. 5

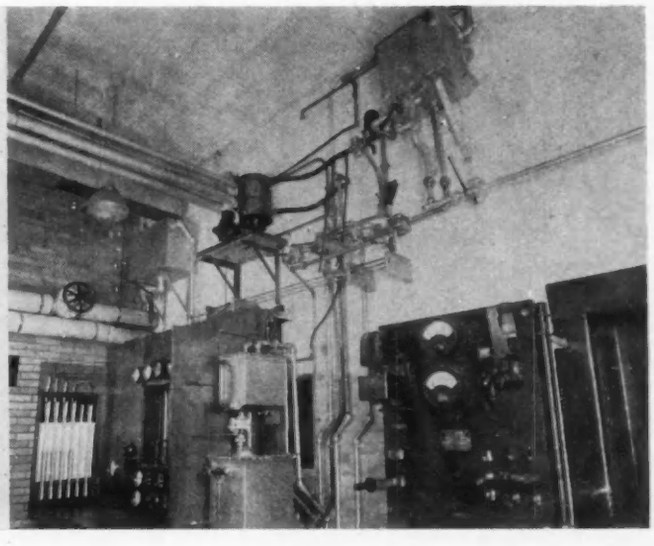


Fig. 6

Figs. 5 and 6 –Engine and Dynamometer Controls

blower driven by an electric motor and the degree of supercharge controlled by a butterfly valve in the air duct. A surge tank is used to reduce pulsations. The impeller section and engine rear section are mounted on the crankcase, but no impeller is provided for supercharge. The rear section is utilized for accessory drives only.

Ignition is supplied by two Scintilla nine-cylinder magnetos mounted on the engine rear section. Eight distributor terminals are grounded in each magneto, and the spark to the plugs is supplied by the remaining terminals in each magneto. Spark timing is 15 deg. for the rear spark-plug and 20 deg. for the front spark-plug.

Cooling air on the cylinder assembly is supplied by a conventional blower, the air duct extending to a comparatively simple sheet-metal baffle arrangement on the cylinder assembly. The total air flow is controlled by a butterfly valve in the duct. In addition, directly in front of the cylinder assembly a horizontal baffle is used to separate the air flowing on the cylinder-head and cylinder barrel. The air to the cylinder-head then is controlled independently of that to the barrel by a second butterfly valve. This arrangement affords practically any combination of baffle pressure (cooling air flow) desired on the cylinder barrel and head. It also provides a means of running with a cool cylinder barrel and a relatively high head temperature, thereby increasing piston-ring life and initiating preignition in a spark-plug at a lower output than otherwise would be possible.

The engine and dynamometer controls are shown in Figs. 5 and 6. Fig. 5 includes, lower row, left to right, a blowby meter, potentiometer, fuel pressure gages, oil-temperature thermometers, oil pressure gage, electric tachometer, revolution counter, electric stop clock, throttle and mixture controls. In the upper row are fuel head gage, fuel temperature thermometer, carburetor and cooling-air temperature thermometers. At the left end are shown the manifold pressure manometer and the rotometer indicating fuel flow.

Fig. 6 is a view from the opposite end of the control bench, showing particularly manometers measuring air pressures in various locations in the cooling air duct and blowby pressure. The fuel-measuring tank may be seen overhead. Except when actually measuring fuel consumption by weight, the fuel level in the scale tank is maintained by a float-controlled solenoid valve shown in black to the right of the scale tank, located on the fuel-supply line. Oil tanks and dynamometer control panel are shown at the right.

Engine Conditions

Engine conditions of operation are tabulated below in comparison with the previously used liquid-cooled engine:

	Liquid-Cooled Engine	Air-Cooled Engine
R.P.M.	2400	2000
Oil-in Temperature	165 ± 5 deg. fahr.	140 ± 5 deg. fahr.
Fuel	87 Octane A.S.T.M.	87 Octane A.S.T.M.
Ignition Advance	Inlet 30 deg.	Front Spark-Plug, 20 deg.
	Exhaust 33 deg.	Rear Spark-Plug, 15 deg.
Induction Air Temperature after Compression	150 ± 2 deg. fahr.	130 ± 2 deg. fahr.
Coolant	Coolant Outlet Temperature, 250 + 10 deg. fahr. — 0	Temperature at Definite Location on Cylinder-Head at Start of Curve, 380 ± 2 deg. fahr.

Test Data

A typical spark-plug preignition test employing the present engine is illustrated in Fig. 7, namely, temperatures, b.m.e.p., and specific fuel consumption plotted versus fuel flow.

Constant preignition was developed during the mixture-control curve at an output of 36 in. hg. manifold absolute

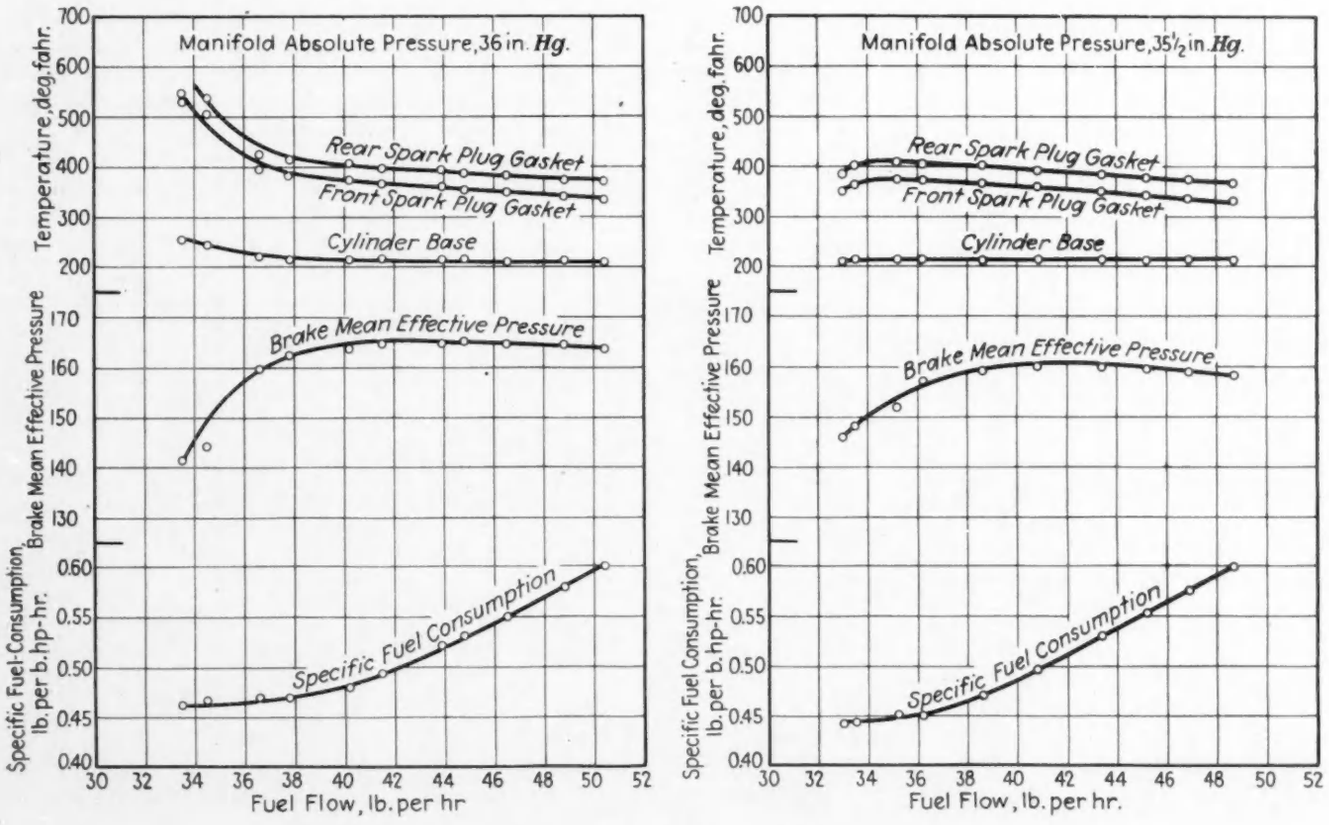


Fig. 7 – Spark-Plug Characteristics – Air-Cooled Single-Cylinder Test Engine
Compression Ratio, 6.7:1; Fuel, 87 Octane; Speed, 2000 R.P.M.; Carburetor Air, 130 Deg. Fahr.

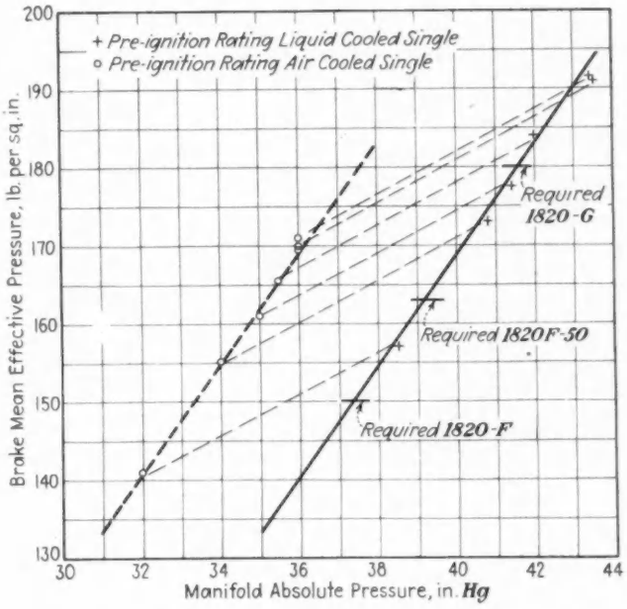


Fig. 8 – Manifold Absolute Pressure Vs. B.M.E.P. for Series of Identical Plugs Tested in Both Liquid-Cooled and Air-Cooled Single-Cylinder Test Engines

pressure. Spark-plug temperatures increased steadily as the mixture was leaned, and preignition in the spark-plug was evident at a rear spark-plug-gasket temperature of 545 deg. fahr., this preignition occurring before the curve of specific fuel consumption had hooked upward at the lean end. Due to the excellent cooling of the cylinder barrel, the cylinder-base temperature rise is small compared to spark-plug-gasket temperatures.

With the output decreased by ½ in. hg. manifold absolute pressure, a sharp distinction in the shape of the temperature curves is noted. Although the specific fuel consumption curve is not extended to the hook, it is extended to a setting at which the temperatures definitely have started to drop off due to a decrease in power and severity of detonation. Due to the steadily decreasing temperatures at progressively leaner mixtures, preignition was not encountered at this output and this curve develops the rating of the plug tested, that is, 161 lb. per sq. in. b.m.e.p. at 35½ in. hg. manifold absolute pressure.

Correlation of Test Data

A difference in the manifold absolute pressure and b.m.e.p. for a given type of spark-plug may be noted between tests on the liquid-cooled and air-cooled single-cylinder engines as shown in Fig. 8. Rating values for manifold absolute pressure versus b.m.e.p. are plotted for a series of identical plugs tested in both engines. For the two test engines it will be noted that the values fall in the same order for any spark-plug; the differences are substantially uniform and capable of translation in terms of the results on either engine.

Equipment Life

With the described changes in test equipment, the method of determining preignition, it will be noted, is retained from the practice with the liquid-cooled unit described earlier in the paper. As might be anticipated under the severe conditions of test operation with both detonation and preignition for a large part of the running time, engine operating time between overhauls is comparatively short. The difference between liquid-cooled and air-cooled operation is negligible in this respect. After the evaluation of several plugs suitable for high-output engines, piston-rings must be replaced and, from time to time, new pistons and cylinder barrels must be installed.

The relatively short life of engine parts in this testing has led to an active effort to modify the method so as to increase the operating time between overhauls. Many variations of equipment design and modifications of the test method were tried to evaluate possible increases in equipment life for equally valid test data.

Cooling air flow restrictions were tried in a number of combinations of air flow over cylinder-head and barrel. The combination just described was finally adopted.

A method involving stopping the mixture at some critical specific fuel consumption before completing the hook again was evaluated and discarded as providing unreliable data.

To encourage preignition at lower b.m.e.p. a high-compression piston was tried but found to be more destructive to rings while not developing preignition.

As a means of increasing local head temperatures and restricting the cooling of the spark-plug-gasket seat, fins immediately surrounding the spark-plug-gasket seat were planed off flush with the cylinder-head. This change, although it failed to bring the spark-plug to preignition temperature earlier, did produce a local hot-spot apparently in the spark-plug insert itself.

Extensive tests were conducted using but one spark-plug in the conventional cylinder-head. As possibly might have been expected, this method also produced a local hot-spot at the approximate location of the last part of the charge to burn. This condition resulted in a measure of engine preignition instead of spark-plug preignition.

The latest contribution to this series of experiments is the employment of a multi-ring piston from which better cooling and longer ring life are anticipated. This unit is now on test. The results of this work will be available at an early date.

It is believed that the foregoing discussion has established the need of a test of the type described.

In pointing out the limitations of the liquid-cooled engine test, it is not implied that the results secured therein are involved, but rather that the tests with the air-cooled engine may be expected to survive as a measuring device through still higher ranges of power output and b.m.e.p. It is more directly comparable with engines of immediate interest.

Although the test described is admittedly somewhat expensive to carry out, it does obviate the possibility of a full-scale engine in service encountering disastrous spark-plug preignition trouble without an effective substitute spark-plug available.

It is claimed for the test, based upon substantial experience, that design and material weaknesses of spark-plugs are developed frequently that otherwise would be obscure or possibly only revealed after many hours of flight service in a full-scale engine. An example of this type is core-gas leakage which is exposed readily in this type of test, whereas it is sometimes an obscure but dangerous phenomenon in flight service.

Scope of the Test

While the equipment shown and method described are far from ideal, they have the merit of producing practical results. Spark-plugs in flight service have shown satisfactory correlation with the test in the sense that plugs rated by it have thus far failed to show any evidence of preignition or of inducing detonation in the engines for which they are approved.

Pending the development of more scientific methods, the described test, or a modification of it, will be an essential tool for spark-plug evaluation and the elimination of unsatisfactory types.

Supercharger Installation Problems

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THAT the efficiency and performance of superchargers on aircraft engines depend heavily upon the character of the particular installation, is the authors' theme. Accordingly, they stress the application of superchargers and their accessories to engine-airplane installations rather than the design details of the superchargers themselves. They evaluate the relative influence of supercharger inlet temperature, supercharger inlet duct pressure drop, and intercooler pressure drop. Four disadvantageous results of high intercooler pressure drop are listed.

An important result of a study of the power required to drive superchargers for large engines is a formula for "installation efficiency," which the authors have established as a barometer of installation excellence.

Two much-discussed problems of supercharging are considered – the method of driving the centrifugal compressor and the location of the carburetor with respect to the supercharger. Reporting the results of laboratory tests, the authors announce that it is possible to obtain take-off power by the use of an exhaust-driven supercharger.

THE success of an engine-supercharger combination for aircraft use is dependent upon how well the airplane installation is made, to a far greater extent than is realized generally. Therefore, the authors propose to discuss the application of superchargers and their attendant accessories to engine-airplane installations and will not, because of military policy, discuss the detail design of the superchargers themselves.

It is conceded quite universally that, in the present state of the art, the centrifugal type of compressor is more adaptable to aircraft-engine usage than any other type because of the large volumetric capacity obtained with a relatively small size and low weight. The displacement type suffers from bulk and weight, and these factors are emphasized as engines approach the 1500-hp. sizes. It rarely is appreciated by those not directly concerned with aircraft-engine development that an engine producing 1500 hp. consumes approximately 5 tons of air, or five times its own weight, for every hour of operation.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 13, 1938.]

The two problems of supercharging that have received much attention have been, first, the method of driving the centrifugal compressor and, second, the location of the carburetor with respect to the supercharger.

Method of Drive

The supercharger may be driven by any of four common methods:

1. Geared directly to the engine, with a fixed speed ratio, as shown in Fig. 1.
2. Geared to the engine through multispeed gears and clutches.
3. Gearing of two or more stages of supercharging to the engine, as shown in Fig. 2.
4. Utilizing the exhaust gas for driving the compressor, by means of a turbine, as shown in Fig. 3.

Carburetor Location

Methods 1 and 2 generally have the carburetor on the inlet side of the supercharger, whereas Methods 3 and 4 usually have the carburetor between the engine and the major portion of the supercharging. This arrangement offers the decided advantage of facilitating cooling of the compressed air by a heat exchanger before introduction to the engine and is

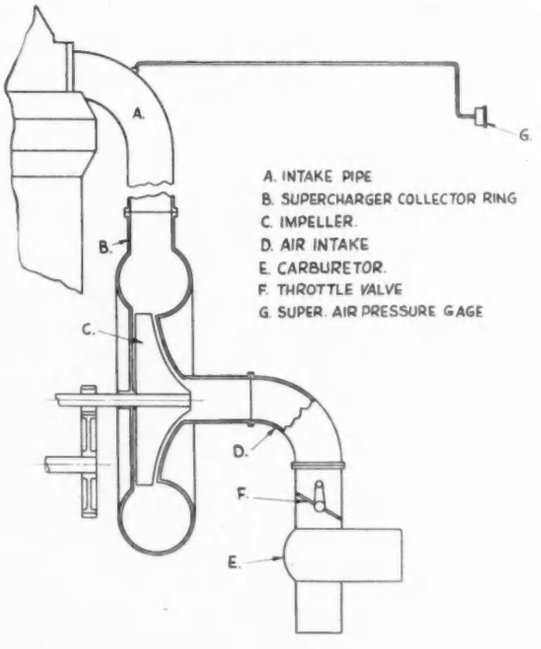


Fig. 1 – Diagram of Geared Supercharger – Single Stage

one of the chief reasons why these latter two methods of drive are favored for the higher altitude applications at critical altitudes greater than 16,000 ft. It may be possible to cool a mixture of fuel and air after having passed through the compressor by proper introduction of backfire screens and proper design, but the authors are of the opinion that, under violent backfiring, an intercooler of better construction than any known to date will be required to withstand the very high pressures which would be encountered in the induction system.

Since the geared-type superchargers have had so much more extensive application than the exhaust-turbine-driven type, the remainder of the paper will be devoted to the installation problems of the latter type, although much of the data are applicable to the geared-drive application.

Fig. 3 shows diagrammatically the operation of the exhaust-turbine-driven supercharger. The exhaust from the engine is conducted through manifold *C* to the nozzle box *J*, which

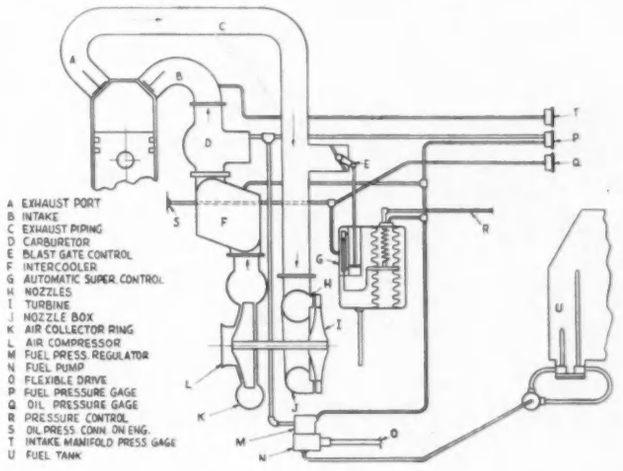


Fig. 3 – Diagram of Exhaust-Turbine-Driven Supercharger

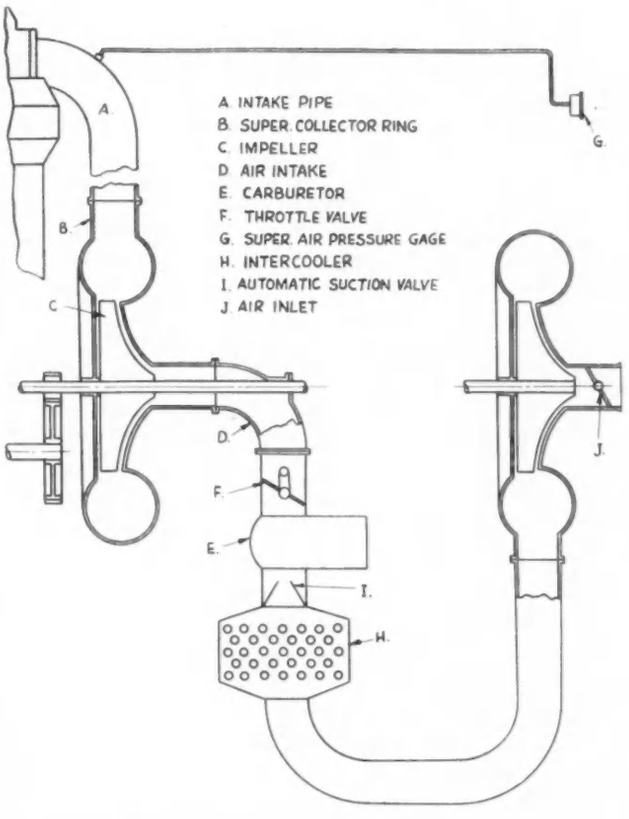


Fig. 2 – Diagram of Geared Supercharger – Two Stages

directs the gases at high velocity into the turbine buckets on wheel *I*. The speed of the wheel is regulated by blast gate *E* which, in turn, is controlled by the automatic regulator *G*. The engine air is taken into the compressor through suitable ducts at *L* and, after compression, it is delivered to intercooler *F* for cooling before entering the carburetor *D* and thence to the engine. The fuel pump is submerged with respect to the fuel supply, by remote drive from the engine to prevent vapor lock which would accompany high-altitude full-power operation.

Regulation

Regulation of the exhaust-turbine-driven supercharger is by means of an automatic regulator of the balanced-diaphragm type with hydraulic servo operating mechanism which can be made to operate at a constant pressure at all altitudes or can be arranged so that the pilot can adjust the pressure to meet operation requirements. The controlled pressure, in both cases, is that pressure immediately preceding the carburetor and just after the intercooler. In no case is it the intake manifold pressure between the carburetor and the engine. In the case of constant-pressure control, the pilot control is limited to the engine throttle and the automatic control tends to maintain sea-level pressure immediately preceding the carburetor regardless of throttle position. This type of installation offers the advantage of an automatic mixture control. If the carburetor has been adjusted properly for sea-level conditions, the only tendency toward enrichment with altitude is that resulting from any increase in carburetor air temperature. When the regulator is installed so as to provide infinite pressure control from the cockpit, the pilot either can set the regulator to maintain a constant carburetor air pressure, or he can select the pressure at which he desires to operate. In the latter case the carburetor either must be equipped with an automatic mixture control or the pilot must use a manual means for obtaining a proper fuel-air mixture ratio. In this type of installation, individual controls are required for the throttle and the regulator.

Power Required for Supercharging

It is important that the amount of power being expended in supercharging the modern aircraft engine be kept in mind to appreciate the importance of a good installation and realize the power wasted in a bad installation. Fig. 4 shows the power required for 1200-hp. and 2000-hp. engines plotted against pressure ratio for different supercharger inlet air temperatures. These curves assume a compressor outlet pressure of 29.92 in. hg., 65 per cent adiabatic efficiency and that the engine consumes 0.110 lb. of air per b.hp. per min. The 1200- and 2000-hp. engines were chosen because they represent an existing size, and a size which may be expected soon. On this figure, four points have been selected showing the power required at 20,000, 25,000, 30,000 and 35,000 ft. with normal atmospheric temperature at the supercharger inlet, and four other points at the same altitudes with the air to the compressor heated to 50 deg. fahr. above the normal atmospheric temperature for that altitude.

Considering the 30,000-ft. points, it will be noted that the 1200- and 2000-hp. engines require 196 and 326 hp. to drive the supercharger. If, however, air to the supercharger is

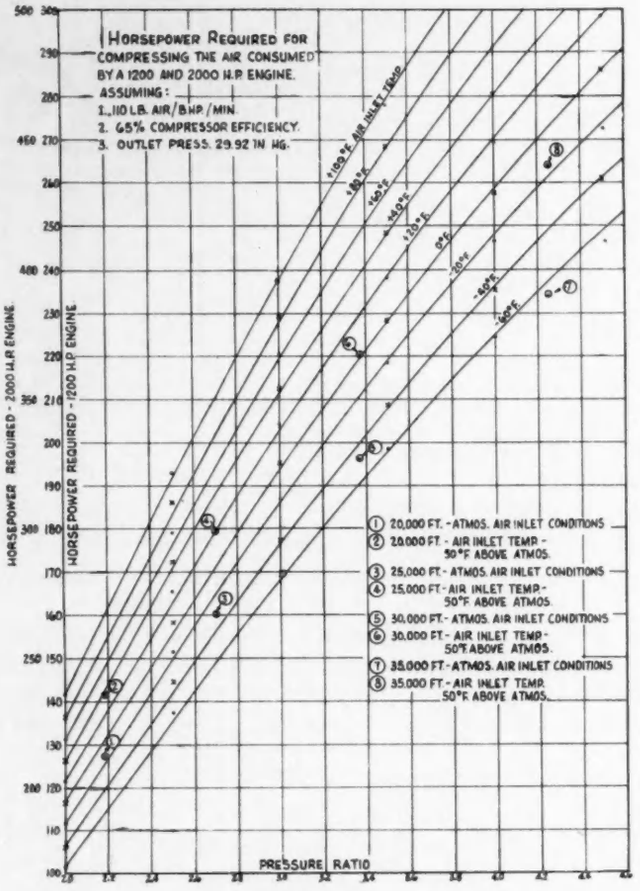


Fig. 4 – Horsepower Required for Compressing the Air Consumed by a 1200- and 2000-Hp. Engine

increased 50 deg. fahr. in temperature, then the power required increases to 220 and 366 hp., an increase of 24 hp. in the one case and 40 hp. in the other. This temperature rise may result from:

1. The air inlet duct being heated by the engine compartment.
2. Exhaust gas entering the air intake.
3. Air entering the air intake which was heated by passing through the intercooler as cooling air.
4. Proximity of the air intake duct to the exhaust collection system, or any combination of these causes.

This increase in power by inlet-air temperature does not include the effect on intercooling requirements, detonation tendency, and possible reduction in volumetric efficiency of the engine.

From this same figure, the influence on horsepower required for supercharging for other altitudes, inlet and outlet pressures, and inlet temperatures may be obtained.

Intercooling

One of the features that makes the exhaust-turbine-driven supercharger particularly attractive is the opportunity for cooling the air before adding the fuel in order to provide a lower temperature of the fuel-air charge to the cylinder.

In designing a supercharger intercooler, not only must the device be an efficient heat exchanger but its pressure drop should be as low as possible. If a high pressure drop is experienced through the intercooler there is an accumulated power waste with the following disadvantageous results:

1. The exhaust back pressure imposed on the engine is needlessly high for a given carburetor inlet pressure.
2. The supercharger rotates at an unduly high speed to overcome the pressure drop through the intercooler.
3. The supercharger pressure ratio, and consequently the final temperature of the air delivered, is unnecessarily high which throws an additional burden on the heat-exchanging requirement of the intercooler.
4. The combination of the high back pressure and high temperature is conducive to engine detonation.

To determine the effect of the pressure loss through the intercooler on the temperature rise through the supercharger, Fig. 5 was computed. These curves are based on a supercharger delivery pressure of 29.92 in. hg., standard altitude pressure and temperatures at the supercharger inlet, and a 65 per cent supercharger efficiency. At 25,000 ft. an intercooler having a pressure drop of 1.25 in. hg. would result in a supercharger temperature rise of 233 deg. fahr., whereas 263 deg. fahr. would be obtained from an intercooler having a pressure drop of 5 in. hg. Although this difference in itself does not seem much, it has a cumulative effect on final supercharger-engine performance as noted previously.

Air-Intake Ducts

The effect of heating the air which is entering the supercharger by 50 deg. fahr. on the power required for supercharging has been considered in Fig. 4, and Fig. 6 has been computed to show the progressive increase in the supercharger air outlet temperature that will result from a constant increase in inlet air temperature, especially at altitude. Like previous curves, assumptions have been made that the pressure delivered at the supercharger outlet is 29.92 in. hg. absolute, the inlet pressure is in accordance with the standard atmosphere, and the supercharger efficiency is 65 per cent. If the inlet air temperature is standard atmospheric, an altitude of 8500 ft. is attained before the outlet temperature exceeds 100 deg. fahr. If the inlet air temperature is raised, by any means whatsoever, by only 20 deg. fahr., the altitude is reduced to 4000 ft. In other words, a 20-deg. fahr. rise in inlet

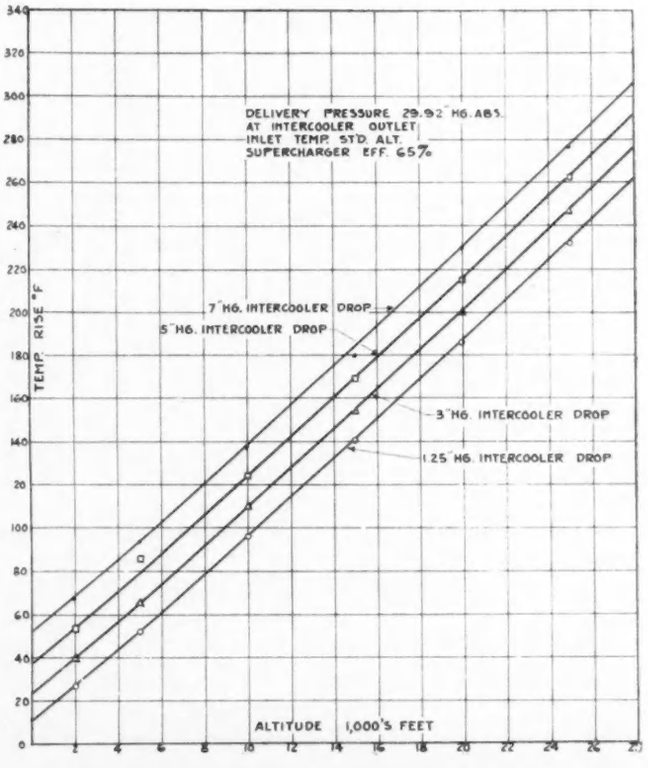


Fig. 5 – Effect of Pressure Loss Through Intercooler on Temperature Rise Through Supercharger

air temperature reduces the altitude at which intercooling is required by 4500 ft.

Likewise, with inlet air at standard atmospheric temperature, an outlet air temperature of 180 deg. fahr. will be reached at 23,000 ft. but, with an increase of 20 deg. fahr. inlet air temperature the same outlet temperature (180 deg. fahr.) will be reached at 18,000 ft., a reduction of 5000 ft. Therefore, at the higher altitude where the intercooler already is loaded heavily the influence of high inlet temperature is even more detrimental. It will be noted that, at the higher temperatures of 80 deg. fahr. and 100 deg. fahr. above atmospheric temperature, the altitude is not influenced so much by a 20 deg. fahr. change in temperature, but inlet temperatures of this magnitude result in unreasonably high outlet temperatures and the intercooler would be so large, to prevent loss in engine power and possible detonation and preignition, that the air duct should be redesigned.

In making a supercharger-engine installation in an airplane, the tendency is toward the smallest possible air inlet duct to the supercharger for the purpose of reducing airplane drag. With a given airplane, the air pressure at the supercharger is entirely a function of air-duct design. Fig. 7 shows the importance of a high inlet air pressure on the temperature rise through the compressor, and this value is, of course, an index of the power consumed in compression. The solid lines indicate adiabatic compression, and the dotted lines are for a compressor efficiency of 65 per cent. With a normal atmospheric inlet pressure the temperature rise at 20,000 ft. is 173 deg. fahr., if this pressure is reduced 2 in. hg. by duct losses, the temperature rise is increased 42 deg. fahr. to an outlet temperature of 215 deg. fahr. This increase represents additional heat to be extracted by the intercooler. With a ram of 2 in. hg. above atmospheric pressure, the temperature rise would be decreased 34 deg. fahr. below normal to an outlet

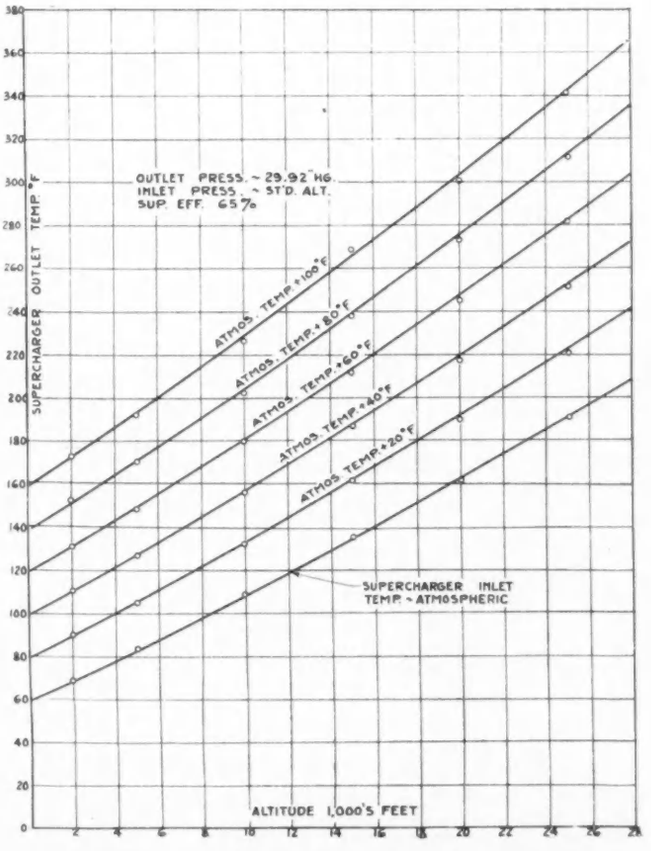


Fig. 6 – Effect of Supercharger Inlet Temperature on Supercharger Outlet Temperature

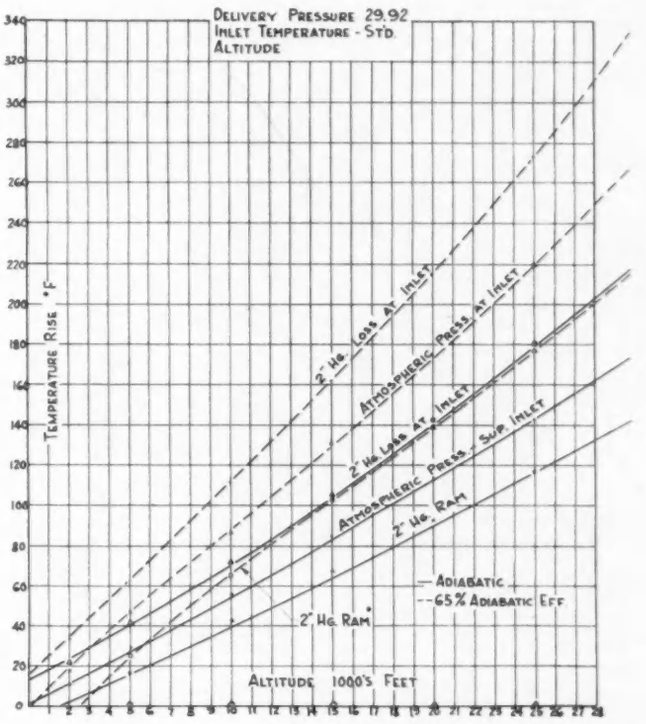


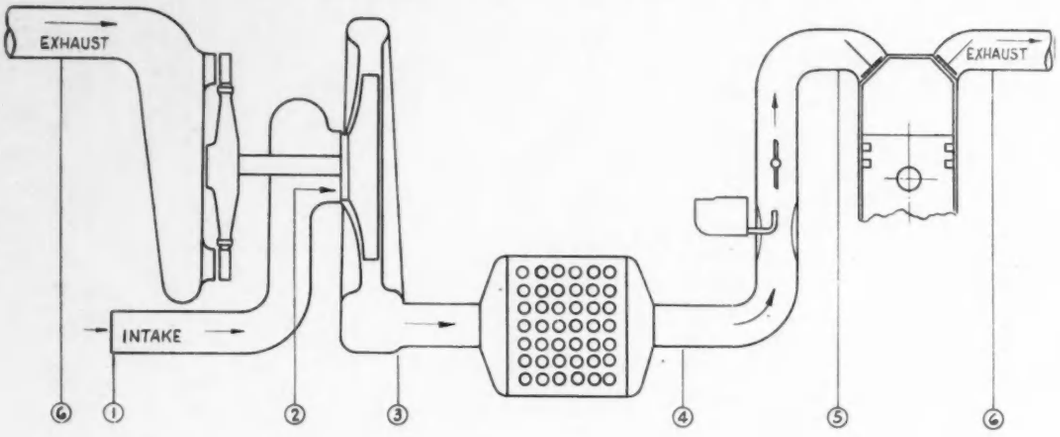
Fig. 7 – Effect of Pressure at Supercharger Inlet on Temperature Rise Through Supercharger

temperature of 139 deg. fahr. This is a total range of 76 deg. fahr. It is realized that, in plotting these curves at constant pressures above atmospheric, at various altitudes a change in airplane velocity proportional to the square root of the relative densities is required to maintain these fixed values. It has been observed that one of the most important items of good supercharger-engine performance is to design the air-inlet ducts in such a manner that the absolute air pressure at the supercharger inlet is a maximum and the air temperature at the same location is a minimum. The importance of this item increases with the critical altitude of the installation and cannot receive too much attention in airplanes designed specifically for high-altitude operation.

Effect of Installation on Intercooling

Since a survey has been made of the influence of several separate installation factors on the air temperature out of the compressor, a group of curves, Figs. 8, 9, and 10, have been prepared to indicate the combined effect of these factors. These curves trace the air consumed by the engine from atmosphere through the compressor and intercooler to the point at which it enters the carburetor. The upper group of curves, *A* and *B*, represent the pressures at the various points in the system, *A* corresponding to a poor installation as defined on each figure and *B* to a good installation. A temperature of 100 deg. fahr. at the carburetor inlet has been selected as a desired temperature for good engine operation. Curves have been plotted corresponding to the good and poor installation with atmospheric temperatures at the inlet, (2) and also with increased inlet temperatures shown on the figures.

Considering the 10,000-ft. curves, Fig. 8, Curve *C* indicates that, with a good installation and atmospheric air temperature, no intercooler would be required since the supercharger outlet temperature is less than 100 deg. fahr. However, a poor installation would result in an intercooler burden corresponding to triangle *XYM* to reduce the outlet temperature from 180 deg. fahr. to 100 deg. fahr.



Figs. 8*A*, 8, 9 and 10 – Effect of Installation on Intercooling Requirements

Fig. 8*A* – Diagram of Installation

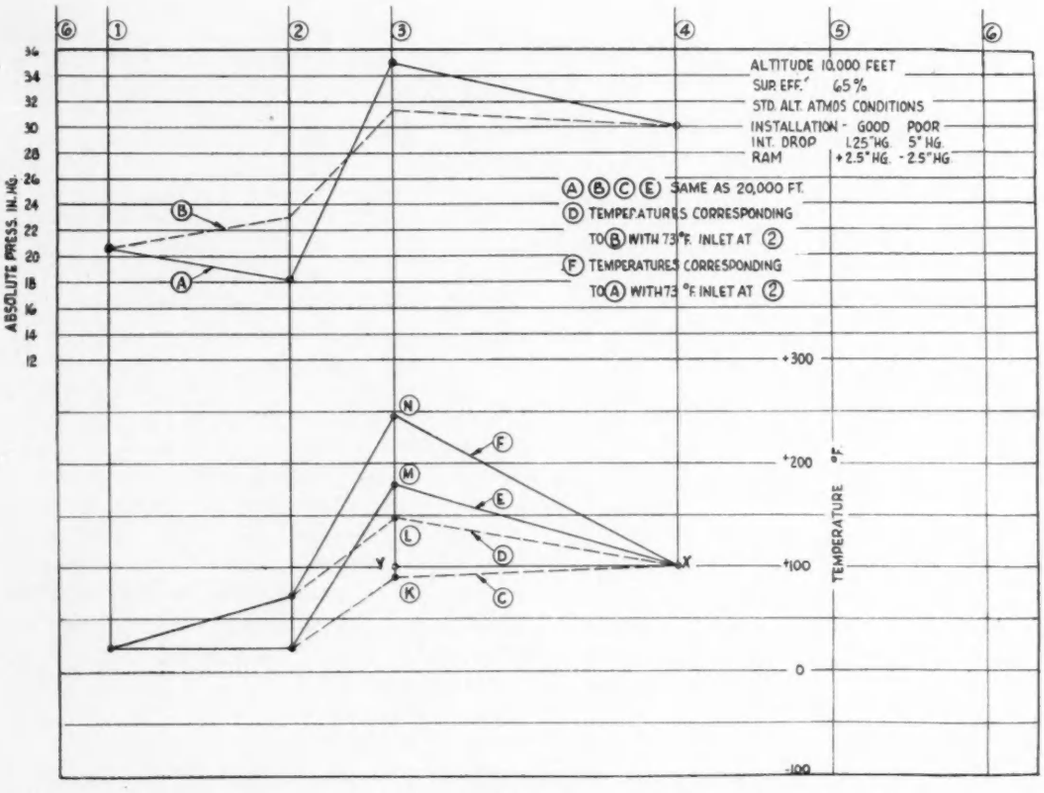


Fig. 8 - Data at 10,000 Ft.

If the air temperature at (2) were to be increased by any means whatsoever from the normal value of 23 deg. fahr. to 73 deg. fahr., then the good installation would require the extraction of heat by the intercooler corresponding to *XYL* and the poor installation *XYN* or a reduction in temperature from 245 to 100 deg. fahr. A comparison of these triangles emphasizes the importance of designing the air ducts for maximum ram and minimum inlet air temperature and the intercooler for minimum pressure loss.

Fig. 9 shows the same type of data as shown by Fig. 8 except for the increase to an altitude of 20,000 ft. However, it will be noted that the triangles have increased in area with *XYG* showing the good installation with atmospheric air temperature. However, if a poor installation is made, then triangle *XYI* results which means the intercooler must cool the air from 242 deg. fahr. to 100 deg. fahr. before entrance to the carburetor. If, however, the inlet air temperature is increased from the normal —12 deg. fahr. to 50 deg. fahr., the triangle *XYH* illustrates that it is almost as bad to raise the temperature at the supercharger inlet by 62 deg. fahr. as to make a poor installation of the air intake duct of the supercharger. If the poor installation and increased air temperature are combined, then triangle *XYJ* results which means that the intercooler must provide for a reduction in the air temperature from 342 deg. fahr. to 100 deg. fahr.

An examination of Fig. 10, which is computed for an altitude of 30,000 ft., indicates that, with the very best installation, an intercooler which will remove heat equivalent to triangle *XYO* is necessary. In other words, the temperature of the air out of the supercharger must be reduced from 200 to 100 deg. fahr. If a poor installation is made and the air temperature is raised from the normal —48 deg. fahr. to 0 deg. fahr., then triangle *XYR* results which will require a reduction in temperature from 380 deg. fahr. to 100 deg. fahr.

Although it is true that the size of triangles does not represent the required cooling surface of the intercooler since the temperature difference increases with the outlet temperature, the triangles do represent the relative amounts of heat to be extracted from the air to provide a desirable carburetor-air temperature for engine consumption.

Figs. 8A, 8, 9 and 10 – Effect of Installation on Intercooling Requirements

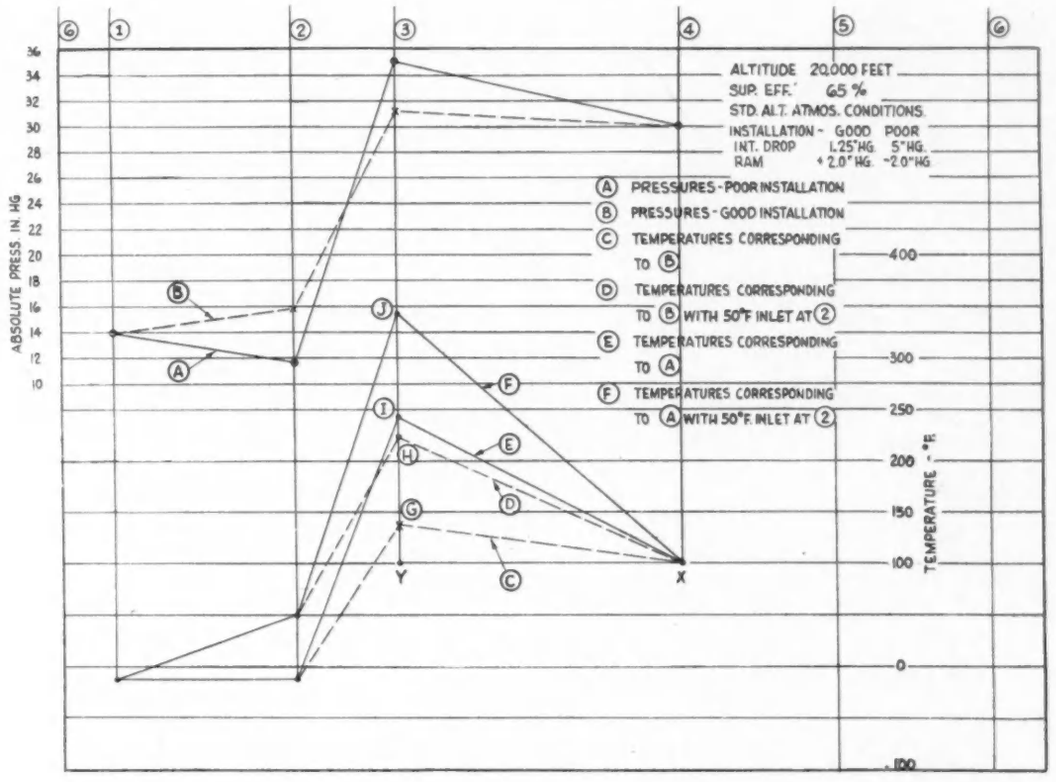


Fig. 9 – Data at 20,000 Ft.

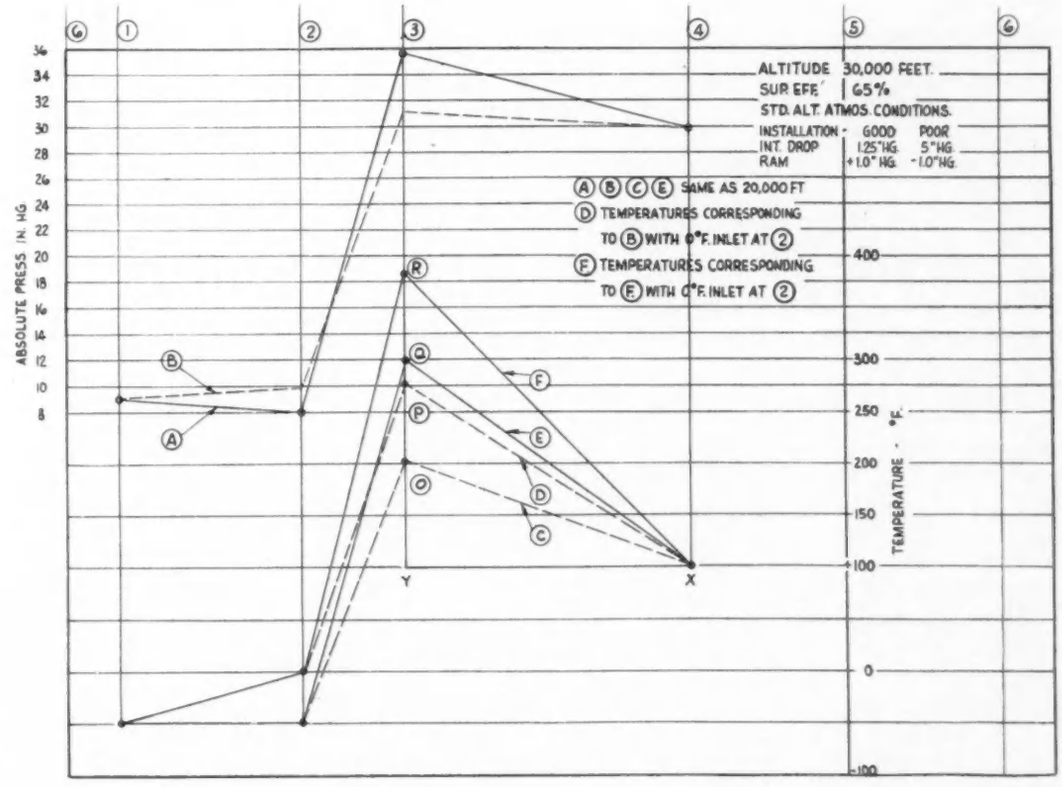


Fig. 10 – Data at 30,000 Ft.

Installation Efficiency

It is possible to evaluate directly from flight test the combined supercharger efficiency, the adequacy of the inlet air duct, and the effect of undesirable heating, by computing in each case what is termed the Installation Efficiency. As designated here it does not include the pressure loss through the intercooler. This factor was not included because some pressure drop through the intercooler is inevitable and it is possible to check this particular factor by simple laboratory flow tests. This checking should always be done and will therefore insure a low pressure drop when installed in the airplane.

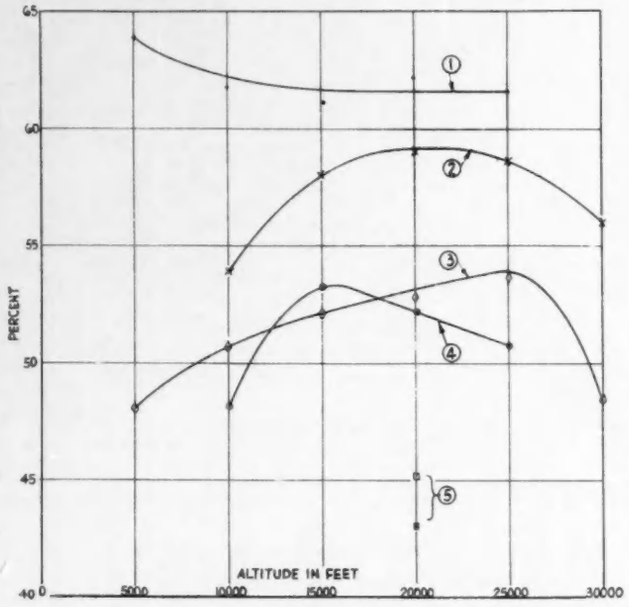


Fig. 11 – Installation and Supercharger Efficiency for Various Airplane Installations

The "Installation Efficiency" as defined by the authors is:

$$\frac{\left[\left(\frac{P_s}{P_a}\right)^{0.291}-1\right]\left(t_a+460\right)}{t_s-t_a}$$ where

P_s = Absolute pressure out of the compressor.
P_a = Absolute pressure of the air in which the airplane is flying.
t_a = Temperature of the atmospheric air, deg. fahr.
t_s = Temperature of the air leaving the compressor, deg. fahr.

The numerator of this formula represents the adiabatic temperature rise as a result of compressing the air from static atmospheric pressure and temperature to the pressure at the supercharger exit. The denominator represents the actual temperature rise from atmospheric temperature to the temperature at the supercharger exit. Therefore, with a good ramming air intake, the actual temperature rise in the denominator would correspond to a lesser pressure ratio than indicated in the numerator. However, this is as it should be since the installation of the supercharger should receive credit for a good air-duct installation. It should be borne in mind that this formula likewise takes into consideration the efficiency of the supercharger itself. Therefore, it is necessary to note the difference between the supercharger efficiency and the installation efficiency. However, the former can be obtained from laboratory tests and the difference in efficiency will indicate how well the installation has been made.

Based on the preceding formula the authors have prepared Fig. 11 showing the installation efficiency of several different installations plotted against altitude. It is obviously impossible to designate the airplanes in which the installations have been made, but this figure does show the wide divergence in efficiency and again emphasizes the necessity for the greatest attention to detail when installing an exhaust-driven supercharger in an airplane. Although it is true that there is some difference in the efficiency of the various superchargers, this difference is not great and would not in any way account for a 43 per cent efficiency at 20,000 ft. for one installation and a 62 per cent for another. It is well known that the installation having the 43 per cent installation efficiency was not good, and a number of changes have been made with substantial improvement. It is unfortunate that complete flight-test data are not available so that a comparison of the efficiency before and after modification could be made.

Curve (1) of this figure indicates a substantially constant efficiency from 10,000 to 25,000 ft. with only a slight increase at 5,000 ft. which is not explainable.

An examination of Curves (2), (3), and (4) indicates lower efficiencies at 10,000 ft. than at 15,000 ft. It is believed that this characteristic was caused in each case by heating of the air prior to entrance to the supercharger. It is the belief of the authors that aircraft manufacturers should make a check on the installation efficiency of their supercharger installations on preliminary flight tests. The intercooler should be designed so the pressure drop is a minimum and the temperature of the air being delivered to the carburetor is reduced to 100 deg. fahr.

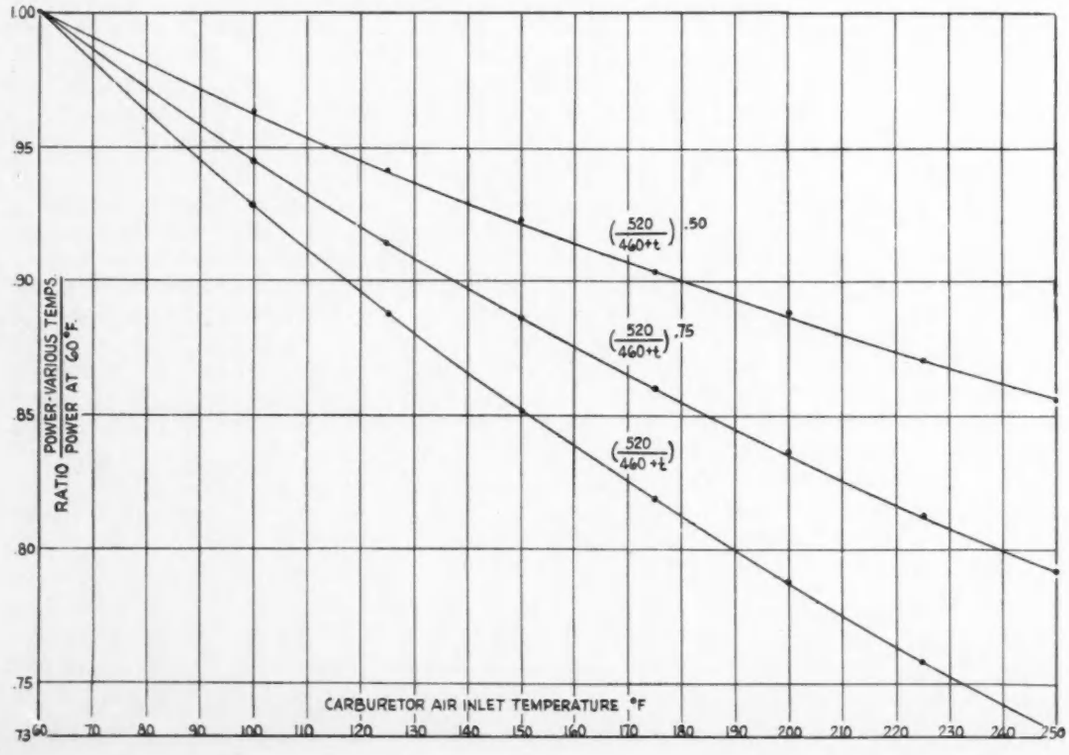


Fig. 12 – Effect of Carburetor Air Inlet Temperatures on Power Output with Various Correction Formulas (Assuming No Detonation)

Effect of Carburetor Air Temperature

If the air being delivered to the carburetor of an engine is at a temperature higher than standard, then there are two major effects which must be taken into consideration when determining the power output of the engine. First, there is a drop in power as a result of reduced charge weight which is considered in the normal horsepower correction factor. The second effect is the tendency toward detonation and preignition of the engine resulting from the high inlet-air temperature. The second factor is dependent upon the fuel used and the individual characteristics of the particular engine, but the first factor is susceptible to correction. Fig. 12 has been prepared showing the effect of carburetor air-inlet temperature on power output with various correction formulas. The common correction is inversely as the square root of the absolute temperature, but there is some evidence that a greater correction factor is obtained on some types of engines wherein the exponent varies from 0.5 to inverse proportion. With any correction factor there is a loss in engine power if the air temperature is not reduced to the 100 deg. fahr. shown by the triangles in Figs. 9, 10 and 11. Of course, however, lower than 100 deg. fahr. air temperatures would result in more power, but possibly at the expense of carburetor icing or poor fuel-air mixture distribution. Considering Fig. 10, and the square-root correction, if the poor installation with the high inlet air temperature is cooled from 340 deg. fahr. to only 250 deg. fahr. by an inadequate intercooler then, as shown by Fig. 12, the power at the 250 deg. fahr. point compared to the 100 deg. fahr. desired point would be in the ratio of $\frac{0.856}{0.962}$ or a reduction of 11 per cent, neglecting possible detonation or preignition. If the engine correction factor more nearly approaches the 0.75 power of the inverse absolute temperature then under the same circumstances, the ratio would be $\frac{0.792}{0.945}$ or a reduction in power of 16.2 per cent. If the inverse ratio were found to be applicable to a particular engine, then the power reductions with excessive carburetor air temperature would be still greater. It might be argued that, even though the intercooler is inadequate and results in a high carburetor air temperature, this power loss can be overcome by an increase in manifold pressure. Within limits, this argument is true but, at the higher critical altitudes with the corresponding increase in carburetor air temperature, detonation and preignition limit the extent to which this compensation may be applied.

Effect of Reduced Back Pressure on Power

In presenting these data, the authors realize that it is not complete or particularly conclusive evidence on whether or not the increase in power resulting from reduced back pressure is sufficient to drive a mechanical supercharger. However, it is being presented as a stimulant to discussion and to aid in the solution of this problem. Curves (1) and (3) of Fig. 13 show the horsepower increase with reduced back pressure for two typical engines, one a nine-cylinder and the other a fourteen-cylinder, radial air-cooled engine. These curves were obtained by maintaining a constant carburetor inlet pressure corresponding to the critical altitude of the engine with the throttle wide open and the carburetor air temperature as shown on the curves. The critical altitude for one engine under these conditions was 8000 ft., Curve (3), and the other was 3200 ft., Curve (2). Then, reducing the absolute exhaust pressure only by connecting the exhaust to vacuum pumps and maintaining constant engine speed, the horsepower increase was obtained. These curves represent, therefore, the power gain with altitude which would be ob-

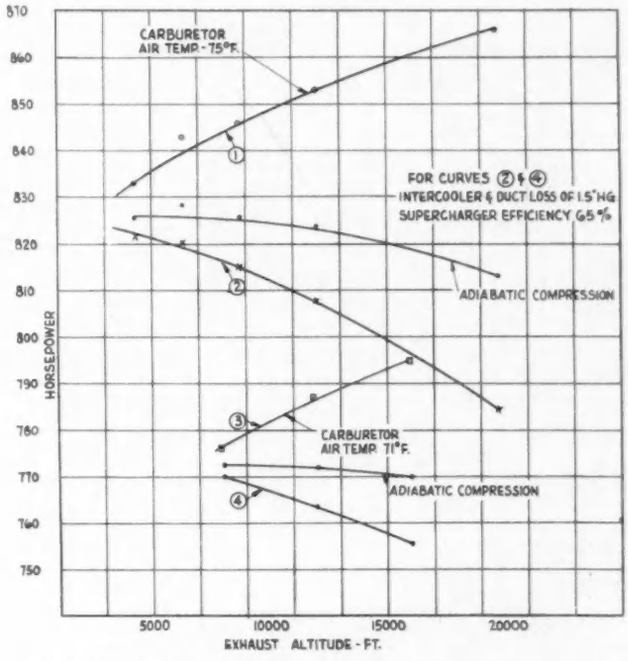


Fig. 13 – Preliminary Data on Effect of Reducing Exhaust Back Pressure on Power and Computed Net Power at Various Altitudes

tained if an auxiliary, separate-driven supercharger were used to maintain a constant carburetor inlet pressure and an intercooler interposed to maintain a constant carburetor inlet temperature. If, then, the horsepower required to compress the quantity of air represented by air-flow measurements during these tests from atmospheric pressure corresponding to the exhaust back pressure to critical altitude pressure is computed, an idea can be obtained as to what would happen to the horsepower of the engine with increase in altitude. For this computation it was assumed that the theoretically applied supercharger would be gear-driven from the crankshaft, that the intercooler and the connecting ducts would cause a pressure loss of 1½ in. hg., and the supercharger efficiency would be 65 per cent. The theoretical supercharger also would be one having exactly the proper impeller speed for each altitude with no power loss as a result of the method of pressure control. Based on these computations, Curves (2) and (4) were prepared deducting from Curves (1) and (3) the horsepower required to drive this theoretical supercharger. It will be noted that, in neither case, do the experimental and computed curves intersect constant horsepower at the two critical altitudes since some power is required to force the air through the intercooler and ducts. Curves (2) and (4) indicate, therefore, that there is not enough power gained by reduction in exhaust back pressure, with increased altitude, to supply the power needed to drive this theoretical supercharger. To maintain a constant horsepower it would, therefore, be necessary to increase gradually the manifold pressure as the altitude is increased. Two curves have been added, one between Curves (1) and (2) and the other between Curves (3) and (4) to show the engine horsepower if the supercharger compression were adiabatic. These curves indicate that, the higher the supercharger efficiency, the nearer the engine horsepower is constant with increase in altitude, and emphasize the necessity of good compressor efficiency.

Exhaust Back Pressure

To obtain some idea of the effect of exhaust back pressure on engine performance, runs were made on two engines, one a fourteen-cylinder, two-row radial air-cooled type with geared

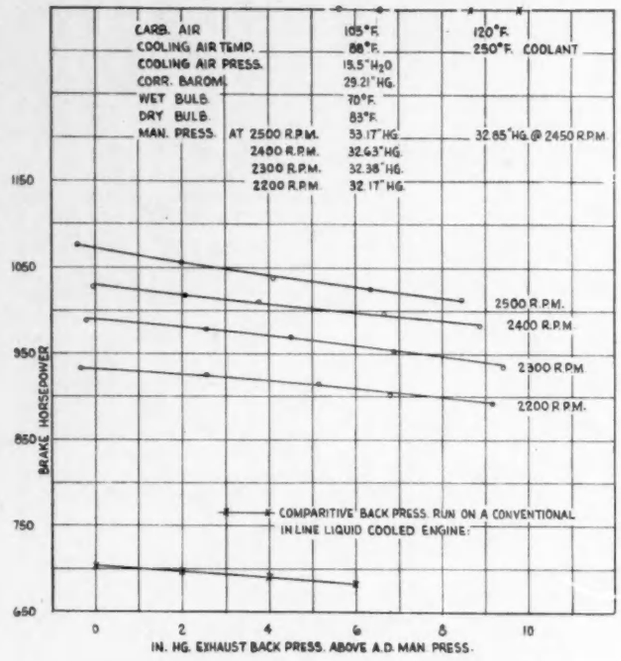


Fig. 14 – Exhaust-Back-Pressure Runs at Best Power at Various Speeds on a Twin-Row Radial Air-Cooled Engine

supercharger and the other a liquid-cooled V-type, as shown on Fig. 14. The air-cooled engine was supplied with 100-octane fuel and the liquid-cooled engine used 92-octane gasoline. Both were run at full throttle, and an exhaust collector was installed to increase the exhaust pressure above atmospheric. The liquid-cooled engine was supercharged to a manifold pressure of 32.85 in. hg. absolute, by an external compressor. The data were plotted as exhaust pressure in excess of manifold pressure rather than above atmospheric pressure since it is the difference between intake and exhaust pressure that influences cylinder performance. The other conditions of operation are shown on Fig. 14. These curves indicate that there is no abrupt change in horsepower with increased exhaust back pressure up to 9 in. hg. above intake-manifold pressure. It is noted that the spacing between the air-cooled engine curves progressively decreases as the speed increases, but this condition may be accounted for by the fact that the manifold pressure does not increase rapidly enough to hold constant power increments for equal increments of speed. The slope of the 2500 r.p.m. curve is more abrupt than that obtained at the lower speeds, which may mean that the output was such that the engine was near detonation and the back pressure caused a more rapid drop in power. To correlate these data, Fig. 15 was prepared, which shows the percentage decrease in horsepower with exhaust back pressure above intake manifold pressure. A mean line drawn through these curves would indicate a 0.55 per cent drop in horsepower for each inch of mercury that the exhaust back pressure exceeds the manifold pressure. This curve checks fairly well for both engines, and the fact that there is good agreement would indicate that no unusual combustion characteristics were obtained in either engine which caused a rapid drop in horsepower with increase in exhaust back pressure. It is difficult to measure exhaust back pressure accurately. The static measurement at a particular cylinder usually gives too low a reading because of the fluctuating pressure and the authors suggest measurement by an impact tube at a point in the collector carrying the exhaust from all cylinders. This procedure was used for these tests and gave satisfactory results. Whatever point in the exhaust system is selected for pressure measurements during the dynamometer check of an engine, the same point should be used during the flight tests. This method will assure the proper correction of engine power for exhaust back pressure.

Exhaust Systems

The chief problems in designing an exhaust system for an exhaust-driven supercharger are:

1. Provide sufficient area throughout the system to avoid excessive pressure loss.

2. Supply adequate flexible connections to prevent undue thermal and vibration stresses, without leakage.

3. Provide enough cooling to prevent failure of the stationary and moving parts of the supercharger which come in contact with the exhaust gases.

Considering the first requirement, sufficient data are available on standard exhaust-disposal systems to design a collector

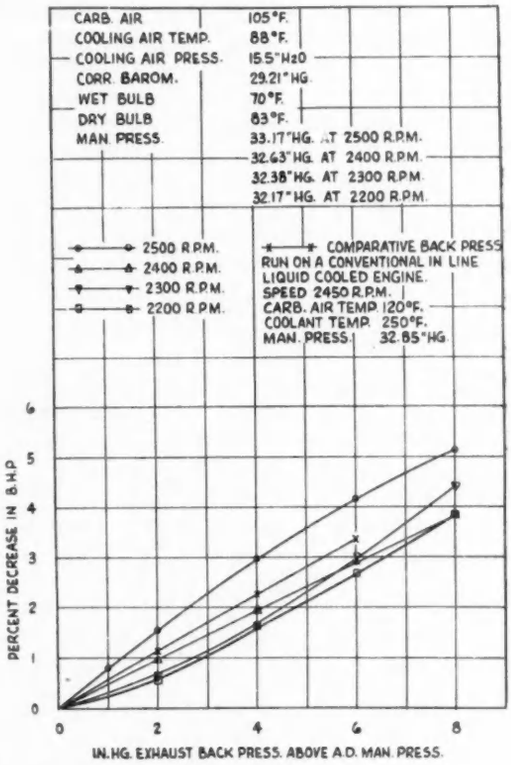


Fig. 15 – Exhaust-Back-Pressure Runs at Best Power at Various Speeds on a Twin-Row Radial Air-Cooled Engine

without excessive pressure loss. As indicated previously, the effect of exhaust back pressure on engine power is not as serious as other factors, for example, supercharger inlet air duct and intercooler design. It should be remembered however, that 0.55 per cent of power is lost with each inch of mercury increase in exhaust back pressure.

The design of ample flexible connections has been, and is, receiving much attention by all interested in engine installations. These connections are absolutely essential to permit the expansion and contraction incident to high temperatures and sudden cooling. Also, the relative vibration between the engine and supercharger must be considered if failures are to be avoided. Exhaust-system leakage results in the reduction in critical altitude of the engine-supercharger combination, and no change in performance is detected at altitudes less than the critical altitude. This type of leakage is not as objectionable as a leak in the induction system between the supercharger compressor and the carburetor, which

greatly overloads the supercharger and intercooler and imposes high exhaust back pressure on the engine. This condition is caused by the automatic regulator attempting to maintain constant carburetor inlet pressure by closing the waste gate, since the regulator cannot distinguish between air being consumed by the engine and that leaking into the atmosphere

The degree to which the stationary and rotating parts of the supercharger which come in contact with the exhaust gases must be cooled is dependent upon the design of these parts and the strength at high temperature of the materials used.

Fig. 16 shows that, with a low-restriction exhaust-disposal system, efficient supercharger, and otherwise good installation, the exhaust pressure gradually decreases with altitude up to 20,000 ft. Above 4000 ft. the exhaust back pressure is less than the supercharger outlet pressure; above 8000 ft., it is less than the cooler outlet pressure; and, above 14,000 ft., is less than the carburetor inlet pressure. This figure shows that, at 20,000 to 25,000 ft., the exhaust pressure is approximately 0.6 in. hg. less than the carburetor inlet pressure and approximately 1.4 in. hg. less than the intake manifold pressure. The intake manifold pressure is in excess of the carburetor inlet pressure for this particular engine because of a low-ratio integral-geared blower.

Take-Off Power

Should an aircraft engine to which an exhaust-driven supercharger is to be applied be built with its normal sea-level rating only and obtain take-off power by means of an exhaust-turbine-driven supercharger, or should the engine be capable of developing its full take-off horsepower independent of the exhaust-turbine-driven supercharger using this supercharger to maintain altitude horsepower only? This is a question frequently asked by the engine and airplane designer.

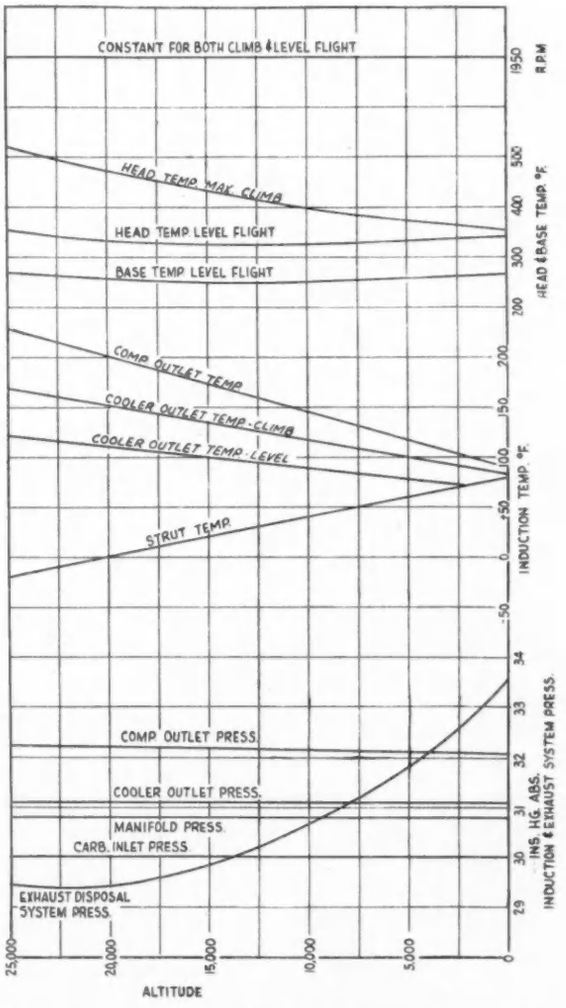


Fig. 16 – Flight Test of Exhaust-Turbine-Driven Supercharger Showing Effect of Properly Designed Exhaust-Disposal System

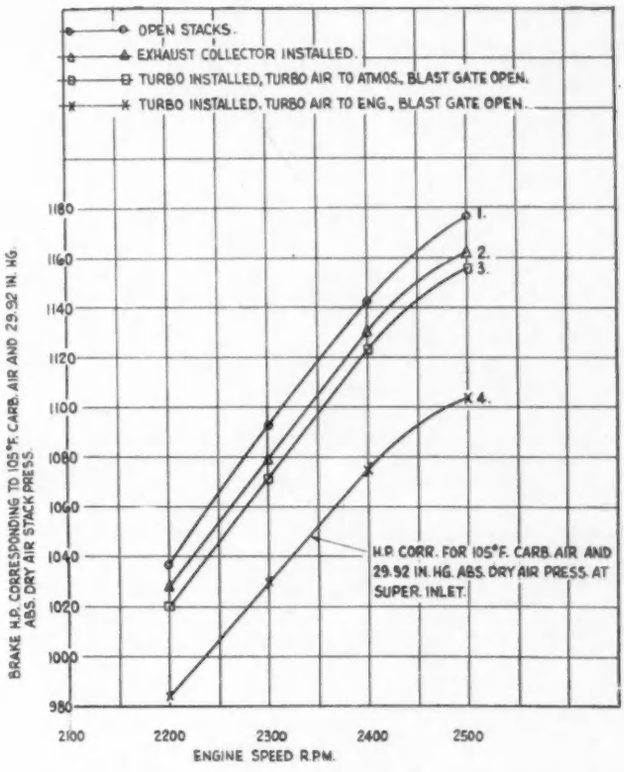


Fig. 17 – Full-Throttle Curves at Best Power – Twin-Row Air-Cooled Engine

To furnish some information on this problem the authors submit dynamometer data on two typical twin-row, radial air-cooled engines obtaining their take-off power by the two methods just designated.

The first engine, engine *A*, had a blower gear ratio which resulted in an impeller tip velocity of 907 ft. per sec. at take-off speed and developed 1400 hp. at full throttle. The second engine, engine *B*, had an impeller tip velocity of 540 ft. per sec. at the same speed and developed the power shown by Curve 1, Fig. 17. On engine *B* an exhaust collector was installed which resulted in a decrease of 12 hp. at 2400 r.p.m. as shown by Curves 1 and 2 of Fig. 17. An exhaust supercharger was then installed and, with the compressor discharging to the atmosphere and with the waste gate open, there was an additional drop of 7 hp. When the exhaust supercharger was fully connected to the engine but with the waste gate open the power dropped 48 hp. at 2400 r.p.m., Curve 4, because of the restriction in the induction system.

With the supercharger installed on engine *B*, data were taken at 1100, 1200, 1300, and 1400 hp. at constant engine speed as shown on Figs. 18, 19, and 20. These curves are cross-plots from "mixture-control" runs at each power and are for a constant fuel-air ratio of 0.094 and with a constant carburetor air temperature of 105 deg. fahr.

Curve 1 of Fig. 18 shows the expected increase in manifold pressure at the higher power outputs. A comparison of Curves 1 and 3 indicates the rise in pressure obtained from the integral, geared blower within the engine, which amounted to 6.9 in. hg. at 1300 hp. A comparison of Curves 2 and 3 indicates the loss in pressure from the exhaust super-

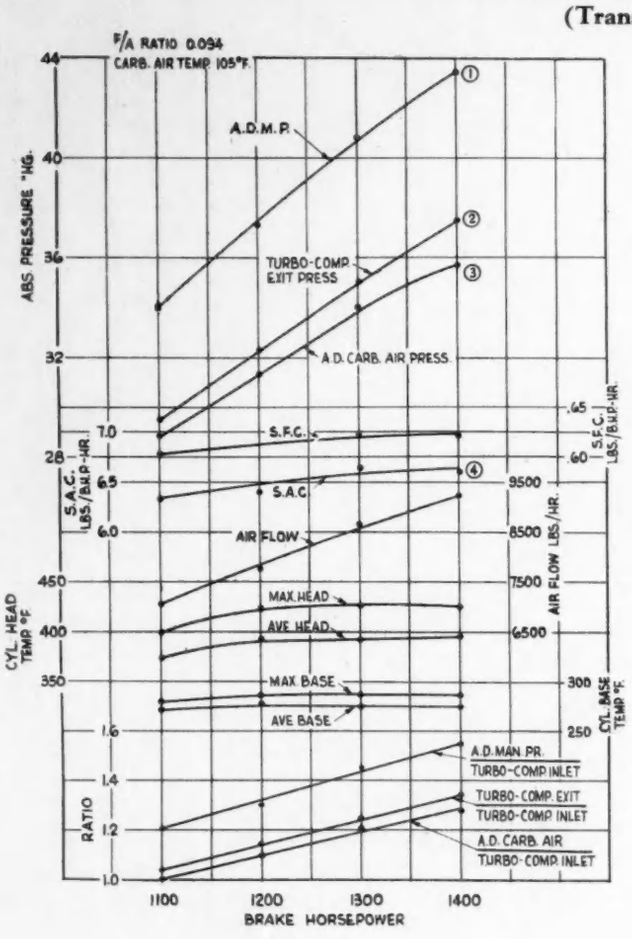


Fig. 18

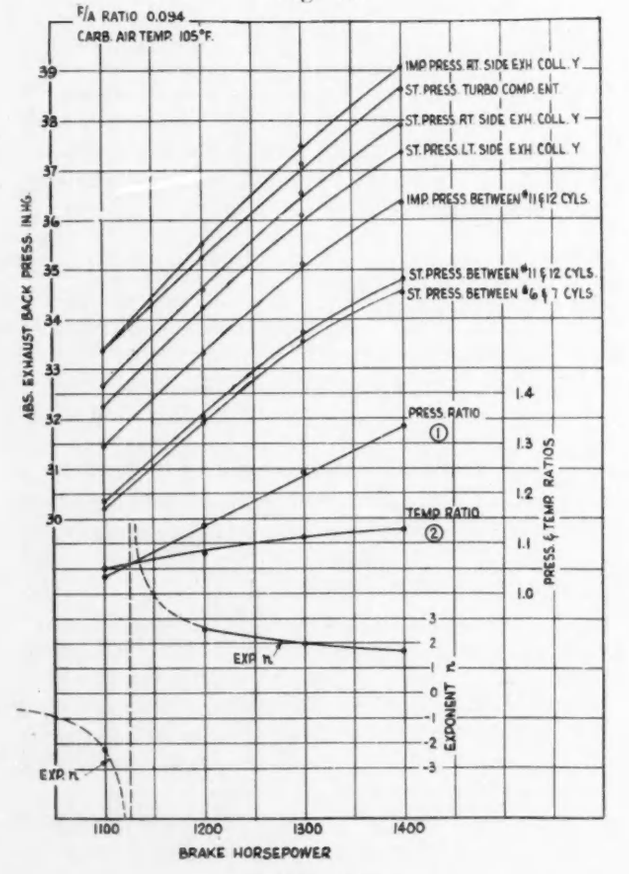


Fig. 19

charger outlet to the carburetor inlet which, at 1300 hp., is 1.2 in. hg., increasing to 1.8 in. hg. at 1400 hp.

The specific air consumption, Curve 4, shows a gradual increase from 6.35 to 6.6 lb. per b.hp-hr. at 1400 hp. The cylinder temperatures did not rise appreciably at outputs in excess of 1200 hp. However, the cooling air temperature at 1400 hp. was 20 deg. fahr. less than the other runs. The lower curves show the pressure ratios from exhaust-driven compressor inlet to carburetor inlet, exhaust-driven compressor outlet and, finally, to intake manifold pressure. The latter curve represents the total pressure ratio of the supercharging system and includes the engine's geared blower.

Fig. 19 shows the variation in exhaust-collector pressure depending on the location of the connection and whether static or impact pressures are measured. It will be noted that, at 1400 hp., the lowest reading is 34.6 in. hg., absolute, taken as a static reading between cylinders 6 and 7, and the highest reading is 39.1 in. hg. taken as an impact reading in the Y after the exhaust has been collected from all cylinders. As stated elsewhere the authors believe the latter reading is more indicative of the true exhaust back pressure. At 1100 hp., as shown on Fig. 19, the exhaust-driven compressor had a temperature ratio in excess of the pressure ratio which accounts for the negative exponent of compression. This condition is caused by the very light load on the supercharger. As the pressure ratio increases for the 1200-, 1300-, and 1400-hp. runs, the exponent gradually decreases as the compressor is loaded to more nearly its normal capacity.

Fig. 20 shows the exponential efficiency and adiabatic temperature-rise ratio, both of which increase as the compressor is loaded to a maximum value of approximately 66 per cent

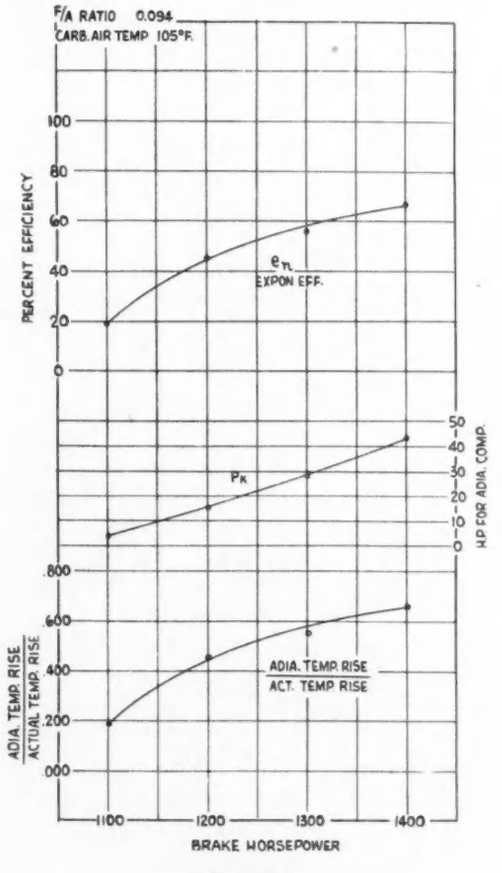


Fig. 20

Figs. 18, 19, and 20 – Full-Throttle Calibration Curves at Constant Speed, Fuel-Air Ratio, and Carburetor-Air Temperature – Twin-Row Radial Air-Cooled Engine with Exhaust-Driven-Turbo Supercharger

at 1400 hp. Too much reliance should not be placed on these quantitative values since they were taken incidental to the engine data. The power for adiabatic compression P_k is also shown for the exhaust-driven compressor which amounts to 43 hp. at 1400 engine hp. With an efficiency of 66 per cent, the actual power being put into the exhaust-driven compressor would be 65 hp. Since "mixture-control" data were available on engine *A* at 1400 hp., a comparison of these data with the "mixture-control" runs on engine *B* furnishes enlightenment on the two methods of obtaining take-off power.

Figs. 21 and 22 show these data under the conditions of operation designated. For engine *B* a carburetor-air temperature of 105 deg. fahr. was selected so that any tendency of the engine to detonate would be aggravated. Engine *A* calibrations are based on an 80 deg. fahr. carburetor-air temperature.

From Fig. 21 it will be noted that, at a fuel flow of 960 lb. per hr., engine *B* obtains its 1400 hp. with an intake manifold pressure of 44.1 in. hg., whereas, engine *A* requires an absolute dry manifold pressure of 47 in. hg., or 2.9 in. more. Of course, the specific fuel consumptions are the same, 0.685 lb. per b.hp-hr. at 960 lb. per hr. fuel flow. An examination of the maximum head temperatures for each of the two engines indicates a temperature of 420 deg. fahr. for engine *B* and 405 deg. fahr. for engine *A*. These temperatures are of questionable value because the two engines were not calibrated on the same test stand. In addition, engine *A* had a 12 deg. fahr. lower air-blast temperature and a 25 deg. fahr. lower carburetor-inlet temperature than engine *B*.

The pressure distribution throughout the induction system for engine *B* is shown on Fig. 21. The supercharger inlet pressure of 28.3 in. hg. absolute is explained by inclusion of air-measuring equipment, which required considerable pressure drop to obtain accurate readings. At a fuel flow of 900 lb. per hr. the exhaust supercharger raised the 28.3 in. hg., absolute, inlet pressure to 37.8 in. hg. exit pressure, or 9.5 in. hg. with an exhaust pressure of 39.2 in. hg. or 1.4 in. hg. in excess of the air pressure at the compressor exit. After obtaining these data, the question then arose as to why the difference in manifold pressure for developing a given horsepower and an examination of the mixture temperatures, shown by Fig. 22, partially answers the question. The mixture temperature at a fuel flow of 960 lb. per hr. for engine *B* is 111 deg. fahr., whereas, the mixture temperature for engine *A* is 145 deg. fahr. It is one of the features of the exhaust-driven supercharger that an intercooler can be interposed between the compressor and the engine and cool the air and finally reduce the mixture temperature. The intercooler for engine *B* cooled the air, at a fuel flow of 960 lb. per hr., from 159 deg. fahr. to 90 deg. fahr. However, on this dynamometer installation, the air was reheated from 90 to 115 deg. fahr. prior to entering the carburetor.

From these data it is concluded that laboratory tests indicate that it is possible to obtain take-off power by the use of an exhaust-driven supercharger applied to an engine having a low blower-gear ratio. Flight tests are required, however, to make certain that an airplane installation can be made which will permit the engine-supercharger combination to function as it did under laboratory conditions.

Acknowledgments

The authors wish to acknowledge their appreciation for the encouragement and suggestions received from Major E. R. Page, Chief of the Power Plant Branch, and for the constructive criticism and comments submitted by Weldon Worth. B. Beaman and J. G. Blackwood ran the engine tests, com-

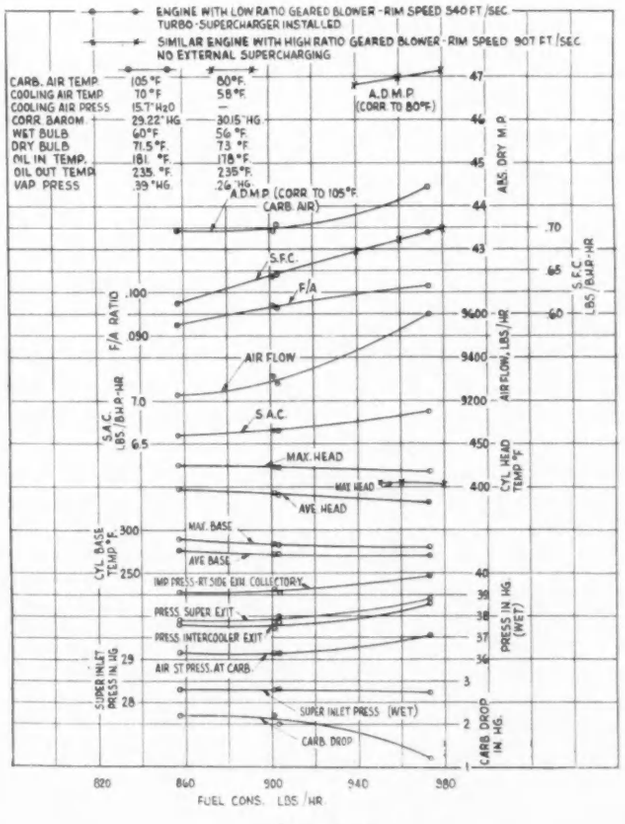


Fig. 21

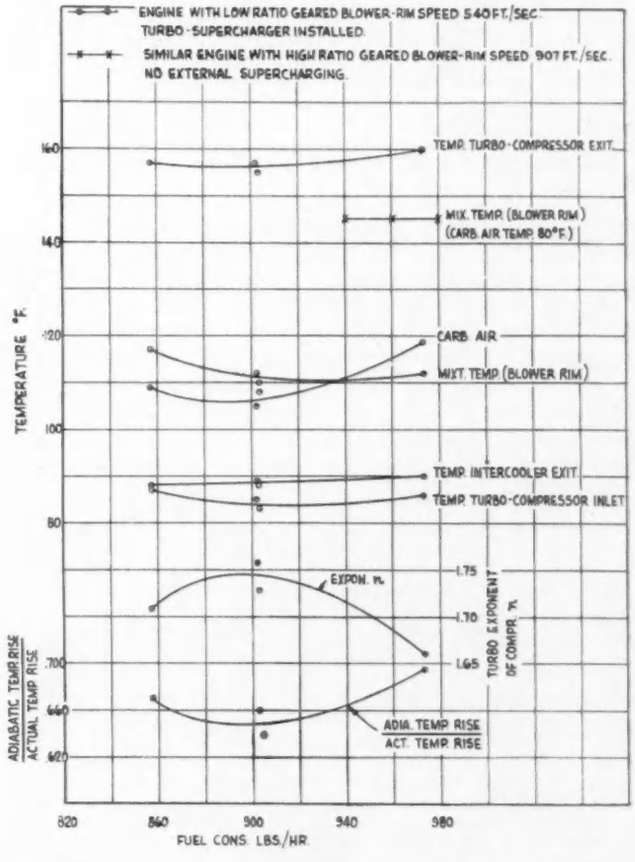


Fig. 22

Figs. 21 and 22 – Constant B.Hp., R.P.M., Mixture-Control Calibration – Twin-Row Radial Air-Cooled Engine

puted the data, and prepared the engine-performance curves and, to them, the authors are particularly indebted because of their interest in the tests and for the large amount of work so satisfactorily accomplished.

Discussion

Stresses Need for Practical Design Data

—Arthur Nutt
Wright Aeronautical Corp.

IT has been the policy of the Army Air Corps for over 20 years to keep all details in connection with the development of the exhaust-turbo supercharger in a secret category and, without question, this fact has contributed to the long delay in the development of this device to a point where it becomes a useful article.

Now we have a paper on this development with some actual engine tests which points out to the designer of installations the penalties which he must pay if he does not make a satisfactory installation. It also points out some of the advantages of the use of the exhaust-turbo installation and leaves the definite impression that, for high-altitude operation, this development is here to stay.

The authors very wisely have avoided mentioning the various practical problems which have to do with the installation of the turbine intercooler and other items which make up the turbo-supercharger installation. It will be necessary for the aircraft and aircraft-engine designers to work closely with the Air Corps on these problems as one of the biggest stumbling blocks to a successful installation has been design of the various units and their proper location in the airplane. It appears to be fatal to attempt to install a turbo supercharger on an installation designed for the conventional engine. The installation should be designed at the time the complete airplane is designed. Otherwise difficulties with airplane balance may result as well as other effects on aerodynamic stability. Considerable information is, no doubt, available and will be made public on this phase of the turbo supercharger development some time in the future, and it is hoped that, in the near future, additional papers will be available from the personnel at Wright Field. Through this means it is hoped that more rapid advance in the development of the turbo supercharger will result. Certainly, whenever it is possible to get more people thinking about a problem, the quicker the development will reach a practical useful stage.

Compares Two Types of Regulation

—S. R. Puffer
Supercharger Engineering Dept., General Electric Co.

THE introduction of the term "installation efficiency" is of particular interest as it provides a simple means of evaluating the performance of an installed supercharger. A possible suggestion might be to designate this efficiency as the "overall installed efficiency." The installation efficiency would then be the overall efficiency divided by the supercharger efficiency. It is conceivable, on such a basis, that with an installation made with proper size ducts without too many bends in the air passages, and with a suitable ramming intake, an installation efficiency of 100 per cent or even greater may be obtained.

Two types of regulation are mentioned, the first in which constant pressure is maintaned immediately preceding the carburetor; and the second in which the pressure preceding the carburetor may be varied to suit engine requirements. The former method has the advantage of being the more simple mechanically as no mixture control is required. It has the disadvantage, however, that, if the engine is operated at part throttle, a higher pressure is held in front of the carburetor than is necessary to maintain partial engine output. This condition results in the supercharger being required to give more compression than is necessary, with the attendant evils so clearly set forth in this paper of increased air temperatures and exhaust back pressures.

The infinite pressure-control system is mechanically complicated by the fact that a reliable mixture control is required, but is otherwise the more preferred of the two. In this system the supercharger does not come into use until the engine throttle is wide open. The supercharger control then acts as a secondary throttle, allowing the supercharger to supply added pressure only to the extent needed to obtain the required power output. This means less compression, lower air temperatures, and lower exhaust back pressures.

The authors have demonstrated clearly the possibility and even the desirability of using the turbo supercharger as a means of obtaining take-off power. In general it can be stated that, when a turbo supercharger is used, as much supercharging should be done before cooling as is possible with as little as possible after the cooling. This statement indicates the use of an engine with a low-gear-ratio built-in supercharger.

If the engine is to be used at high output continuously, this principle of course holds true. On the other hand, if an engine is to be used mostly at cruising power, the use of a smaller and lighter supercharger corresponding in output to the reduced power might be enough of an attraction to warrant a gear ratio in the engine high enough to obtain take-off power without the use of the turbo. In this case, as the engine does not turn at full speed during cruising, the amount of temperature added by the built-in supercharger would be reduced materially, and it is believed that it would not be serious.

In any case, however, it is pointed out that a rather close balance has to be maintained between the capacity of the supercharger and the turbine that drives it if best results are to be secured. For this reason, any installation should be closely coordinated between the builders of the plane, the engine, and the supercharger.

Proposed Method of Test for Ignition Quality of Diesel Fuels

(*Concluded from page 464*)

flywheel are just seen, one at t.d.c. from compression pressure and the other about 10 deg. a.t.d.c. from combustion.

h. Raise upper contact one half division (approximately 0.0005 in.) on adjusting screw where only one flash from combustion will be seen.

13. Rating.

a. The injection shall be reset to the rating position of 13 deg. b.t.d.c., which will make the injection flash appear at t.d.c.

b. The final compression ratio shall be adjusted so as to give coincidence of the injection and combustion neon flashes. The combustion-chamber length shall then be recorded. It is advisable to check combustion contact-point setting by moving injection timing and watching whether or not the combustion and injection flashes move and are still coincident. If so, the setting is correct.

c. From the combustion-chamber length, recorded when the sample fuel has been adjusted to operate at standard conditions with 13-deg. delay period, the cetane number of the unknown sample shall be estimated and two reference-fuel blends selected, differing by not more than eight cetane numbers, which will bracket the unknown sample. When two reference fuels have been found, one of which has a longer delay period and the other a shorter delay period than the sample, alternate readings shall be taken by adjustment of the combustion-chamber length until the neon light flashes coincide at t.d.c. Alternate readings between the sample and reference fuels shall be taken in the following order: Sample, reference fuel No. 1, reference fuel No. 2, sample, reference fuel No. 1, and reference fuel No. 2. After changing from one fuel to another, 5 min. shall be allowed before taking a reading to insure complete flushing of the injection system. The average combustion-chamber lengths obtained from the foregoing series shall be used in calculating the cetane number.

14. Calculation – The cetane number of the fuel shall be calculated by interpolation from relative combustion-chamber lengths as obtained in Section 13 c. The cetane number shall be reported to the nearest whole number. When the calculated value falls exactly half-way between two whole numbers, the nearest even number shall be reported.

Maintenance

15. Maintenance – The following points are important in maintenance technique:

a. Injector should be inspected and cleaned at least every 8 hr. of operation.

b. Compression plug should be cleaned when it becomes tight due to carbon.

c. Special care should be exercised in keeping the fuels and fuel system clean.

d. A general overhaul, including the grinding of valves, cleaning of piston and rings, and cylinder and cylinder-head, should be made every 75 hr. of operation.

e. Periodic inspection of connecting-rod and main bearings is recommended.

Automotive Two-Cycle Diesel Engines

By F. G. Shoemaker

Detroit Diesel Engine Division, General Motors Corp.

THE properly designed two-cycle Diesel is the answer to the problem of building an automotive-type Diesel engine that compares favorably with conventional gasoline engines as to size, weight, and power, Mr. Shoemaker asserts. His paper is chiefly concerned, he announces, with the problem of producing such an engine that will use the same materials, design practices, manufacturing methods, and mechanical parts as are common practice in production-type gasoline engines.

Scavenging is effected in these engines, Mr. Shoemaker explains, by using piston-controlled inlet ports and poppet exhaust valves in the head.

The author shows how a change in design of the blower to a three-lobe helical-rotor type reduced noise and improved discharge characteristics. Piston cooling is effected by coring out the rim of the piston between the piston crown and the ring belt and cooling the crown directly by oil spray, he explains, adding that this cooling reduces the ring-belt temperature well below the coking or gumming point of ordinary oils.

In the remainder of the paper he discusses injection, engine-balance problems and the commercial problems involved in designing the engines and parts to meet a wide variety of applications.

IN undertaking the discussion of the development of automotive-type two-cycle Diesel engines it has been considered unnecessary to repeat any of the historical background which is too well known to require further repetition. In so far as possible the point of view will be that of an engineer confronted with the problem of making a compact light-weight Diesel engine to fit into our own domestic market. Rather than presenting at length details of the engines so developed, the discussion will be primarily of the problems encountered.

Wherever an engine is used, there is always a demand for more power, reduced size and weight, lower fuel consumption, longer life, and less cost. The inherent thermal efficiency of the Diesel engine is an accepted fact, so it is the other factors that constitute the engineering problem.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 14, 1938.]

Present Status of the Automotive Diesel

The Diesel engine for automotive and industrial purposes must compare favorably in size, weight, performance, and cost with existing gasoline engines in order that the user may take advantage of the savings in fuel that are possible. These savings are only realized when the amount of fuel consumed is a considerable proportion of the operating cost, so that, in general, the comparison should be based on the heavy-duty type of engine.

Most of the American automotive-type Diesel engines have been built by gasoline-engine manufacturers following the same design features that have characterized satisfactory gasoline practice. But the maximum gas pressures in Diesel engines are roughly twice those of comparable gasoline engines, so that some increase in the strength and weight of the engine structure has been necessary to maintain the same degree of reliability with the use of conventional materials and manufacturing methods.

Table 1 shows a comparison of the weights of a number of American gasoline and Diesel engines of approximately the same displacement and the same number of cylinders. Since the comparisons also are based upon engines of each type made by the same manufacturer, any differences due to design practice are largely canceled out.

The results of this comparison show that, for engines of 200 to 700 cu. in. displacement, the weight of the four-cycle Diesel engine is about 59 per cent greater on a displacement basis and about 51 per cent greater on a power-output basis than the equivalent gasoline engine. However, the rating of the Diesel engines in b.hp. per cu. in. of displacement is about 6 per cent greater than that of the gasoline engines, showing that the combustion capacity is slightly in favor of the Diesel engine.

This evidence, based on actual commercial engines, shows that the automotive-type Diesel engine is heavier than the same size gasoline engine and, based on the weight of the material alone, must be somewhat more expensive to manufacture. The question naturally is raised: "How can the automotive-type Diesel engine be reduced further in weight per horsepower to compete favorably with existing gasoline engines?"

Limitations of Increased Output

The present four-cycle engines, gasoline and Diesel, are the result of many years of careful work and must be accepted as the best commercial solutions that are now known of the conflicting factors of material cost, manufacturing cost, fuel

Table 1 – Weight and Power Comparison – Four-Cycle American Gasoline and Diesel Engines

Type	Displacement, cu. in.	Rated B.Hp.	Rated Speed, r.p.m.	Total Weight, lb.	Weight, lb. per cu. in.	Weight, lb. per b.hp.	B.Hp. per Cu. In.	Ratio, lb. per cu. in.	Ratio, lb. per b.hp.	Ratio, hp. per cu. in.
D	196	49	2400	900	4.60	18.4	0.250	1.79	1.83	0.99
G	205	52	2600	525	2.56	10.5	0.253	1.00	1.00	1.00
D	274	72	2400	1190	4.34	16.6	0.262	1.31	1.32	1.00
G	260	68	2800	860	3.31	12.6	0.261	1.00	1.00	1.00
D	298	83	2600	1020	3.42	12.3	0.279	1.71	1.59	1.07
G	282	73	2800	565	2.00	7.75	0.259	1.00	1.00	1.00
D	474	120	2000	1780	3.75	14.8	0.253	1.83	1.58	1.16
G	479	104	2200	975	2.04	9.4	0.218	1.00	1.00	1.00
D	468	103	2000	1435	3.07	13.9	0.220	1.51	1.55	0.98
G	468	105	2400	950	2.03	9.0	0.225	1.00	1.00	1.00
D	707	176	1800	2520	3.56	14.3	0.250	1.39	1.17	1.19
G	707	148	2000	1810	2.56	12.2	0.210	1.00	1.00	1.00
				Average Ratio			Diesel	1.59	1.51	1.06
							Gasoline	1.00	1.00	1.00

economy, and durability. The two-cycle Diesel engine must find its place under these same balanced economic influences.

The substitution of lighter metals or improved design detail can be applied equally well to either type of engine so that any gains from this source may be obtained in both gasoline and Diesel engines.

Any increase in rotative speed results in a further attenuation of the engine breathing capacity and a reduction in the power per stroke even though the total power is increased somewhat. This increased breathing restriction, together with the higher friction losses, materially affects the fuel consumption. The greater inertia and centrifugal loads at high speed increase the working loads, and the resultant reduction in engine life is well known. Higher engine speeds then cannot be obtained without some corresponding strengthening of the structure with a resultant increase in weight. It is interesting to note that the rated speeds of the Diesel engines in Table 1 are already about equal to those of the corresponding gasoline engines so that large gains in this direction are not likely.

The comparison also shows that the rated b.m.e.p. of the Diesel engine is now greater than the corresponding gasoline engine, since the average b.hp. per cu. in. is 6 per cent greater and the rated speed is somewhat lower. It is not likely that this figure can be increased by ordinary means sufficiently to equalize the 59 per cent difference in weight. This condition immediately suggests the possibility of supercharging, since the limitations of fuel and detonation, such as affect supercharged gasoline engines, do not exist in the Diesel in which it should be possible to increase greatly the air density with a suitable blower. The successful application of supercharging to many large Diesel engines, where increases in power of 50 per cent are not uncommon, shows the effectiveness of this method of improving engine output. In the case of some very large and heavily constructed engines with a low initial rating this application has been accomplished without changing the design. But in well-developed automotive engines of the type covered in Table 1, such liberties are not permissible without seriously affecting the life and durability of the engine.

In order to increase appreciably the power per stroke by supercharging, it is obvious that the working pressures must be increased, and this increase necessitates a corresponding strengthening of the working parts if materials, design practices, and durability are to be kept at par. This condition is equally true if pressure charging is used to increase both the mean effective pressure and the speed. Assume for the moment that no change in design is required and that the blower and its drive weigh nothing at all. This condition would require an increase of about 50 per cent in the b.m.e.p. of the supercharged four-cycle engine to put it on a par with the unsupercharged commercial gasoline engine as regards weight per horsepower. Whatever increase in engine weight actually is required will raise this figure proportionately. Giving the gasoline engine the same magic materials and the same weightless blower again would upset the comparison so that other means must be found for improving the Diesel engine performance.

This consideration brings up the question: "Why not two-cycle?" It is obvious that, once the cylinder is filled with air, the compression, injection, combustion, and power functions are identical in either four- or two-stroke engines, and that the injection and combustion speed limitations are similar. In a like manner the gas pressures and the working loads are the same per stroke. Hence, if by some means the cylinder can be scavenged of exhaust and filled with air during part of each revolution instead of using a whole revolution for the process, the number of power strokes can be doubled and the output of each cylinder increased 100 per cent with the same working parts as in the four-cycle engine whether supercharged or unsupercharged. Or, for equal power, speed, and uniformity of torque, the two-cycle engine can have half the number of cylinders as the equivalent four-cycle engine. This reasoning immediately presents the possibility of a reduction in the overall length and weight of the engine.

But the scavenging air must be supplied by some sort of an external blower, and this part adds some weight and size to offset partially the reduced number of cylinders. The actual weight of a number of typical displacement-type rotary blowers is shown in Table 2. The data cover a range of sizes from 1500 down to 80 rated engine hp. and show the weight per horsepower based on aluminum housings and rotors, as well as on all-iron construction. The last three blowers in the

Table 2 – General Motors Two-Cycle Diesel Engine Blowers – Rotary-Displacement Type

Blower Output C.F.M.	Blower Output R.P.M.	Rated Engine Output B.Hp.	Rated Engine Output R.P.M.	Blower Weight, lb.	Blower, lb. per b.hp. Aluminum	Blower, lb. per b.hp. Cast Iron
5620	1650	1535	750	1252	0.816	1.257
4770	1404	1200	750	1252	1.043	1.607
3530	1500	900	750	1357	1.509	2.320
943	3880	165	2000	60	0.364	0.568
629.5	3880	110	2000	52.5	0.477	0.702
471.5	3880	82.5	2000	45	0.545	0.755

table are for automotive-type engines and, even when built entirely of iron, they can be made for considerably less than 1 lb. per hp. Hence, speaking broadly, the two-cycle engine of a given output should weigh half as much as a four-cycle engine of the same power, plus about 1 lb. per hp.

This potential weight saving offers an opportunity to make an automotive-type Diesel engine that compares favorably with conventional gasoline engines as to size, weight, and power, and it is with this problem that this paper is concerned chiefly.

Two-Cycle Design Problems

As pointed out previously, the objective of the development has been to produce an automotive-type Diesel engine comparing favorably on a size, weight, and power basis and using the same materials, design practices, manufacturing methods, and mechanical parts as are common practice in production-type gasoline engines.

Two-Cycle Power

Before considering the design problems of multicylinder engines it is first necessary to demonstrate that the two-cycle cylinder actually can produce enough power to justify its adoption. Fig. 1 shows the performance of two typical two-cycle uniflow Diesel engines of conventional design and materials to indicate that this increase in power can be attained and is not limited by cylinder size. The recent work of the National Advisory Committee for Aeronautics on two-cycle Diesel engines shows that an even greater output is possible and that supercharging the two-cycle Diesel engine is feasible.

Having demonstrated that the two-cycle cylinder can produce mean effective pressures comparable to the same four-cycle cylinder with a resultant doubling of the power per cubic inch of displacement, it is of interest to see how the scavenging is effected.

With piston-controlled inlet ports in the cylinder barrel and poppet exhaust valves in the head, the air can be made to traverse the length of the cylinder in such a manner as to scavenge completely the entire cylinder. An excess of air of about 35 per cent to 45 per cent over the displacement has been found to be desirable to insure thorough scavenging and obtain additional cylinder cooling. In Fig. 2 is presented a curve showing the actual excess air by test in an automotive-type two-cycle engine.

The impression seems to be quite general that an excessive amount of power is required to supply the scavenging air, and that the engine speed is limited by breathing to a greater extent than in four-cycle engines. The four-cycle engine breathes through one valve for about 240 deg. of crank rotation, whereas the two-cycle engine breathes through two valves for about 120 deg. The velocity through the valves is therefore the same in each case at any given crankshaft speed and the limiting rotational speed is, in general, the same for both engines. The two-cycle pressure-scavenged engine has the further advantage of having the air supplied by a pressure blower which has a very large inlet passage and is not affected materially by speed, so that the volumetric efficiency is sustained at high speed. This characteristic is shown clearly in Fig. 2.

Fig. 3 shows a typical scavenging pressure vs. speed characteristic curve plotted on logarithmic coordinates. This curve shows that the pressure varies approximately as the square of the speed, and the engine therefore can be considered as a fixed orifice of an appropriate size. In a like manner, the leakage path of the displacement blower has the characteristic of a fixed orifice. Hence the air will flow through the leakage

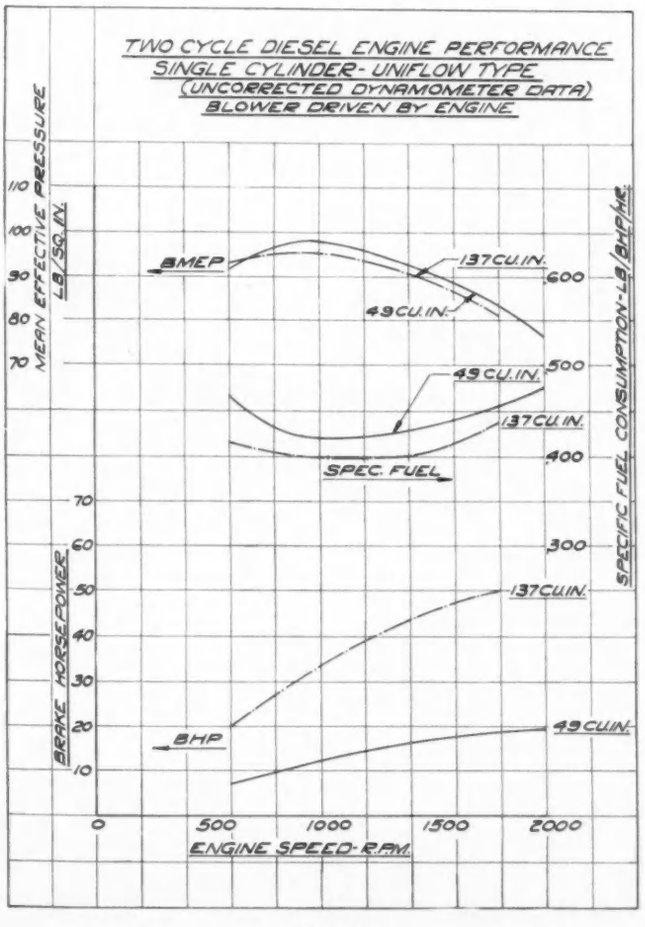


Fig. 1

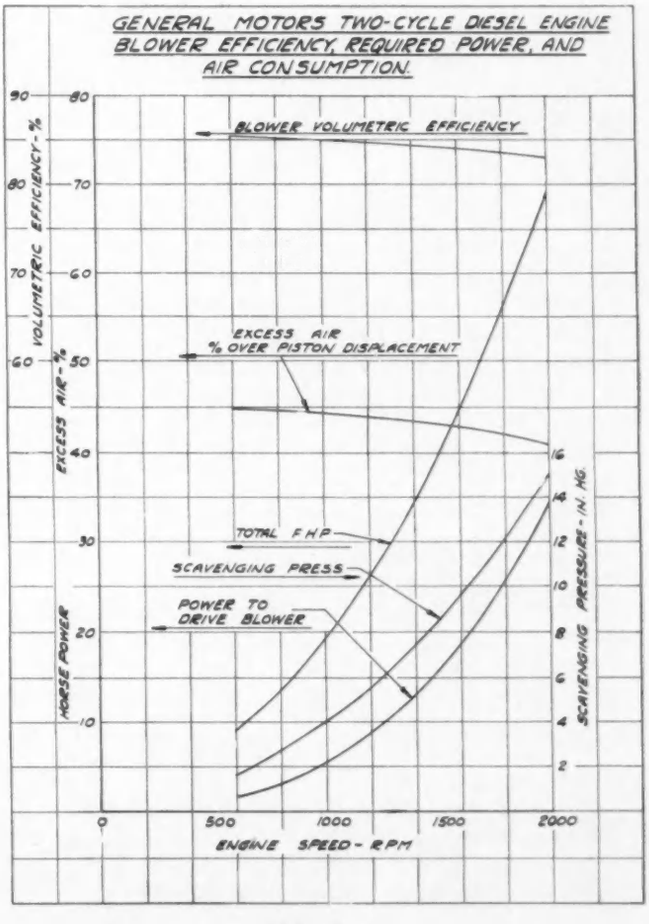


Fig. 2

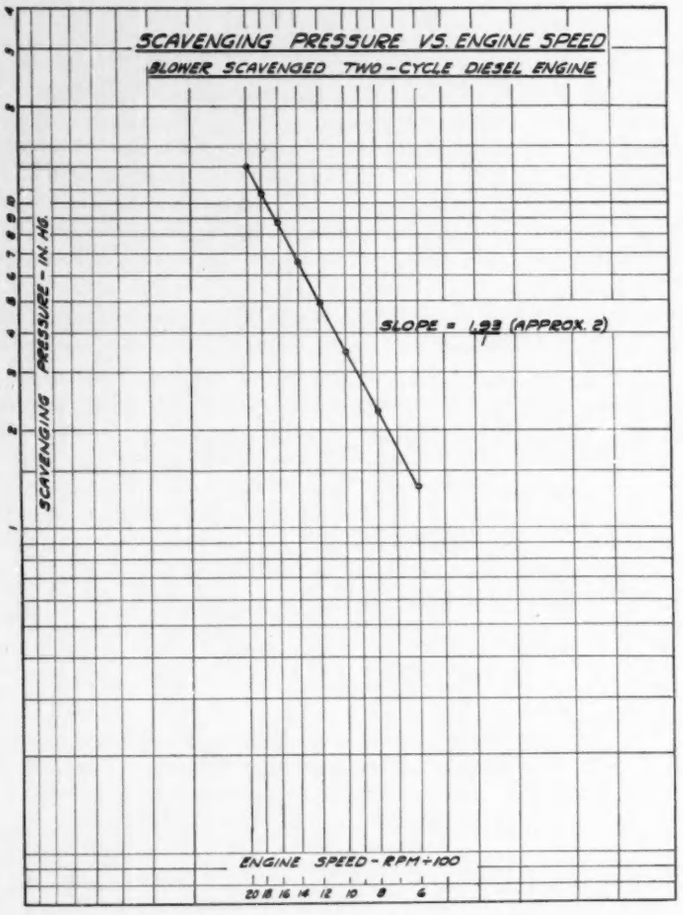


Fig. 3

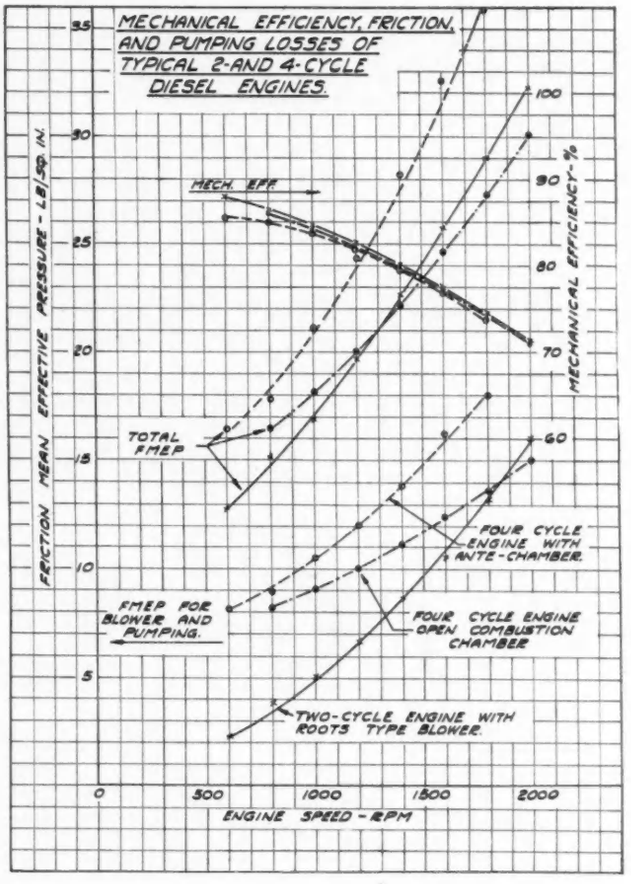


Fig. 4

paths and the engine in the same proportion at any given pressure. The volumetric efficiency of the complete blower-scavenged two-cycle engine is therefore practically constant over the entire speed range. The maximum speed is limited only by the permissible scavenging power loss. Actual air-consumption tests of multicylinder engines show but a slight drop in the blower volumetric efficiency at maximum rated engine speed. This characteristic is shown by the curve in Fig. 2 which gives the excess air factor and is the result of a test on an automotive-type engine.

Perhaps the best answer to the scavenging power controversy is a comparison of the mechanical efficiency of actual engines. Fig. 4 shows the mechanical efficiency, friction, and pumping losses of two representative modern four-cycle Diesel engines such as those specified in Table 1, and of a similar two-cycle engine. The fact that the two-cycle mechanical efficiency is slightly above each of the four-cycle engines would seem to answer this question. The curve of the power absorbed by the blower shows that about one-half of the total friction loss is required for the supply of scavenge air in the two-cycle engine. A pumping loss of about the same proportion is a common assumption for four-cycle engines.

No difficulty has been experienced in making poppet valves of the two-cycle engine operate at speeds well above the rated speeds quoted for the commercial engines in Table 1.

Offhand it would appear that, for optimum scavenging, the exhaust back pressure should be as low as possible, and that extremely large exhaust pipes and mufflers would be required as is common practice in large stationary installations. But it should be borne in mind that, in high-output engines, the function of the blower is to get air into the cylinder, and any excess that is blown into the exhaust is largely wasted except for cooling effects where badly needed. It has been found that, with proper port- and valve-timing, the power and economy actually are improved slightly by a reasonable back pressure and that mufflers of conventional design and capacity are quite satisfactory. The exhaust silencing has been found to be comparable with conventional commercial engines. Fig. 5 shows the effect of a large change in exhaust back pressure upon the power of an automotive-type two-cycle Diesel engine.

One of the most common objections advanced against the

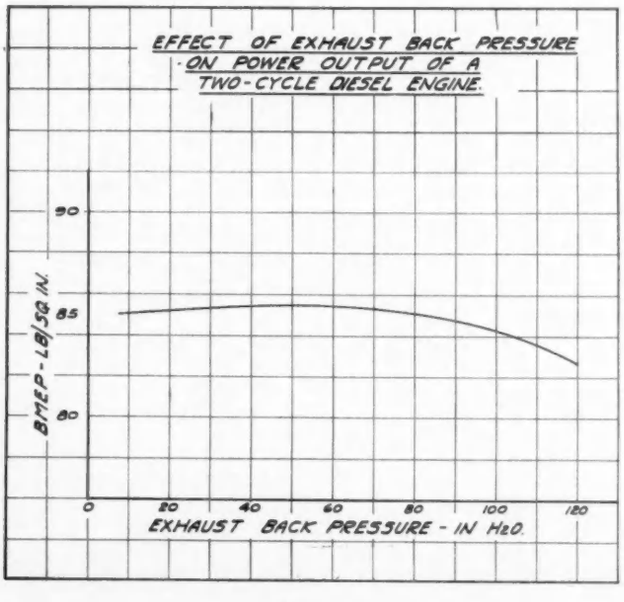


Fig. 5

high-speed displacement blower of the Roots type has been the annoying noise of the air inlet. Anyone who has worked with these units is familiar with this problem. An analysis of the air flow into such a blower with straight cycloidal rotors shows the displacement to vary in a most abrupt manner, causing violent pressure waves to be set up in the inlet and outlet. It has been found that, by modifying the cycloidal form and making the rotor lobes helical instead of straight, the inlet flow can be made absolutely uniform and the discharge greatly improved. Fig. 6 shows two indicator diagrams which were taken in the inlets of two blowers, one equipped with straight two-lobed rotors, and the other with

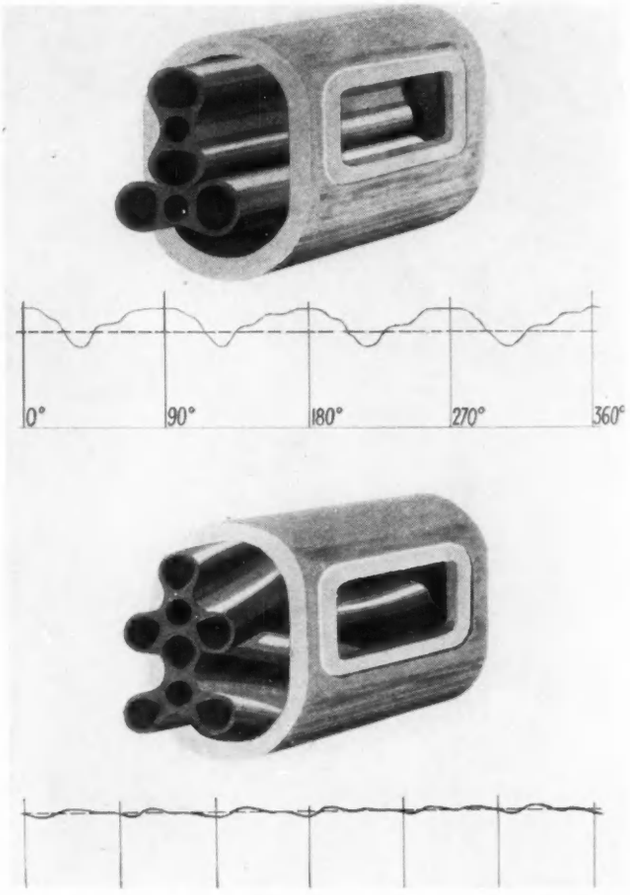


Fig. 6 – Comparison of Inlet Pressure Characteristics of Straight Two-Lobed Rotors Vs. Helical Three-Lobed Rotors

helical three-lobed rotors. The decidedly smoother pressure characteristics of the latter are evident. Experience with actual engines shows the intake noise with three-lobed helical rotors to be very much like an ordinary gasoline engine with the carburetor removed. Very satisfactory silencing can be obtained with conventional intake silencers or air cleaners.

Piston Cooling

In all internal-combustion engines piston and ring cooling is one of the first limitations to be encountered when the specific output is increased. This condition is particularly true of Diesel engines because of the higher pressures and combustion temperatures and, when output is increased further by supercharging or by use of the two-stroke cycle, the problem may become of first importance.

It has been found that, with normal piston design, the amount of heat liberated above the piston requires such a steep temperature gradient from the piston crown to the cylinder wall that the ring belt, which lies in this path, operates at a temperature above the critical point of ordinary lubricating oils, and ring-sticking is likely to result. The use of piston materials of higher thermal conductivity has not eliminated this trouble. After a very thorough investigation of the temperature gradient by means of fusible plugs and by thermocouples in engines under operating conditions, it was found that the heat path could be blocked off effectively by coring out the rim of the piston between the piston crown and the ring belt, and cooling the crown directly by an oil spray. This construction is shown in the sectional drawing of Fig. 7. This method reduces the temperature of the ring belt well below the coking or gumming point of ordinary oils and lowers the piston skirt temperature to a point where the usual taper is no longer necessary.

Since the piston inertia loads in a two-cycle engine reduce the maximum bearing loads, an iron piston can be used to advantage.

Injection System

So far nothing has been said about the fuel injection system as the discussion has been devoted largely to the effect of the two-stroke cycle on weight, power, size, and speed. Since the injection system departs in many ways from the usual pump-pipe-nozzle system it may be of interest to point out some of its characteristics.

Considering the injection problem in the reverse of the usual manner, it is obvious that, for optimum performance, the pressure in the cylinder is a definite function of the rate of combustion. In a like manner, the rate of combustion is a function of the rate of fuel supply which, in turn, is controlled by the velocity of the pump plunger or, finally, by the characteristic of the injection cam. Now, if the engine is to operate over a wide speed and load range, the rate of fuel delivery into the cylinder may need to vary 6 or 8:1. With fixed area spray orifices, this fuel variation requires a variation in pressure of about 40 or 60:1 and, for proper atomization at low speed, the pressures at high speed are likely to be 15,000 to 20,000 lb. per sq. in. Considering the fact that fuel oil is at least 100 times more elastic than steel, it is obvious

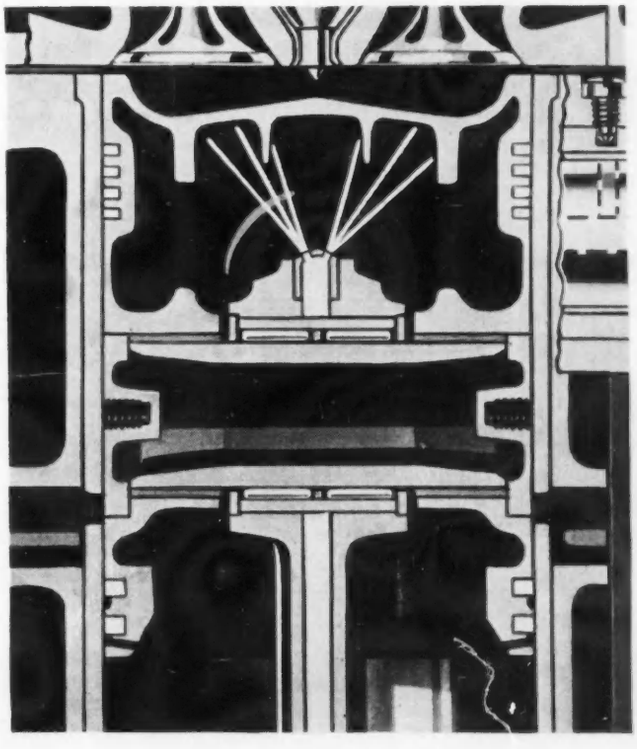


Fig. 7 – Piston Oil Cooling

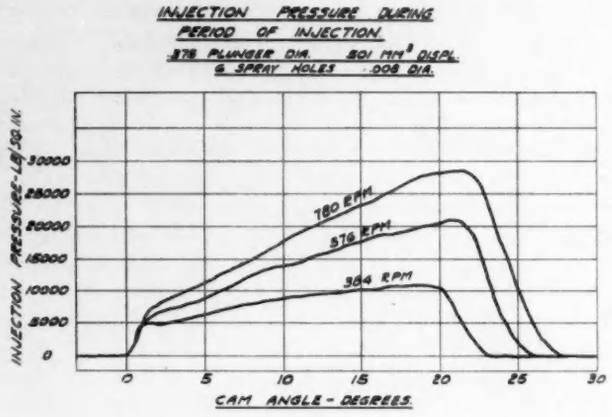


Fig. 8

that the mechanism for delivering the fuel into the cylinder, in accordance with the optimum rate designed into the cam, must be composed of more steel than fuel oil – hence the unit injector with the pump plunger close to the spray orifices. The simple spring-loaded discharge valve can be used to advantage when there are no high-pressure waves in the fuel-supply lines.

The diagrams in Fig. 8 show the pressure at the spray nozzle of a somewhat larger unit injector with a differential needle valve. The pressures were measured with a carbon-stack electrical indicator having a 1/16-in. piston in the nozzle itself, each card being the composite of five or six successive injections. It is also noteworthy that the pressure, and hence the instantaneous discharge, increases as the injection proceeds without any pressure waves or secondary injections even at the low speed of 384 r.p.m. Fig. 9 shows the computed instantaneous discharge velocity through the spray orifices.

The injectors used in the smaller Model 71 automotive engines are similar to the Model 201 preceding except the spring-loaded discharge valve noted previously is used in place of a differential needle valve. A cross-sectional view of the smaller injector is shown in Fig. 10 and its position in the cylinder, in Fig. 11.

Two-Cycle Engine Balance

In order to produce an engine meeting the present commercial requirements for freedom from vibration, it is necessary to consider the problem of engine balance. In a conventional four-cycle engine the crank throws are arranged in pairs, two pistons moving up and down together. One piston is on the working stroke while the other is acting as an air pump to scavenge and fill the cylinder with fresh air. With six or more cylinders this arrangement eliminates both primary and secondary inertia forces and inertia couples. It might be said that the scavenging air compressor is used to balance the engine.

In a two-cycle engine with equal firing intervals, the crank throws must each be in different planes and, with half the

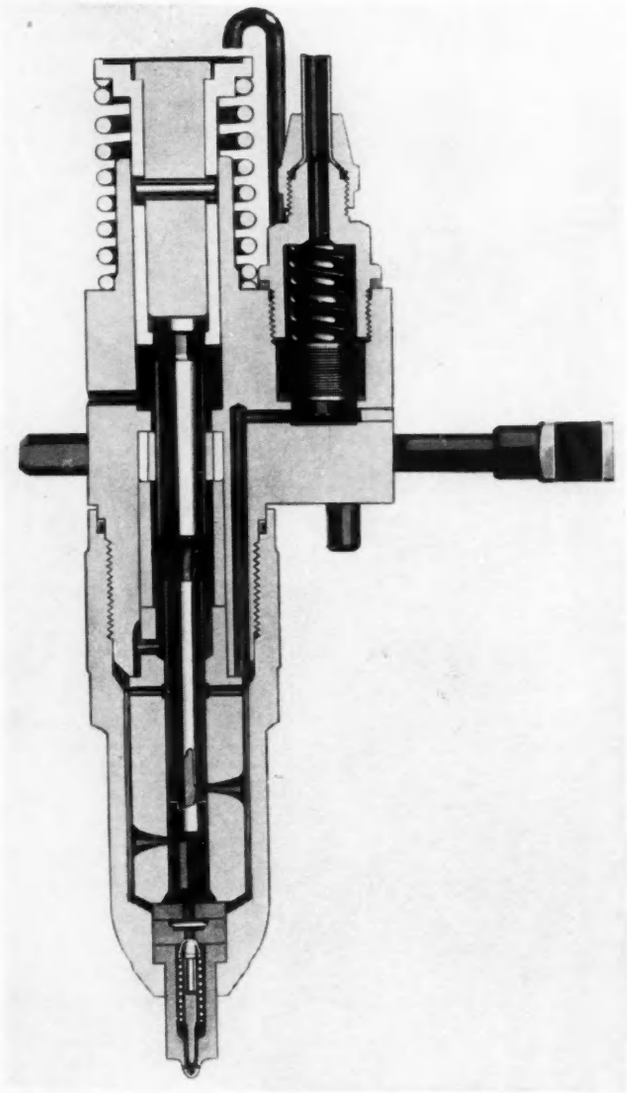


Fig. 10 – Cross-Section of Series 71 Unit Injector

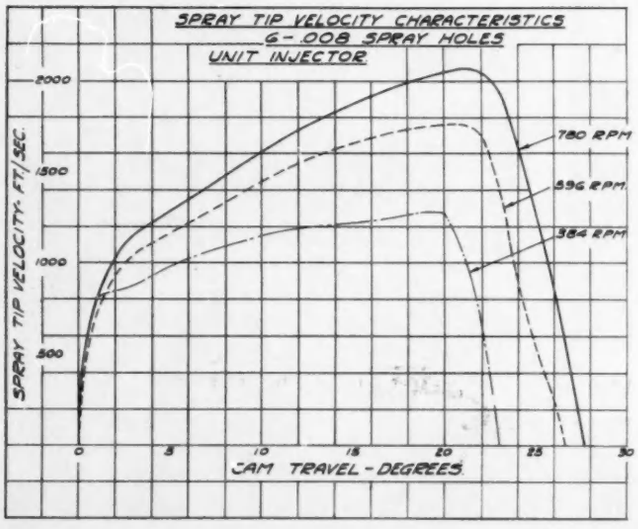


Fig. 9

number of cylinders as a four-cycle engine, there remains a primary or secondary couple due to the inertia forces of the pistons. The air compressor, usually being some kind of a rotary blower, cannot be arranged easily to produce the required counterbalancing forces, and therefore some other mechanism must be provided. But it is first necessary to determine the nature of the unbalance.

All in-line or vee arrangements of two-cycle engines with the cranks arranged to give equal firing intervals have some degree of dynamic unbalance. Table 3 has been arranged to show some of the more common crank arrangements and the nature and magnitude of the unbalance. The only appreciable unbalanced forces are those of either primary or secondary order, higher orders being negligible as in other engines. The unbalanced rotating forces of the crankshaft and the connecting-rod big-ends can be counterweighted in the conventional

Table 3 – Typical Two-Cycle-Engine Unbalanced Forces and Couples

NO. OF CYL.	CRANK ARRANGEMENT	REMARKS	FIRING ORDER	CYL. SPACING	INERTIA FORCES		ROCKING COUPLES	
					PRIMARY	SECONDARY	PRIMARY	SECONDARY
1			-----	----	km	pkm	0	0
2			1-2-1	EQUAL	0	2pkm	Klm	pklm
3		CAN NEGLECT SECONDARY COUPLE	1-3-2	EQUAL	0	0	1.732 Klm	1.732 pklm
4			1-3-2-4	EQUAL	0	0	1.414 Klm	4.0 pklm
		PREFERRED. DOES NOT REQUIRE PROVISION FOR SECONDARY COUPLE.	1-3-4-2	EQUAL	0	0	3.1623 Klm	0
5			1-4-3-2-5	EQUAL	0	0	.449 Klm	4.98 pklm
6		PREFERRED. DOES NOT REQUIRE PROVISION FOR SECONDARY COUPLE.	1-4-2-6-3-5 1-5-3-6-2-4	EQUAL	0	0	3.464 Klm	0
			1-5-3-4-2-6	EQUAL	0	0	0	3.464 pklm
			1-4-3-5-2-6	EQUAL	0	0	1.732 Klm	3 pklm
8		PREFERRED. DOES NOT REQUIRE PROVISION FOR SECONDARY COUPLE.	1-8-2-6 4-5-3-7	EQUAL	0	0	.45 Klm	0
			1-8-2-5 6-3-4-7	EQUAL	0	0	.13 Klm	1.41 pklm

K = .0000284 RN^2 R = CRANK RADIUS, IN. N = RPM
l = DISTANCE BETWEEN CYLINDERS, IN.
m = RECIPROCATING MASS PER CYLINDER, LB.
P = RATIO - CRANK RADIUS TO CONNECTING ROD LENGTH

manner. The inertia unbalance has been expressed in terms of the piston weight, connecting-rod length to stroke ratio, and cylinder spacing, with a suitable coefficient in each case so that, for engines of the same cylinder size, the relative unbalance is expressed by coefficients alone.

Considering the three-, four-, and six-cylinder engines, it is seen that the cranks can be arranged to eliminate all unbalance except a primary rocking couple in the four and six, and both a primary and secondary rocking couple in the three. In general, these couples are due to the pistons at one end of the engine moving opposite to those at the other end. This condition is shown in Fig. 12 in which typical four-cycle and two-cycle six-cylinder crankshaft arrangements are compared. In each the free up and down inertia forces all cancel out. The lack of a rocking couple in the four-cycle engine can be seen, as can the presence of an unbalanced couple in the two-cycle engine.

It so happens that the required counteracting primary forces, when acting at the ends of the cylinder-block, are almost exactly alike for the three-, four-, and six-cylinder engines. It has also been found in the three that, in an actual engine, the unbalance due to the secondary rocking couple is not appreciable and can be neglected. Thus it is possible to produce two-cycle engines of three, four, and six cylinders that are in balance and with the torque smoothness of six-, eight-, and twelve-cylinder four-cycle engines of similar power.

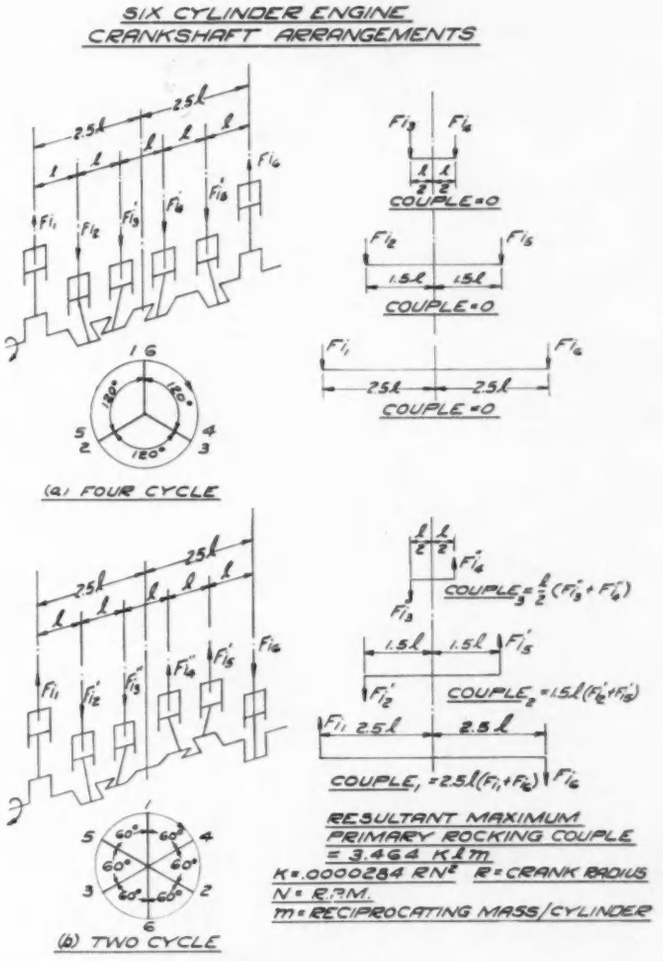


Fig. 12

By careful design of the crankshaft and reciprocating parts a natural torsional frequency of the crankshaft is obtained which is sufficiently high to permit governed speeds comparable to other engines of similar power. For higher speeds or where extra masses, in the form of heavy pulleys or drives, are connected to the front end of the crankshaft, a torsional-vibration balancer is required.

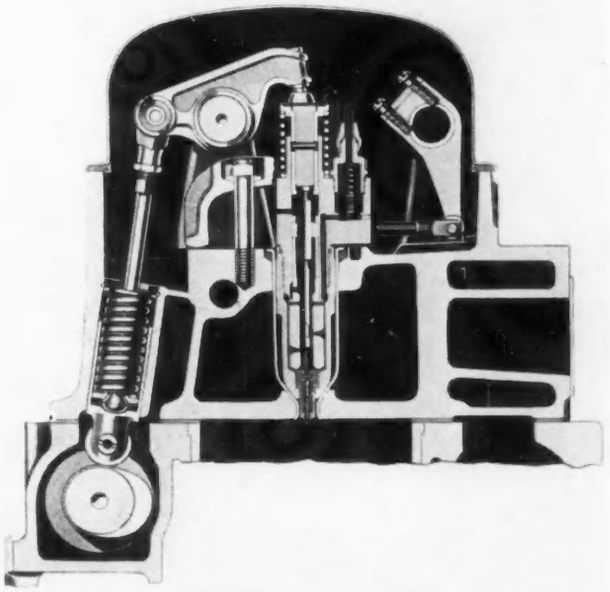


Fig. 11 – Unit Injector in Cylinder-Head with Operating Mechanism

Commercial Design Problem

These fundamental two-cycle problems were worked out during several years of investigation beginning with the large-cylinder sizes now used in the engines for Diesel-electric locomotives and proceeding to successively smaller cylinders as more experience was gained. Both single-cylinder laboratory

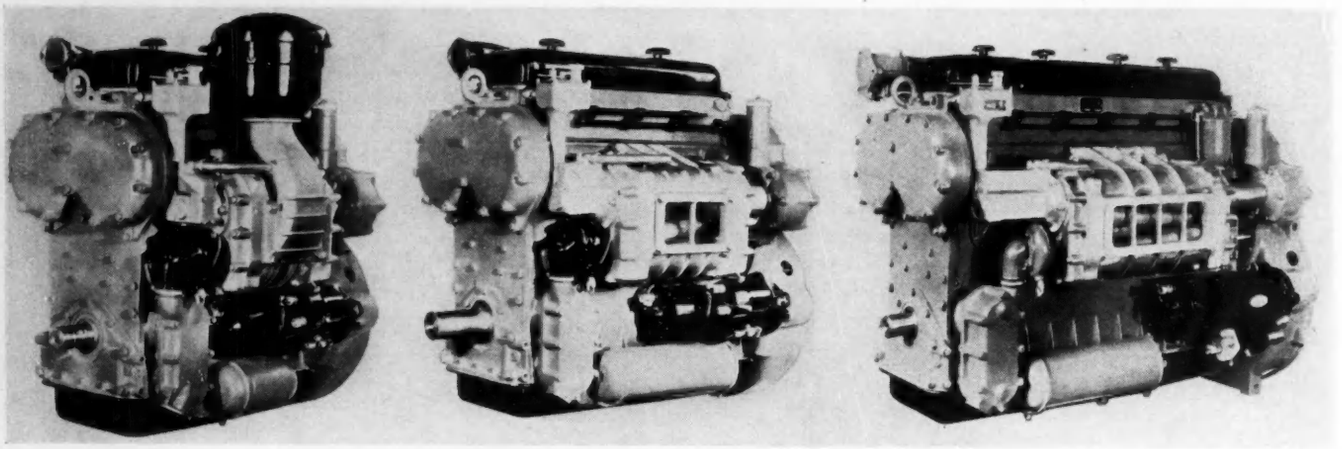


Fig. 13 – General Motors Series 71 Uniflow Two-Cycle Diesel Engines

Table 4 – General Motors Series 71 Two-Cycle Diesel Engines – Principal Specifications

	Model 3-71	Model 4-71	Model 6-71
Number of Cylinders	3	4	6
Bore, in.	4¼	4¼	4¼
Stroke, in.	5	5	5
Total Displacement, cu. in.	212.69	283.58	425.37
Maximum B.Hp.	82.5	110	165
Governed Maximum Speed, r.p.m.	2000	2000	2000
B.M.E.P. at Maximum Speed, lb. per sq. in.	77	77	77
Maximum Torque, ft-lb.	283	375	562
Maximum Torque, r.p.m. at	800–1000	800–1000	800–1000
Fuel Consumption, lb. per b.hp-hr.	0.45	0.45	0.45
Industrial Rating:			
Maximum B.Hp.	45	60	90
Governed Speed, r.p.m.	1200	1200	1200
Compression Ratio, Nominal	16:1	16:1	16:1
Piston Speed, ft. per min. at 2000 r.p.m.	1668	1668	1668
Firing Order, R. H. Rotation	1-3-2	1-3-4-2	1-5-3-6-2-4
L. H. Rotation	1-2-3	1-2-4-3	1-4-2-6-3-5
Weight of Basic Engine, lb. (dry)*	1160	1330	1635

* Engine weight includes: starting motor, speed governor, oil cooler, oil filter and fuel filter. Generator, cooling fan, intake silencer, air-intake elbow, and engine mountings are not included in the above weights.

engines and multicylinder service test engines were made in several sizes from 50 to 500 cu. in. displacement per cylinder. Having demonstrated that the two-cycle principle could be worked out in a small cylinder, there remained the problem of incorporating these developments in a commercial engine suitable for all the multitude of uses that are the natural field for a small Diesel engine. Since fuel economy is the outstanding feature of the Diesel engine, its greatest usefulness will be in those services where the most fuel is used, such as in truck, coach, marine, and industrial powerplants. To cover this wide range with one type of engine requires a high degree of adaptability as to power, speed, accessories, and installation arrangement. Furthermore, the present market for Diesel engines in any one field will not support a volume production of each size and model, so that there remains the additional problem of using the same parts for a multitude of models. If this purpose can be accomplished, the advantages of volume production in parts are gained and, at the same time, a supply of a limited quantity of any required model.

The uniflow two-cycle Diesel engine with a unit fuel injector lends itself naturally to such a program. With one cylinder size the power can be increased simply by adding more cylinders. The cross-section and ends of the engine remain the same for any number of cylinders so that, from the design and manufacturing standpoint, the principal variation among all models is length.

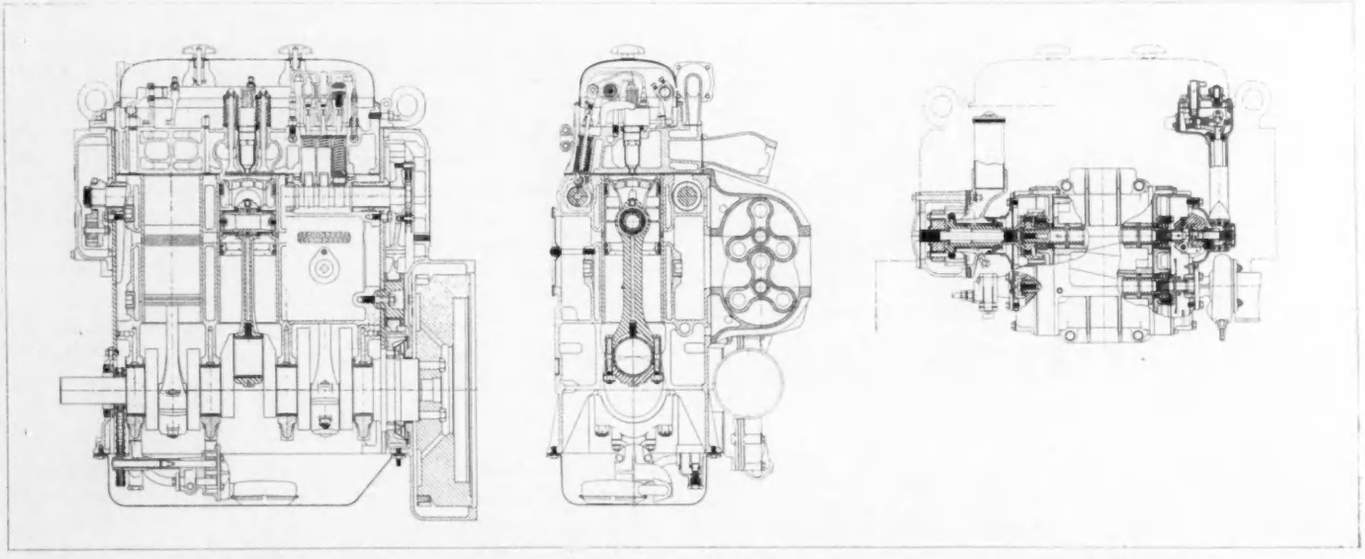


Fig. 14 – Cross-Sectional Views of General Motors Model 71 Uniflow Two-Cycle Diesel Engine

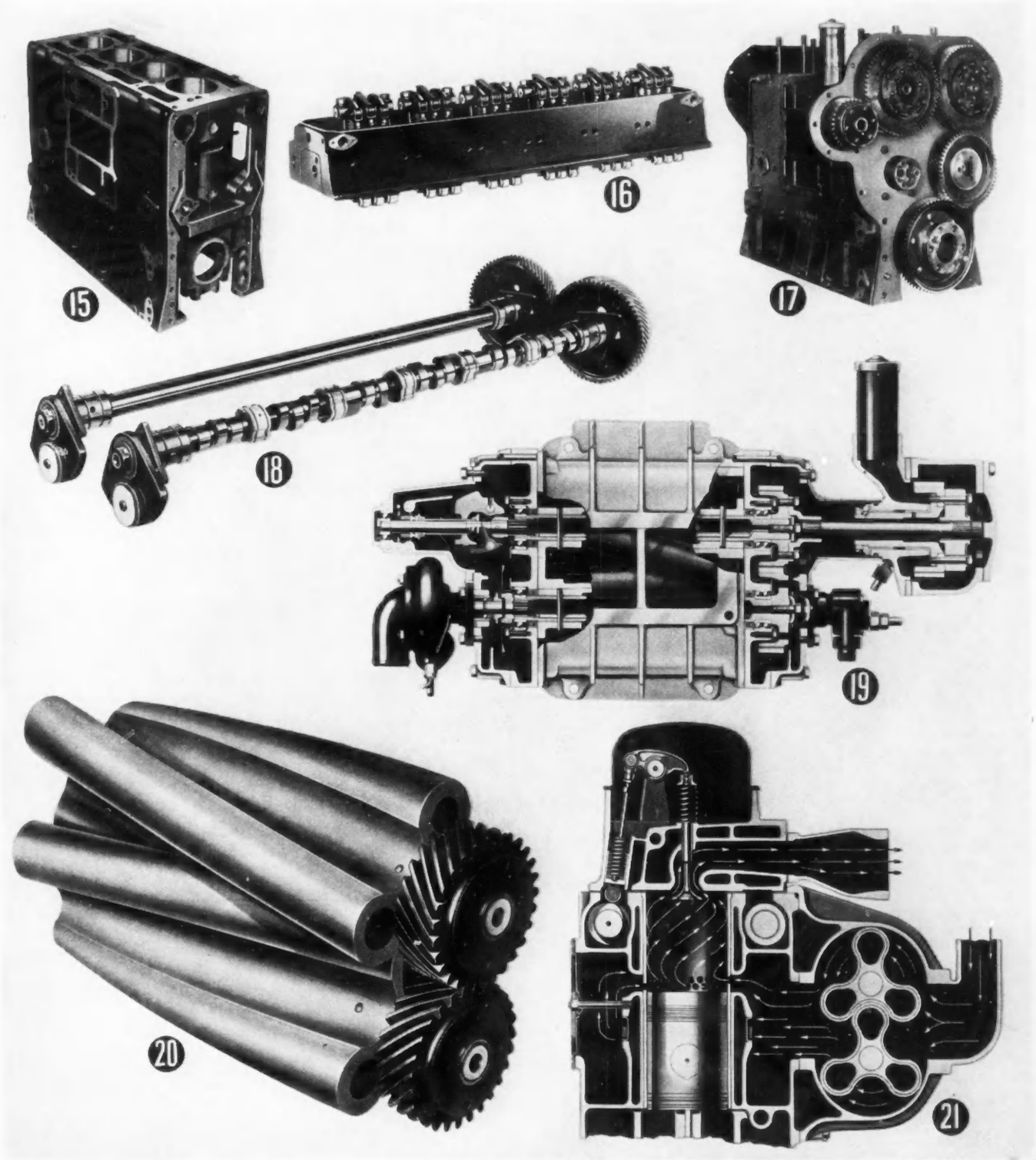


Fig. 15 – Crankcase and Cylinder Block

Fig. 16 – Cylinder-Head Assembly

Fig. 17 – Gear Train

Fig. 18 – Camshaft and Balancer Shaft with Balance Weights, Gears, and Camshaft Intermediate Bearings

Fig. 19 – Scavenging Blower Assembly

Fig. 20 – Scavenging Blower Rotors and Timing Gears

Fig. 21 – Uniflow Scavenging System

How this principle has been worked out in the General Motors Series 71 engines is best shown by the photographs and drawings of the engines themselves. Table 4 gives the principal specifications and Fig. 13 shows the photographs of the three-, four-, and six-cylinder engines from left to right respectively. Their similarity is apparent. The longitudinal section, cross-section, and end view are shown in Fig. 14.

The cylinder-block, which is identical except for length in the multicylinder models, is shown in Fig. 15. The symmetry of design, both lengthwise and crosswise, is apparent. This construction permits turning the block end for end and thus placing all the accessories on the opposite side.

The one-piece overhead-valve cylinder-head assembly is shown in Fig. 16. The head studs are arranged symmetrically so that the head can be put on the engine either way in order to place the exhaust manifold on either the right or left side,

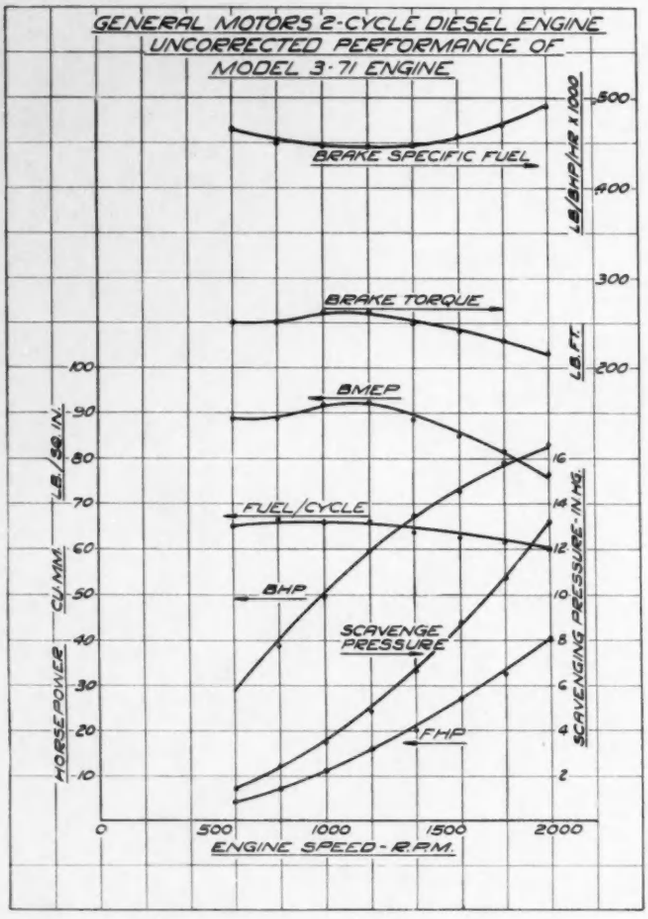


Fig. 22

regardless of the blower position or the direction of engine rotation.

The gear train, shown in Fig. 17, is at the flywheel end of the engine and drives the camshaft, balancer shaft, and the blower. The direction of crankshaft rotation is changed by placing the idler gear on the opposite side, the direction of rotation of all the other parts remaining the same. The camshaft and balancer shaft are interchangeable to permit reversing the cylinder-head.

The balancer for the primary rocking couple is shown in Fig. 18. The plain shaft rotates in the opposite direction from the camshaft. The two balance weights at the front end produce up and down forces and the two counterweights cast integral with the camshaft and balancer shaft gears at the rear end produce corresponding down and up forces, resulting in a fore and aft rocking couple that balances out the equal and opposite couple of the piston inertia forces. These forces are so near alike for all three engines that the same gears are used on all with the addition of a small counterweight inside the gears of the four- and six-cylinder engines.

Fig. 19 shows the scavenging blower assembly including the governor, fuel pump, and water pump which are all driven directly from the rotor shafts. The helical three-lobed rotors are shown in Fig. 20.

The fundamental principle of uniflow scavenging is shown clearly in Fig. 21 which is a sectional view through the blower, air box, cylinder, and cylinder-head.

In a similar manner, the unit injector is referred to again, as shown in Fig. 11. The direct operation of the pump plunger by the engine camshaft and the thorough cooling of the injector are apparent.

The unit injector itself, as shown in Fig. 10, is referred to again. Each injector is a complete self-contained fuel pump and spray nozzle which can be removed easily for inspection

Table 5 – General Motors Two-Cycle Diesel Engines – Weight and Power

Model	No. of Cylinders	Displacement, cu. in.	Rated B.Hp.	Rated Speed, r.p.m.	Dry Weight, lb.	Weight, lb. per cu. in.	Weight, lb. per b.hp.
371	3	212	82.5	2000	1160	5.47	14.1
471	4	284	110.0	2000	1330	4.69	12.1
671	6	425	165.0	2000	1635	3.85	9.9

Table 6 – Comparison of Overall Dimensions – Four-Cycle and Two-Cycle Diesel Engines of Approximately the Same Rated Output

			Overall Dimensions, in.*			Installation Volume	
Engine Type	Rated B.Hp.	Displacement, cu. in.	Length	Height	Width	Cu. Ft.	Per Cent of Four-Cycle
Four-Cycle	83	298	39	32 1/4	22 1/2	16.4	
Four-Cycle	72	274	42 3/4	36 3/4	25 3/8	22.9	
Two-Cycle	82.5	212	31 3/4	35 1/4	23	14.9	73
Difference between two-cycle and average of four-cycle engines			−9 1/8	+3/4	−15/16		
Four-Cycle	120	474	44 3/4	38	25 7/8	25.5	
Four-Cycle	103	468	48	39 3/8	24 1/2	26.9	
Two-Cycle	110	284	38 7/8	35 1/4	23 3/4	18.8	72
Difference between two-cycle and average four-cycle engines			−7 1/2	−3 7/16	−1 7/16		
Four-Cycle	176	707	54 1/2	42 3/4	26 1/2	35.8	
Two-Cycle	165	425	50 3/8	35 1/4	25 1/8	25.9	74
Difference between two-cycle and four-cycle engines			−4 1/8	−7 1/2	−1 3/8		

* Does not include air cleaner or fan.

Table 7 – Comparison of Engines – Gasoline, Four-Cycle Diesel, Two-Cycle Diesel

	Rated B.Hp.	Weight, lb. per cu. in.	Weight, lb. per b.hp.	Length Less than Four-Cycle Diesel, in.	Height Less than Four-Cycle Diesel, in.	Width Less than Four-Cycle Diesel, in.	Installation Space, per cent of Four-Cycle Diesel
Gasoline	68/73	2.66	10.2				
Four-Cycle Diesel	73/82	3.88	14.5				
Two-Cycle Diesel	82	5.47	14.1	9.1	−0.7	1.0	73
Gasoline	104/105	2.03	9.2				
Four-Cycle Diesel	103/120	3.41	14.4				
Two-Cycle Diesel	110	4.60	12.1	7.5	3.5	1.4	72
Gasoline	148	2.56	12.2				
Four-Cycle Diesel	176	3.56	14.3				
Two-Cycle Diesel	165	3.85	9.9	4.1	7.5	1.4	74

or replacement without disturbing any of the other injectors. Fuel from the transfer pump circulates through all the injectors, thus cooling the working parts and also preventing air binding even when starting from a dry tank.

The other working parts of the engine such as the water pump, oil pump, oil cooler, oil filter, and so on, are quite conventional and need no particular discussion. As a matter of fact, most of the parts of the engine can be found in the ordinary internal-combustion engine. The assembly and adjustments are all in accordance with well-known manufacturing and service practices.

Results

But, in the final analysis, the engine user is interested mostly in what the engine will do. This performance is shown in Fig. 22, which is an uncorrected dynamometer test of a model 3-71 engine. The four- and six-cylinder models have similar characteristics with power in proportion to the number of cylinders.

As pointed out at the beginning of this discussion, the object of the development was to produce a light, compact, Diesel engine suitable for automotive and industrial purposes. Table 5 gives the actual weight, power, and size of these two-cycle Diesel engines. Table 6 gives the average overall dimensions of two four-cycle Diesel engines and similar dimensions for the two-cycle, from which the difference in length, width, and height and the reduced installation space have been determined. The gasoline engines are not shown but are approximately the same dimensions as the four-cycle Diesels. The saving in length of the two-cycle design is a most important item in vehicle installations.

The data from Tables 1, 5 and 6 have been consolidated in Table 7. This table gives a concise summary of the specific weight, power, and size of the gasoline, the four-cycle Diesel, and the two-cycle Diesel engines covered by this discussion. From a strict design point of view, the six-cylinder, two-cycle engine should be used in making comparisons because it represents the basic design, the four- and three-cylinder models having been made by simply taking out cylinders in the center of the engine. Therefore, considering the six first, it is seen that it is equal to the gasoline engine in weight per horsepower and both lighter and smaller than the four-cycle Diesel. The four-cylinder two-cycle Diesel is heavier than the gasoline engine but lighter and smaller than the four-cycle Diesel, whereas the three-cylinder two-cycle model is about the same weight as the four-cycle Diesel but very much shorter. Obviously, with designs made specifically for the four- and the three-cylinder models, the preceding weights can be reduced considerably.

Since these engines have been placed on the market only recently, it is reasonable to expect the same rate of improvement as is common to all commercial engines. Their present rating is well below the potential capacity of the cylinder, and further development should insure a gradual improvement in overall performance.

Discussion

N.A.C.A. Experience Indicates Aluminum-Alloy Pistons

–Ernest G. Whitney

Associate Mechanical Engineer, National Advisory Committee for Aeronautics

THE ability of the uniflow two-stroke-cycle principle to produce high power outputs per unit of piston displacement has been ably demonstrated by the Junkers opposed-piston two-stroke aircraft Diesels which have so much successful flying to their credit and which recently established a world's distance record for seaplanes. Further substantiation of the two-stroke Diesel's capability for high outputs has been obtained, as mentioned by Mr. Shoemaker, in single-cylinder tests by the National Advisory Committee for Aeronautics. Power controlling factors introducing design differences peculiar to the two-stroke cycle include inlet and exhaust areas and timings, scavenging air pressure, and direction of scavenging air introduced into the cylinder. As stated by Mr. Shoemaker, the injection process and combustion-chamber detail to assist mixing of the fuel and air do not differ substantially from those of the four-stroke Diesel except in so far as the combustion-chamber shape may be influenced by the poppet-valve arrangement.

With regard to the use of poppet exhaust valves, it seems indicated by the Committee's tests that a limit in maximum practicable engine speed will be determined thereby at around 2500 r.p.m. This deduction resolves from the high valve-mechanism stresses introduced at high speeds by the requirements of rapid exhaust-valve opening to drop the cylinder pressure to that of the entering air without undue sacrifice of efficiency by early opening of the valves on the power stroke.

Concerning the General Motors adoption of cast-iron pistons in preference to those of aluminum alloy, the experience of the N.A.C.A. has been in direct disagreement. Cast or forged aluminum-alloy pistons with no auxiliary cooling devices are used successfully and exclusively in both two- and four-stroke Diesel test engines at specific outputs double those at which the General Motors units are rated. A single attempt to substitute a cast-steel piston of equal weight for the conventional aluminum-alloy construction in one of the four-stroke test engines resulted in prohibitive overheating at quite moderate loads. Attempts to oil-cool the piston by the spray method used by General Motors were unsuccessful, although a more prolonged effort might have produced better results. Nevertheless, it would appear that aluminum-alloy pistons should be capable of successful adaptation to the automotive-type two-stroke engine and be a desirable application for, although the added weight of the cast-iron piston may reduce somewhat the bearing and rod stresses, the overall engine weight is increased and not only by the piston weight but by the extra counterbalancing weight required to neutralize the tipping moments.

In further consideration of the weight question, it would be interesting if Mr. Shoemaker could estimate the increase in engine weight attributable to the counterbalancing gears, shafts, weights, and so on, necessitated by the tipping moments. It is noted that the General Motors two-stroke automotive engines described in Table 4, when introduced into their respective displacement classes of Table 1, show an average ratio of lb. per cu. in. of 2.13, lb. per hp. of 1.33, and hp. per cu. in. of 1.58 when using values of 1.00 for the gasoline engine. It then appears that the two-stroke Diesel under consideration is heavier

than the similar displacement four-stroke Diesel by the ratio of 2.13:1.59 although the lb. per hp. favors the two-stroke-cycle by the ratio of 1.33:1.51. It is assumed that some of this excess weight of the two-stroke engine comes from the form used to make for interchangeability of parts and flexibility of rotation direction and installation for all units, and possibly could be reduced by careful design for any particular installation. To compete with the gasoline automotive type on a weight basis, however, it appears that the two-stroke output must be increased one-third – which should be quite possible by further development.

It would be interesting if Mr. Shoemaker could reveal whether the engine timing permits boosting or not, what is the order of magnitude of the maximum cylinder pressures, and whether he has any determinations of injection rates.

Supports Three of Author's Conclusions

– P. H. Schweitzer
The Pennsylvania State College

MR. SHOEMAKER'S curves show that the two-cycle engine is not behind the four-cycle in mechanical efficiency even if the power to drive the blower is considered as part of the friction. Our experience confirms this conclusion. We have measured on a 5- by 6½-in. two-cycle independently charged engine 11.2 lb. per sq. in. friction m.e.p. at 900 r.p.m. and 136 deg. fahr. lubricating oil temperature with atmospheric manifold pressure, and about 3 lb. per sq. in. more with 10 in. hg. Furthermore the increase of the friction with speed was only about 0.4 lb. per sq. in. per 100 r.p.m., which checks remarkably with Mr. Shoemaker's Fig. 4 if the blower m.e.p. is deducted. The M.A.N. formula cited by Mr. Heldt,[a] which obviously refers to four-stroke-cycle engines, would give a friction increase of 1.2 lb. per sq. in. per 100 r.p.m. for a General Motors size engine. The explanation of the good showing of the two-stroke-cycle engine is that it is relieved from the breathing difficulty which is grave with the high-speed four-stroke cycle engine with tortuous air passages.

I was interested in Mr. Shoemaker's statement that an increase in the exhaust back pressure occasionally increases the power output. That is exactly what we have found, in one case a 7 per cent increase in power output just by throttling the exhaust. Our interpretation of this result differs, however, from Mr. Shoemaker's. We decided that the time integral of our exhaust opening was too much and, for best results, should be reduced. One way to discourage excess air from going through the engine is by throttling the exhaust; another way is by closing the exhaust sooner. The second method is clearly preferable because it helps to trap more air. If by throttling the exhaust the output increases, I would say that the original port or valve timing is not the optimum, at least for that particular speed.

I can subscribe fully to Mr. Shoemaker's statement that a two-cycle engine behaves like a fixed orifice. By changing both the intake and exhaust pressures we have found that the air consumption of the engine was independent of the intake pressure as long as the pressure difference, intake minus exhaust, was kept constant. We have found a straight-line relation between air consumption, in cu. ft. per min., and difference between intake and exhaust manifold pressure, in in. hg.

The injection pressures shown on Fig. 8 are very attractive and the injection system is no doubt an important contributor to the favorable results obtained with the engine. On Fig. 9 the notation of the ordinate "Spray Tip Velocity" is, I believe, erroneous and spray velocities probably have been plotted.

Further Comparison of Two-Cycle and Four-Cycle Diesels

– E. S. Dennison
Electric Boat Co.

THE author reasons plausibly that the surest way to his goal is to depart from the four-stroke in favor of the two-stroke cycle, since the output from a given cylinder is at once doubled. But there are reservations to this simple bit of arithmetic, and the tabulations give evidence that these reservations are of the first order. Enough facts are at hand to permit a comparison of four-cycle and two-cycle Diesels in their present states of development, and as built for automotive service.

Were it possible to produce two-cycle engines weighing one-half as much per horsepower as the equivalent four-cycle engines, plus one pound per horsepower, then the two-cycle weights should be found to lie in the range 7 to 10 lb. per hp., not 10 to 14. Much more enters into the question than the weight of a blower. It will be clearer to refer to weights per unit of displacement. The six four-cycle Diesels (Table 1) average 3.78 lb. per cu. in. The three two-cycle Diesels (Table 5) average 4.67 lb. per cu. in., or about 25 per cent higher. This differential is doubtless enough to cover the blower and its drive, together with the balancing elements which are peculiar to the subject design. If the output per cubic inch were actually doubled, a wide margin of weight would remain, favoring the two-cycle. Unfortunately, upon closer inspection most of this margin disappears.

Breathing restrictions, so-called, affect two-cycle and four-cycle engines in quite different degree. At a piston speed of 1668 ft. per min., 13 in. head of mercury is needed to supply air to the two-cycle cylinder (Fig. 22), a state of affairs which would place the four-cycle engine out of action, or nearly so. Two valves, open during 120 deg. crank angle, are not equivalent to one valve open during 240 deg. In the abbreviated period of opening, dwell is practically eliminated – the valves are always in the act of either opening or closing. Through the restricted passage, a gas volume 40 per cent in excess of piston displacement must be passed, again in contrast to the four-cycle engine. The difference affects both b.m.e.p. and the running speed at which it may be produced. For the two-cycle engine, Table 4 quotes 77 b.m.e.p. at 1668 ft. per min. piston speed. A corresponding limit in four-cycle practice is about 100 b.m.e.p. at 1800 ft. per min.; these figures are within reach of several current designs.

To gain a correct idea of relative weights in terms of pounds per horsepower, one must combine all three factors entering into it – weight per cubic inch, mean effective pressure, and running speed. Accepting the figures just quoted for each of these items, the two-cycle engine retains a weight advantage of about 15 per cent. A margin of this order is perhaps a permanent one. But there is nothing drastic in a reduction amounting to 15 per cent. It is scarcely enough to open up for the two-cycle engine any field of application which is not also open to the four-cycle. For most purposes the two are interchangeable as to weight, and are likely to remain so.

Four-cycle high-speed Diesel design has advanced in recent years by the aid of auxiliary combustion-chambers, working according to various systems. The writer is personally in contact with one development which illustrates the present trend in a convincing way. Maximum cylinder pressure does not exceed 700 lb. per sq. in., 40 per cent higher than in an equivalent gasoline engine – not twice as high. Fuel injection pressure is about 4200 lb. per sq. in., and nothing is gained by attempting to raise it. Performance is agreeably insensitive to a moderate variation of injection timing. A higher load may be carried with acceptable exhaust than in any available example of an open combustion-chamber, and with improved economy.

It is no coincidence that scarcely one four-cycle automotive Diesel today has unaided direct injection of fuel, whereas the two-cycle offerings have direct injection and nothing else. Cooling and scavenging set limits which the designer cannot escape. In consequence, the automotive two-cycle Diesel is speedily being left in sole possession of all cylinder pressures in excess of 800 lb. per sq. in., and all injection pressures in excess of 5000 lb. per sq. in. This situation, together with an aggravated cooling problem, imposes upon the builders a duty of pioneering in materials and methods. It is evident that in this case these efforts have met with a considerable degree of success.

Claims Two-Cycle Superiority Well Established

– L. B. Jackson
Fairbanks, Morse & Co.

MY first reaction upon reading this paper was that it was a defense of the two-cycle engine. The standing and performance of the two-cycle engine are such that no defense is necessary. Many years of satisfactory operation in a wide horsepower range have demonstrated conclusively its superiority both in reliability and efficiency; its simplicity is proverbial.

It is the writer's opinion that Mr. Shoemaker's company, in developing the type of two-cycle engine that it has chosen, has selected the type most difficult from an engineering viewpoint. The resources, technical skill, and manufacturing ability of his company undoubtedly will make it possible for it to make this a successful engine – something that other manufacturers of this type failed to do. It would be interesting to know whether in its research work it gave consideration to other two-cycle uniflow types of construction such as sleeve-valve or slide-valve construction.

Referring to Table 3, two-cylinder engine, the secondary rocking coupling depends upon the point about which moments are taken. If this point is midway between the cylinders, it is zero. The word "equal" in the column of cylinder spacing should be deleted.

The remarks as to the preferred firing orders of the four- and six-cylinder engines apply only when pairs of geared counterweights revolving once per revolution are added to each end of the engine. Considerably smaller geared counterweights would be required by the second six-cylinder firing order where revolved twice per revolution. This condition would indicate that the second listed firing order was preferable.

We regret that the graphs of Fig. 8 do not show results at 1000, 1200, 1500, and 1800 r.p.m.

[a] See "Some Recent European Developments in High-Speed Diesel Engines," by P. M. Heldt, presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 14, 1938.

Rating Aviation Fuels in Full-Scale Aircraft Engines

Completion of First Program and Progress on Second Program—Report of the Cooperative Fuel Research Committee

Presented by H. K. Cummings
Director, C.F.R. Full-Scale Detonation Projects

DATA in this report represent work carried out in accordance with assignments made in the first report of the C.F.R. Committee presented at the 1936 Annual Meeting of the Society, as follows:

1. Establish the validity of the C.F.R. Recommended Procedure for Rating Fuels in Full-Scale Aircraft Engines for fuels above 87 octane number.

2. Conduct full-scale engine tests in the range from 87 octane number to the highest octane number available.

3. Concurrently with Assignment 1, develop or revise knock-test methods leading to correlation with full-scale engine data.

The data presented are in accordance with Assignments 1 and 2, and the data required for Assignment 3 have been obtained in part. Further tests on olefinic fuels are reported.

Foreword

THE first report of the Cooperative Fuel Research Committee[1] covered the investigation in four representative full-scale engines of three series of base fuels:

a. Those whose knock characteristics are essentially unchanged under varying test conditions.

b. Those whose knock-rating decreases sharply with increasing severity of test conditions.

c. Those whose knock-rating improves with increasing severity of test conditions.

It arrived at the following general conclusions:

"The (a) and (a) plus lead fuels are evaluated correctly by the Motor Method. This result would be expected since the Motor Method is evaluated in terms of the agreement between the reference fuels and the test fuels – and, in this case, they are of substantially similar type.

"In the higher octane-number brackets the (a) plus benzol fuels are overrated by the Motor Method as shown by their behavior in some aircraft engines. Fuels containing large fractions of benzol have been found to be erratic and subject to wide variations in behavior with changes in engine type and operating conditions. Although benzol is the only aromatic fuel included in the work, it would appear that, where fuels of more than 80 Motor-Method octane number are concerned, the Motor-Method rating should be regarded with caution and verified by service-test data for any particular engine and installation if the total aromatic content exceeds 40 per cent by volume.

"Fuels of the (b) series are sensibly rated correctly by the Motor Method. This conclusion is subject to confirmation if fuels of this type (of high acid heat) of higher octane number than those tested should become available later.

"The Motor Method overrates the gasoline blends of the (c) series by approximately two octane numbers in the range 73 to 80."

The first report also outlined the further work to be carried out as follows:

"1. Establish the validity of the C.F.R. Recommended Procedure for Rating Fuels in Full-Scale Aircraft Engines for fuels above 87 octane number.

"2. Conduct full-scale engine tests in the range from 87 octane number to the highest octane number available.

"3. Concurrently with Item 1, develop or revise knock-test methods leading to correlation with full-scale engine data."

1. Introduction

This report presents the data which have been obtained in accordance with Assignments 1 and 2 just given. Data required for Assignment 3 have been obtained in part. These data should be sufficient to guide the groups working on Laboratory Detonation Projects in making suitable knock-test modifications. At the time the first report of the C.F.R.[1] on the rating of a variety of fuels in full-scale aircraft engines was approved for publication, it was decided that further tests on olefinic fuels were desirable. It was believed that full-scale ratings of one such fuel were insufficient proof that A.S.T.M. ratings of all olefinic fuels would correlate with their full-scale behavior. Further tests on olefinic fuels were recommended, and it was agreed that two unleaded 80-octane-number fuels (straight-run gasoline plus unhydrogenated polymer products) and one 80-octane-number leaded fuel made from a base consisting completely of vapor-phase cracked stock should be tested in one engine. These tests have been completed, the data analyzed, and the results are presented in this report.

At about the time the first C.F.R. report on full-scale ratings of aviation fuels was presented, the National Research Council of Canada (N.R.C.) became a cooperating member.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 13, 1938.]

[1] See S.A.E. TRANSACTIONS, Vol. 31, May, 1936, pp. 161-175: "Rating Aviation Fuels in Full-Scale Aircraft Engines," by C. B. Veal.

The N.R.C. lacked engine equipment suitable for test of fuels of above 87 octane number but wished to participate in the C.F.R. program with fuels in the 73 to 87 octane-number bracket. Sufficient supplies of most of the fuels used in the C.F.R. work in the 73-87 octane-number bracket were available, and these fuels were furnished to the N.R.C. together with certain supplementary fuels (to replace fuels which had been used up entirely). The work of the N.R.C. has been completed, and a comprehensive report rendered. The Canadian data have been analyzed, and the results are presented in this report. This work completes the original program.

Full-scale tests in the octane-number bracket above 87 have followed in general the program outlined in the first report.[1] Full-scale matching tests of C.F.R. M-1 and S-1 reference-fuel blends and of S-1 plus lead with current commercial leaded fuels of 87 A.S.T.M. octane number and of 100 Army octane number have been carried out by three laboratories, and the data from two laboratories have been analyzed and are presented in this report. Complete data on the results from the third laboratory have not yet been submitted for analysis. One laboratory has completed tests of 100 Army octane number leaded blends, one containing isopropyl ether, and the other a high percentage of aromatics. Full data on these tests have not yet been submitted for analysis.

2. Canadian Tests

a. Engine – The engine used was an Armstrong-Siddeley fourteen-cylinder double-row radial air-cooled Jaguar. This engine has a bore and stroke of 5 by 5½ in. (capacity 1512 cu. in.), has one exhaust valve and one intake valve per cylinder and, for these tests, used the standard compression ratio of 5:1. The centrifugal blower impeller is driven at 12.9 times crankshaft speed. The normal rating of the engine is

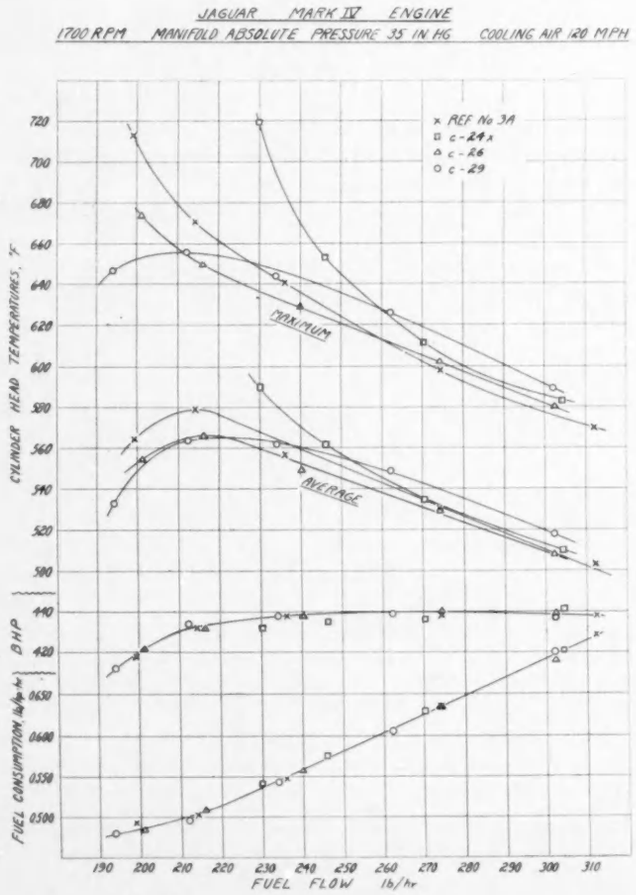


Fig. 1 – Typical Comparison of Three Test Fuels with One Reference Fuel

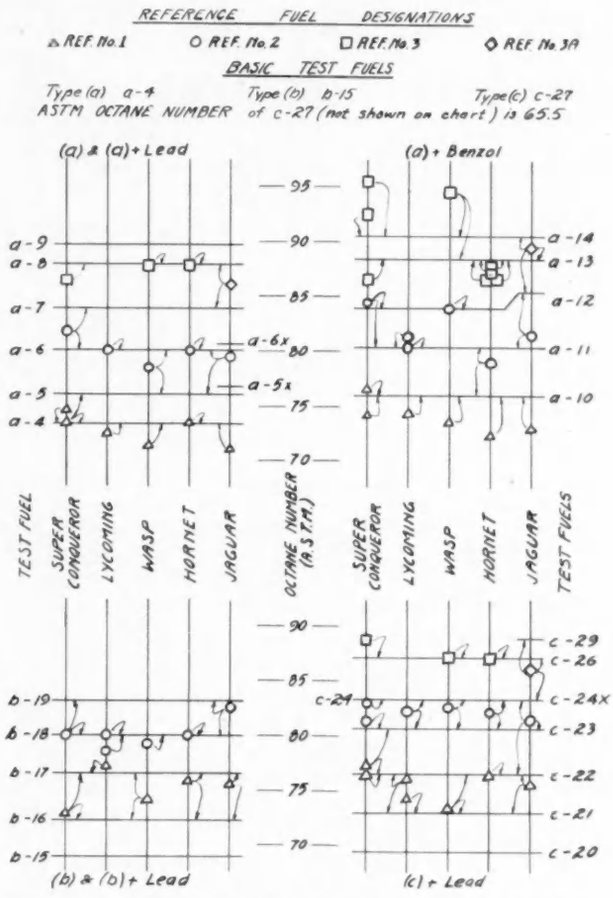


Fig. 2 – Relative Knock-Ratings in Full-Scale Engines for Test Fuels and Reference Fuels

385 b.hp. at 1700 r.p.m. and an altitude of 11,500 ft. when using fuel of 80 octane number. Design features in general follow double-row radial-engine practice as it is now known in this Country. The relation of valves to the cylinder-head and to the crankshaft is substantially similar to single- and double-row radial air-cooled engine practice now in use in this Country.

b. Fuels Used – The fuels used were mostly those with which the tests described in the first report[1] were carried out. Some of the test fuel blends used for these tests were exhausted and were replaced with blends made from the same base stocks. Reference Fuel No. 3 employed in the previous tests was used up and was replaced by Reference Fuel No. 3A, made from a similar base stock having the same octane number. The average A.S.T.M. knock ratings of these replacement fuels were as follows:

Fuel	Octane Number
Reference Fuel No. 3A	86.6
Test Fuel a-5x	76.8
Test Fuel a-6x	80.6
Test Fuel c-24x	83.5
Test Fuel c-29	88.8

Laboratory inspection data for certain of the test fuels are given in Appendix 1.

c. Engine Test Procedure – The test procedure was in general that given on page 163 of the first report[1]. Each reference fuel was, as far as possible, bracketed between two test fuels. A typical comparison of three test fuels with one reference fuel is shown in Fig. 1.

d. Appraisal of Test Data and Inspection of Individual Test Runs – Appraisal and inspection of test data have been carried out by the methods outlined on pages 165 and 166 of the first report[1].

e. Presentation of Results – In the first report graphical presentation of the engine results of all test and reference fuels was condensed into Fig. 7 (page 168) of that report. Other methods of graphical presentation were included in Figs. 8 to 15 of that report, and all methods of presentation were discussed on page 169. For the present report only the graphical method used in Fig. 7 has been followed. Fig. 7 (first report) has been modified to include the Jaguar test data, and is thus shown in Fig. 2.

f. Conclusions – The Jaguar results shown in Fig. 2 are in substantial agreement with those previously obtained on the four other engines. The higher benzol blends, (a) plus benzol, are shown to be somewhat overrated by the A.S.T.M. Method. (This finding agrees with the conclusions drawn in the first report). The cracked fuels, (b) plus lead, also are shown to be somewhat overrated by the A.S.T.M. Method. (This finding is at variance with the conclusion of the first report which indicated that such fuels were rated correctly by the A.S.T.M. Method.)

3. Wright Tests of Olefinic Fuels

a. Engine – The engine used was a nine-cylinder single-row air-cooled radial Wright Cyclone GR-1820-G. This engine has a compression ratio of 6.4 : 1, a blower gear ratio of 8.37 : 1, is nominally rated at 850 b.hp. at 2200 r.p.m. and an altitude of 5800 ft. on 87 octane-number fuel.

[2] Determined with balanced-pressure diaphragm and neon-lamp indicator, designed and fabricated by Wright Aeronautical Corp.

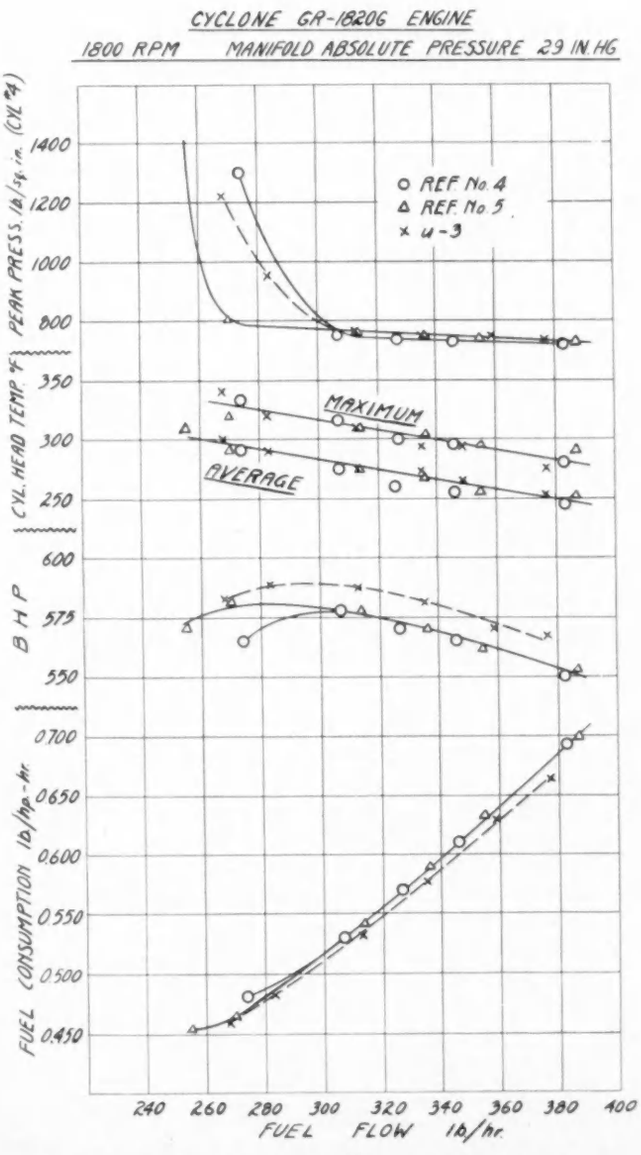


Fig. 3 – Test Data for Vapor-Phase Fuel and Two Reference Fuels

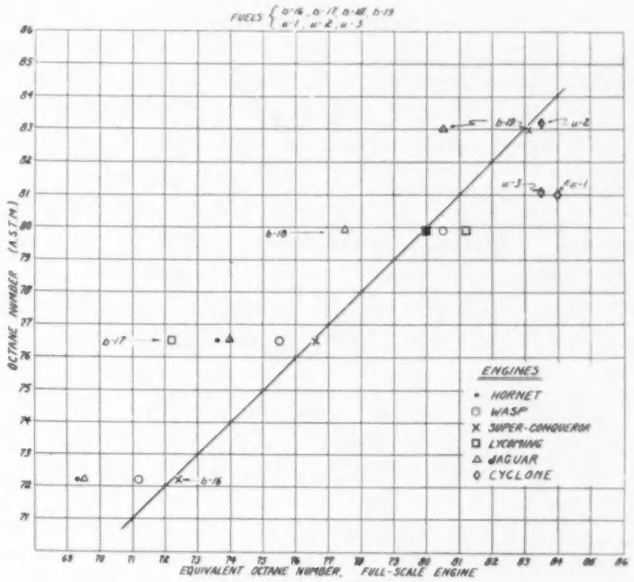


Fig. 4 – Correlation Chart for Olefinic Fuels

b. Fuels used – The test fuels comprised two blends of straight-run gasoline with different proportions of unhydrogenated polymer, and a blend of lead in a base entirely composed of vapor-phase stock. Two straight-run gasolines of about 74 octane number were leaded and used as reference fuels. The average A.S.T.M. knock ratings of the test and reference fuels were as follows:

Fuel	Description	Octane Number
u-1	Polymer No. 1 Test Fuel	81.0
u-2	Polymer No. 2 Test Fuel	83.2
u-3	Leaded Vapor-Phase Test Fuel	81.1
No. 4	Low Reference Fuel	81.0
No. 5	High Reference Fuel	83.8

Laboratory inspection data for the foregoing test and reference fuels are given in Appendix 1.

c. Engine Test Procedure – The test procedure followed that given on page 163 of the first report[1] with the exception that each test fuel was bracketed between two reference fuels rather than each reference fuel being bracketed between two test fuels. In two of the three sets of tests, peak explosion pressures[2] were measured. Fig. 3 shows the test data for the vapor-phase fuel and the two reference fuels.

d. Appraisal of Test Data and Inspection of Individual Test Runs – Appraisal and inspection followed the methods used on pages 165 and 166 of the first report[1] with the addition of consideration of explosion-pressure data where available.

e. Presentation of Results – The relation between the full-scale ratings of these test fuels and their laboratory ratings is shown in Fig. 4. For comparison, the same data for the (b) type fuels from Fig. 10 in the previous report[1] and from Fig. 2 in the present report, also are shown.

f. Conclusions – From Fig. 4 it will be noted that the A.S.T.M. knock-ratings of the three fuels evaluate the full-scale engine "knocking" value of the three fuels within limits of experimental error previously observed for olefinic fuels, which is of interest since the polymer and vapor-phase fuels represent types which are extremely sensitive to engine test conditions.

4. 87 A.S.T.M. and 100 Army Octane Fuels

a. Fuels Used – Both the Wright Aeronautical Corp. and Pratt and Whitney Aircraft have conducted comparative tests of their current supplies of regular commercial 87 A.S.T.M. octane-number fuels containing 2½ to 3 cc. of tetraethyl lead per gal. and 100 Army octane number fuel containing 3 cc. of tetraethyl lead. These test fuels have been compared with unleaded blends of C.F.R. S-1 and M-1 reference fuels, or with S-1 and S-1 plus lead. Average ratings of the two WAC test fuels by the A.S.T.M. Method in terms of unleaded reference fuel blends follow, but average octane numbers have not been reported for any of the four test fuels.

Fuel	Description	Per Cent S-1 in M-1
l-1	Leaded 100 Octane (Army) WAC	99.1
l-2	Leaded 87 Octane (A.S.T.M.) WAC	88.1
l-3	Leaded 100 Octane (Army) P. & W.	.....
l-4	Leaded 87 Octane (A.S.T.M.) P. & W.	.....

The approved reference-fuel specifications and available laboratory inspection data are given in Appendix 1.

b. Engines – Wright Aeronautical Corp. used two nine-cylinder single-row radial Cyclone GR-1820-G engines. One had a compression ratio of 6.7:1 and a blower-gear ratio of 7.00:1, whereas the other was similar to that used for the tests on olefinic fuels (see Section 2 preceding). The ratings of the particular engine models used are (1) 900 b.hp. at 2200 r.p.m. both at sea level and at 6000 ft. altitude on 90-octane-number fuel and (2) 850 b.hp. at 2100 r.p.m. at 8700-ft. altitude on 87-octane-number fuel.

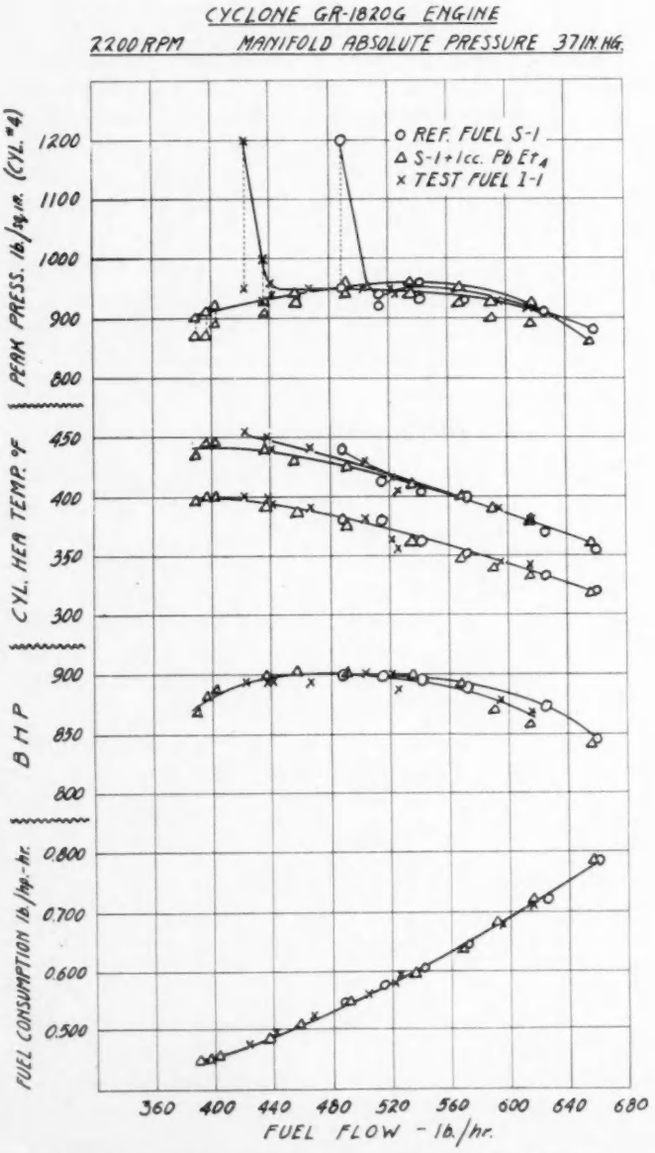


Fig. 5 – Comparison of 100 Army Octane Leaded Fuel with 100 Per Cent C.F.R. S-1 Reference Fuel, and with S-1 Reference Fuel plus 1 Cc. Tetraethyl Lead per Gal.

Pratt and Whitney Aircraft used a double-row Wasp R-1830-C engine having a compression ratio of 6.7:1, a blower-gear ratio of 7.15:1, rated at 900 b.hp. at 2550 r.p.m. at 12,000-ft. altitude with 100-octane fuel.

c. Engine Test Procedure – Engine test procedure was that in Section 2 preceding each test fuel being bracketed by suitable reference fuels.

d. Presentation of Results – A typical comparison of 100 Army octane-number leaded fuel with 100 per cent C.F.R. S-1 reference fuel and with S-1 plus 1 cc. $PbEt_4$ per gal. is shown in Fig. 5. Fig. 6 shows a comparison of 100-octane leaded fuel with 97½ per cent S-1 and 100 per cent S-1. The full-scale relation of the test fuels to the reference fuels is shown in Fig. 7.

e. Conclusions – It will be noted from Fig. 7 that the A.S.T.M. method underrates the 87-octane-number fuels by 6 to 8 octane numbers. The 100 Army octane-number fuels have average A.S.T.M. octane numbers below 99, and their full-scale ratings are in excess of 100. Thus, the A.S.T.M. method is underrating such fuels by 2 to 4 octane numbers. The significance of this result can be seen by reference to the curves for 97½ per cent S-1 and 100 per cent S-1 in Fig. 6, which represent a 2-octane-number difference.

The conclusion in respect to the 87-octane-number fuels is in conflict with the conclusions in the first report. This divergence is evidently due to the fact that the first tests compared the (a) plus lead fuels with reference fuels which consisted of a substantially similar base containing approximately equal lead concentration at each octane number level, whereas the second tests used unleaded reference fuels (substantially similar to iso-octane-heptane blends). The introduction of the unleaded fuels has served to show divergences which otherwise would not have been apparent.

5. Summary and Discussion of All Tests

The three sets of tests discussed in the foregoing show the following:

a. The Canadian tests show satisfactory agreement with the results of the first report.[1]

b. The Wright supplementary tests of olefinic fuels show that the A.S.T.M. method satisfactorily predicts the full-scale value of very highly unsaturated fuels of both leaded and unleaded type. This conclusion is valid for such fuels up to 84 A.S.T.M. octane number. This completes the program on olefinic fuels.

c. The tests of commercial 87 A.S.T.M. octane number and 100 Army octane number (98 to 99 A.S.T.M. octane number) fuels (2½ to 3 cc. $PbEt_4$) show that the A.S.T.M. method underrates 87-octane leaded fuels by 6 to 8 octane numbers and 100-octane leaded fuels by 2 to 4 octane numbers.

The preceding findings, taken in conjunction with the findings given in the first report, would lead to the following tentative conclusions:

d. That unleaded iso-octane blends of 84 A.S.T.M. octane number or more will be lower in full-scale knock-rating than leaded blends of equal A.S.T.M. octane number containing in excess of 2 cc. of tetraethyl lead per gal., thus unleaded blends should be regarded with caution for full-scale service.

e. That benzol blends generally will not be equal in full-scale knock-rating to leaded blends of equal A.S.T.M. octane number when this number is in excess of 84. However, when benzol blends of 84 or more octane number are compared with iso-octane blends of equal A.S.T.M. octane number, they

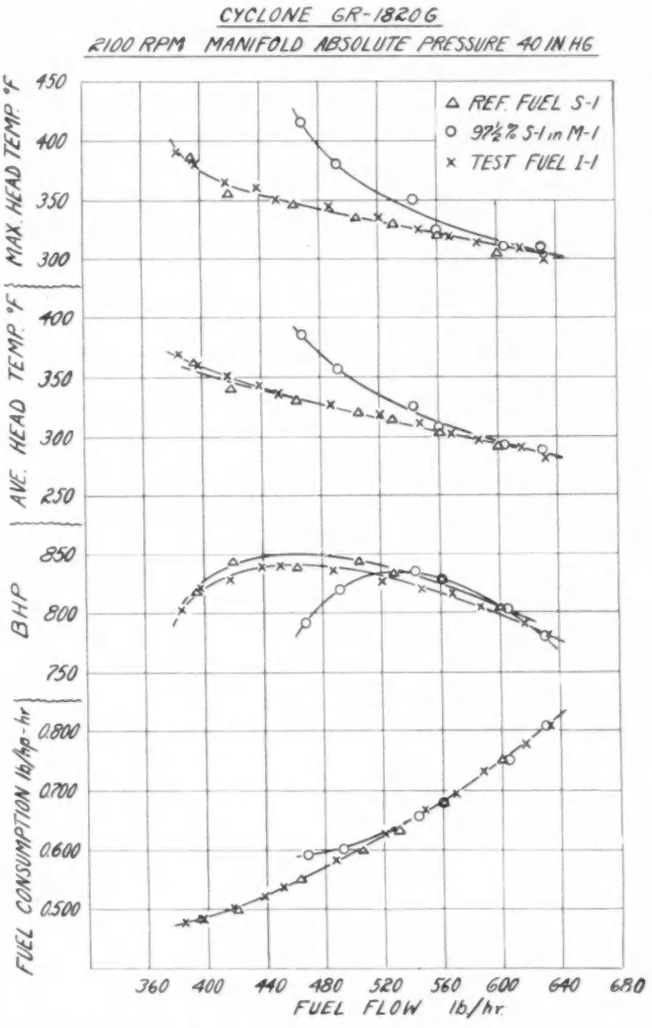


Fig. 6 – Comparison of 100-Octane Leaded Fuel with 97½ Per Cent S-1, and with 100 Per Cent S-1

will, on the average, be higher and rarely lower in full-scale rating. The behavior of high-octane-number benzol blends in full-scale engines is widely variable as a result of engine type and operating conditions. It is apparent, however, that any revised knock-test method should not be more severe on aromatic blends than is the A.S.T.M. method.

f. That the deficiencies of the A.S.T.M. method in respect to leaded fuels should be investigated further by tests of fuels of equal A.S.T.M. octane number and varying lead content.

6. Present Status of Project

In considering Assignments 1, 2, and 3 of the first report (see Foreword), the present status is as follows:

Assignment 1 – The recommended procedure of bracketing each test fuel between two suitable blends of C.F.R. M-1 and S-1 reference fuels has proved to be successful and to be a decided improvement upon the method of bracketing reference fuels between two test fuels. The establishment of high and low octane-number reference fuels, substantially similar to iso-octane and heptane, and which do not need the addition of lead until the 100-octane-number level is reached, has proved to be a decided advantage in determining actual full-scale octane numbers. It has been found that the utility of the method is improved considerably by measurement of cylinder peak explosion pressures.

Assignment 2 – Substantial progress in full-scale tests with commercial fuels of 100 octane number and with reference fuels of over 100 octane number has been made. More data are required in view of the present extensive use of 100-octane-number fuels, which are, however, at present confined to one general type. Before 100-octane-number fuels employing bases of diverse chemical type can be used in service with reasonable safety, extensive full-scale investigation on the dynamometer is obviously essential.

Assignment 3 – Revision of the A.S.T.M. method has been shown to be necessary. A Laboratory Procedure Group has carried out extensive studies and is considering immediate modifications which will give results more in accordance with the data just presented than does the A.S.T.M. method.

7. Future Work

In respect to conclusions 5d, 5e, and 5f it may be mentioned that the Wright Aeronautical Corp. and Pratt & Whitney Aircraft are testing fuels of equal A.S.T.M. octane number and varying lead content. These tests should give data upon which a knock-test method giving suitable values to leaded fuels can be worked out. Although unleaded fuels of 87 or more A.S.T.M. octane number are not in extensive use at present, the problem of a knock-test method with suitable lead response is of considerable importance.

The full-scale rating of aromatics is the subject of much controversy even though such fuels have at present no commercial or military significance in this Country. However, aromatic fuels are regarded with much favor abroad, and American engines at times operate on such fuels. In view of this situation it has been decided that tests on leaded aromatic blends of 95 A.S.T.M. octane number should have the highest priority in tests in the immediate future. Two blends, each containing 4 cc. of tetraethyl lead per gal. and approximately 60 per cent of C. P. aromatics, have been made available for use in full-scale tests.

There is now considerable interest in the possible flight use of fuels of more than 100 octane number. Full-scale data for such fuels are essential for the development of a revised knock test method in accordance with Assignment 1 of the first report (see Foreword). The octane scale has not been defined beyond 100, but tetraethyl lead in C.F.R. S-1 reference fuel is now used as a tentative standard for evaluating the "knocking" characteristics of fuels higher than 100 octane number.

Fig. 8 shows the relative indicated mean effective pressure

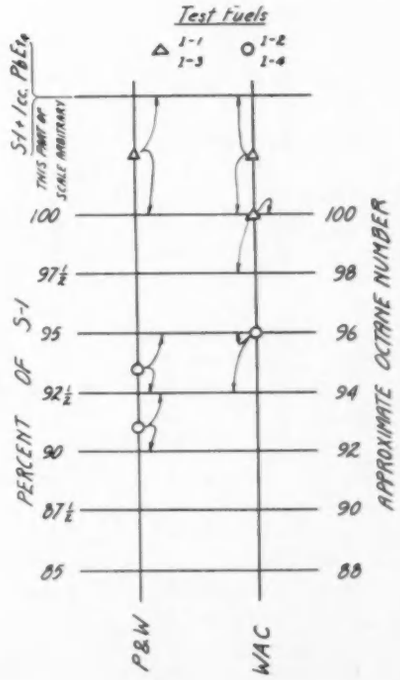


Fig. 7 – Full-Scale Relation of Test and Reference Fuels

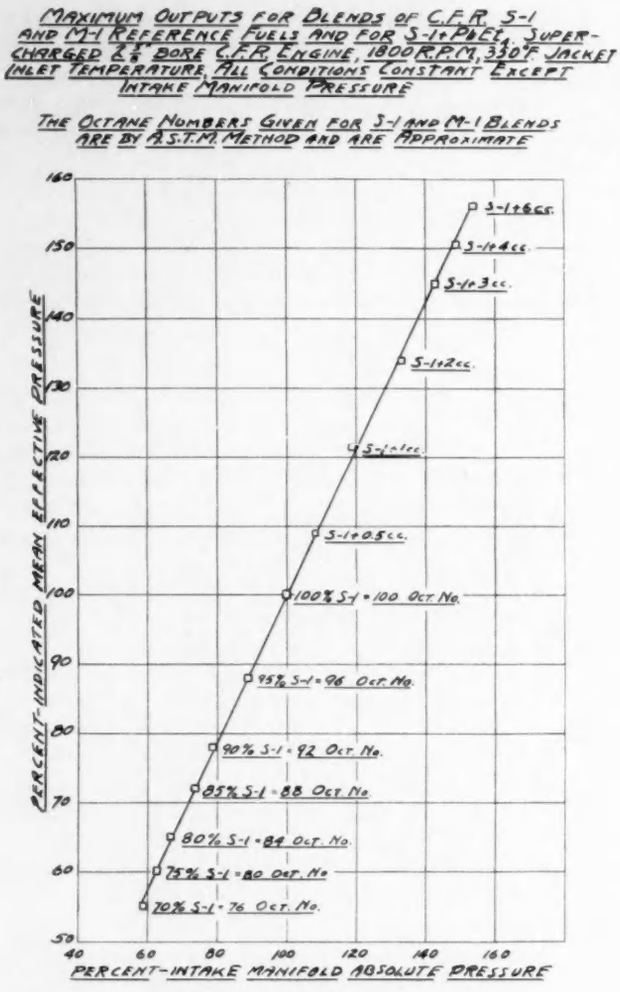


Fig. 8 – Relative Indicated Mean Effective Pressure for S-1 and M-1 Reference-Fuel Blends and for S-1 with Tetraethyl Lead up to 6 Cc. per Gal. in a Supercharged C.F.R. Engine

for S-1 and M-1 reference-fuel blends and for S-1 with tetraethyl lead up to 6 cc. per gal. in a supercharged C.F.R. engine. This figure serves to show the effective size of an octane number over the range of present interest and also shows the effect of lead additions to S-1. The effect of lead additions can be judged in terms of the interval between 95 per cent and 100 per cent S-1. Fig. 8 is, of course, only valid for the particular engine and particular set of test conditions used and may not be related to aviation-engine findings.

Appendix I – Fuel Data

The data presented herein were contributed mainly by members of the C.F.R. Exchange Group, under the direction of D. P. Barnard, and by members of the C.F.R. Aviation Fuels Division working on laboratory detonation projects under the direction of T. B. Rendel. Owing to the fact that the fuel samples were distributed primarily for knock-rating, the available inspection data are less complete than might be desired.

A. Canadian Tests – Inspection data and knock-ratings for all but five of the fuels used by the National Research Council of Canada are given in Appendix A of first report. Inspection data on test fuels c-24x and c-29 and on reference fuel No. 3A are not available, but they may be taken as very similar to the fuels which they replace. The octane numbers given for these three fuels in the main report are averages of ratings by at least six laboratories. Samples of replacement test fuels a-5x and a-6x and check samples of test fuels a-8, a-12, and b-19 were received from Ottawa. The average results of cooperative tests are given in Table 1. No significant change was found in the ratings of test fuels a-8 and b-19, but the rating of test fuel a-12 was found to be 85.3 instead of 83.9 octane number. This change is shown graphically on Fig. 2.

B. Tests of Olefinic Fuels – Data on the test and reference fuels used in the Wright Aeronautical Corp. tests of supplementary olefinic fuels are given in Table 2. Test fuels u-1 and u-2 contained respectively 25 and 50 per cent of unhydrogenated polymer in the same unleaded base gasoline. Test fuel u-3 was prepared by leading a 72-octane vapor-phase base stock. Unfortunately, this test fuel was not subjected to the acid heat test.

C. Tests of Commercial Fuels – Table 3 gives available

Table 1 – Inspection Data, Canadian Test Fuels

	Replacements Test Fuels		Check Samples		
	a-5x	a-6x	a-8	a-12	b-19
Reid vapor pressure, lb. per sq. in.	5.2	5.6	5.4	3.7	4.8
Gravity, deg. A.P.I.	62.1	61.9	61.6	40.0	63.9
A. S. T. M. distillation, deg. fahr.:					
Initial boiling point	119	119	115	147	127
10 per cent point	167	166	166	172	161
50 per cent point	208	211	210	183	195
90 per cent point	261	263	260	216	245
End point	322	319	313	293	314
Recovery, per cent	98.6	98.6	98.5	98.5	99.0
Loss, per cent	0.9	1.0	0.6	0.5	0.6
Octane No. (260 deg. fahr.)*	77.1	80.9	88.5	86.4	84.5
Octane No. (A. S. T. M.)	76.8	80.6	88.0	85.3	83.2

* British Air Ministry Method.

Table 2 – Inspection Data, WAC Test and Reference Fuels

	Olefinic Test Fuels			Reference Fuels	
	u-1	u-2	u-3	No. 4	No. 5
Reid vapor pressure, lb. per sq. in.	5.1	4.2	13.0	6.2	6.4
Gravity, deg. A.P.I.	66.1	65.2	64.1	65.3	65.6
A. S. T. M. distillation, deg. fahr.:					
Initial boiling point	122	128	82	111	115
10 per cent point	162	178	112	147	149
50 per cent point	196	208	209	189	190
90 per cent point	227	229	326	232	230
End point	261	262	361	277	271
Recovery, per cent	98.5	99.0	95.5	99.0	99.0
Loss, per cent	0.8	0.4	2.5	0.8	0.8
Octane No. (A. S. T. M.)	81.0	83.2	81.1	81.0	83.8
Octane No. (Army)	81.7	84.3	31.8	81.9	85.3
Acid heat, deg. fahr. (sulfuric)	46	75	..	2	8

Table 3 – Inspection Data, Commercial Fuels

	WAC Fuels		P.&W. Fuels	
	l-1	*l*-2	*l*-3	*l*-4
Reid vapor pressure, lb. per sq. in.	6.6	6.8	6.6	6.0
Gravity deg. A.P.I.	69.7	65.1	69.5	65.8
A. S. T. M. distillation, deg. fahr.:				
Initial boiling point	98	106	112	116
10 per cent point	143	143	150	148
50 per cent point	195	183	199	191
90 per cent point	221	228	225	236
End point	265	274	276	282
Recovery, per cent	98.1	97.7	98.0	98.0
Loss, per cent	1.1	1.2	1.1	2.0
A. S. T. M. rating				
S-1 in M-1, per cent	99.1	88.1	..	
$PbEt_4$ in S-1, cc.	..	..	0.04	
Army rating				
S-1 in M-1, per cent	..	91.7		
$PbEt_4$ in S-1, cc.	0.03	..		
Acid heat, deg. fahr. (sulfuric)	15	2	6	2

inspection data on the commercial leaded 100-octane (Army) and 87-octane (A.S.T.M.) fuels tested by the Wright Aeronautical Corp. and Pratt & Whitney Aircraft. Only a single 1-gal. sample of the P. & W. 100-octane fuel (l-3) was available for laboratory test, and no sample of the P. & W. 87-octane fuel (l-4). The data shown for fuel l-4 are taken from the vendor's records. Three out of four laboratories agreed that fuel l-3 was superior to iso-octane by the A.S.T.M. Method.

The approved specifications for the C.F.R. reference fuels S-1 and M-1 are as follows:

Specification for High-Octane Reference Fuel (Technical Iso-Octane) of the Cooperative Fuel Research Committee

I. The material covered by this specification is intended as a reference fuel (designated as reference fuel S) for use in fuel tests in full-scale aircraft engines.

II. Detailed Requirements

1. Specific Gravity – d $^{20}/_{4}$ shall be 0.6922 to 0.6926.

2. Distillation Range – Distillation conducted in accordance with A.S.T.M. specification D86-35 shall be as follows:

Initial, 204-208 deg. fahr.
10 per cent evaporated, 206-210 deg. fahr.
90 per cent evaporated, 207-211 deg. fahr.
End point, 223-233 deg. fahr.
Recovery, 98.5 per cent minimum.
Residue, 1 per cent maximum.

3. Unsaturation – Unsaturation shall be determined by bromine number in accordance with the method of A. W. Francis.[3] The amount of bromine absorbed shall be not greater than 4.0 mg. per gram of fuel. (This amount corresponds to 0.3 per cent maximum unsaturation as di-isobutylene.)

4. Gum – The fuel shall be subjected to an accelerated aging test with oxygen as specified in U. S. Army Specification 2-92. After this test the gum shall not exceed 3 mg.

5. Sulfur – Sulfur shall not exceed 0.015 per cent as determined by A.S.T.M. Method D-90-34T.

6. 2,2,4-Trimethylpentane Content – The material will be rejected unless it has a definite freezing point within the range —108.00 to —108.20 deg. cent. as determined by the National Bureau of Standards.

7. Absence of Antiknock Compounds – Antiknock compounds of the metallic, aromatic, amine or other type shall not be present.

8. Octane Number –

a. Octane number of the material as delivered is not specified as it is considered that this value is of only minor importance in view of the antiknock requirements specified below for the material when blended with tetraethyl lead.

b. The material, when blended with 6 cc. of tetraethyl lead per U. S. gal. shall, as a minimum, be equal to certified octane plus 5 cc. of tetraethyl lead per U. S. gal. when tested by the Air Corps Method (U. S. Army 2-94 Knock Test, Fuel, Aircraft Engine). In conducting the knock test the Air Corps Method shall be modified by increasing the knock intensity to such an extent that the difference between 5 and 6 cc. tetraethyl lead per U. S. gal. in the material meeting this specification can be detected definitely and repeatedly. The lead response value shall be the subject of at least two determinations which shall be conducted on a minimum of two engines.

[3] See *Industrial and Engineering Chemistry*, Vol. 18, 1926, pp. 821, 822, and 1095: "Estimation of the Unsaturated Content of Petroleum Products," by A. W. Francis.

Table 4 – Distillation Data, Reference Fuels

	S-1	S-1 in M-1 95 per cent	S-1 in M-1 90 per cent
A. S. T. M. distillation, deg. fahr.:			
Initial boiling point	205	189	198
10 per cent point	207	203	205
50 per cent point	208	210	210
90 per cent point	208	228	237
End point	234	361	360
Recovery, per cent	98.4	98.8	98.4
Loss, per cent	0.6	0.4	0.4
Barometer, mm. hg.	740.5	740.5	740.5

Specification for Low-Octane Reference Fuel (Michigan Gasoline) of the Cooperative Fuel Research Committee

I. The material covered by this specification is intended as a reference fuel (designated as reference fuel M) for use in fuel tests in full-scale aircraft engines.

II. Detailed Requirements

1. Gravity – A.S.T.M. Method D287-33 (Federal Specification Board Method 40.1). A.P.I. gravity shall be 61.3 to 61.7 deg.

2. Distillation Range – A.S.T.M. Method D86-35 (Federal Specification Board Method 100.14). Distillation range shall be as follows:

Initial	115 deg. fahr. ± 15 deg. fahr.
10 per cent evaporated	171 deg. fahr. ± 5 deg. fahr.
20 per cent evaporated	206 deg. fahr. ± 5 deg. fahr.
30 per cent evaporated	235 deg. fahr. ± 5 deg. fahr.
40 per cent evaporated	263 deg. fahr. ± 5 deg. fahr.
50 per cent evaporated	290 deg. fahr. ± 5 deg. fahr.
60 per cent evaporated	312 deg. fahr. ± 5 deg. fahr.
70 per cent evaporated	327 deg. fahr. ± 5 deg. tahr.
80 per cent evaporated	341 deg. fahr. ± 5 deg. fahr.
90 per cent evaporated	358 deg. fahr. ± 5 deg. fahr.
End point	392 deg. fahr. ± 10 deg. fahr.

3. Acid Heat – Acid heat shall be determined as specified in Navy Department Specification M302, Section F-2(g), and shall be between 2 deg. fahr. and 6 deg. fahr.

4. Gum – A.S.T.M. Method D381-34T (Federal Specification Board Method 330.2). Gum shall not exceed 3 mg.

5. Sulfur – A.S.T.M. Method D90-34T (Federal Specification Board Method 520.12). Sulfur shall not exceed 0.015 per cent.

6. Reid Vapor Pressure – A.S.T.M. Method D323-32T (Federal Specification Board Method 120.1). Vapor pressure at 100 deg. fahr. shall be between 4.8 and 5.2 lb. per sq. in.

7. Octane Number –

a. A.S.T.M. Method D357-34T (Federal Specification Board Method 600.11). Octane number shall be between 18 and 20. The certified grades of iso-octane and heptane shall be used for determinations of octane number.

b. U. S. Army Specification 2-94, knock test, fuel aircraft engine. Octane number shall be between 12 and 14. The certified grades of iso-octane and heptane shall be used for determinations of octane number.

III. Inspection – Three determinations of each of the detailed requirements in Section II shall be made, if possible, by three different laboratories. In the case of distillation range the three determinations at each required point shall be averaged and compared with the requirement.

As a matter of interest, Table 4 gives distillation data by a single laboratory on reference fuel S-1 and on two of the reference-fuel blends used in the present tests.

Deflection Tests of Axles and Transmissions

By E. G. Boden

Experimental Engineer, The Timken Roller Bearing Co.

A DESCRIPTION of a testing machine for checking the deflections in rear axles and transmissions is presented in this paper. The technique of making a test is explained, and the method of obtaining and recording the data is outlined. Some important weaknesses as shown up in the tests are discussed, and suggestions are made for correcting them.

The author concludes that, in order to obtain maximum fatigue life, such gears must be mounted most rigidly. He points out, however, that this ideal is not accomplished in commercial installations, but is modified by economic considerations.

IN order to obtain maximum fatigue life from rear-axle and transmission gears, when such are considered to be as near commercially perfect as possible, it is essential that they be mounted on the most rigid bearings and installed on rigid shafts and housings. This ideal is not accomplished in commercial installations, being modified by economic considerations. Axles and transmissions designed for automotive service, including tractor applications, should give a reasonably definite number of hours of service with a minimum amount of replacement, depending upon the type of consumer demand being catered to. The axle of a low-priced car would be expected to have, say, only 30 per cent to 50 per cent of the life of that of a high-priced car before any service is required. This same reasoning will apply to tractors, trucks, and other equipment. In most instances it is more economical to renew parts of the machine at frequent intervals than to design such parts to have a fatigue life equal to that of the whole machine.

To determine the probable expected life of axles and transmissions, the rigidity of the gear mounting as a whole should be known and the respective deflections segregated. When such data are available, it becomes relatively easy to determine if housings should be stiffened, if the bearings should have more capacity, or if the rigidity of the gear shafts should be increased. When such facts are known, the guess-work as to the cause of failure or inefficient operation can be reduced largely if not eliminated. About 12 years ago our company decided to determine these relative factors and a testing machine was obtained, which since has been modified, and is shown in Fig. 1. The machine is similar to the one originally developed by the Gleason Works, the capacity of the present machine having been doubled. Since the time of installation hundreds of tests have been conducted for our customers with exceedingly beneficial results both to them and to ourselves.

The machine is driven by a 15-hp., 1200 r.p.m., induction motor, driving through two Brown & Lipe seven-speed transmissions, providing a reduction ratio of approximately 100:1. The axle to be tested is placed on pedestals at the spring seat and bolted to the machine, which, in turn, is bolted to the floor. The regular axle wheels are replaced by gears of 14 in. pitch diameter, 4-in. face, which mesh with similar gears on a solid cross-shaft of 3-in. diameter, supported by pedestals bolted to the machine base. On each end of the cross-shaft a

[This paper was presented at the National Tractor Meeting of the Society, April 14, 1938.]

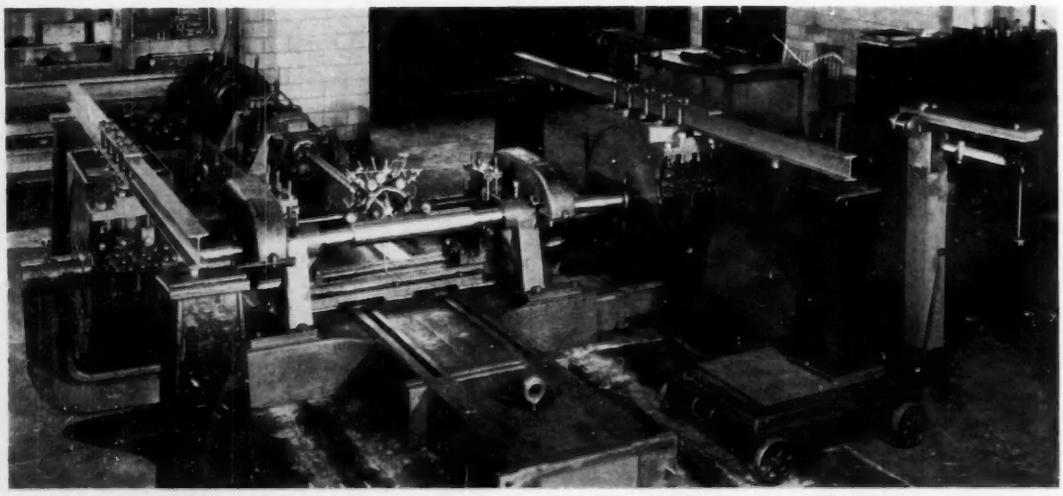


Fig. 1 – Deflection Test Machine

30-in. diameter brake drum is mounted, 13 in. wide, on which a prony brake is arranged, having steel bands equipped with brake lining. A beam 10 ft. long enables torque absorbed by the prony brake to be measured in either direction of rotation, thus giving a leverage of 5 ft. between the drum center and the weighing scales. The axle-shaft torque, therefore, can be absorbed on either or both drums up to the capacity of the brake bands. A torque of 10,000 ft-lb. can be carried on each drum, and the input torque in the transmissions is a maximum of 3,000 ft-lb. for safe transmission tooth loading.

In order to analyze the relative deflections of the various components making up an axle or transmission, it has been found necessary to locate the support carrying all indicators from a section of the axle which, in itself, is more or less rigid. This point of application of the brackets carrying the indicators is located most conveniently between the two pinion bearings. All tests made by our company have been located from this point, and the arrangement of front dial indicators measuring the relative deflections is shown in Fig. 2. A yoke is secured to the pinion housing by means of a number of staggered set-screws. A bracket is carried from this yoke around the rear of the axle, from which the various indicators are mounted. Fig. 3 shows the typical data of deflections and tooth contacts which are recorded. The usual routine test consists of checking the following deflections, which are all in relation to the pinion bearing housing as shown in Fig. 3:

Pinion End Movement – On the end of the pinion. Indicator No. 1.
Pinion Lift – On top of the inner bearing cone rib and at the small end of the pinion teeth. (By means of a button fitted in the end of the pinion.) Indicators Nos. 2 & 2*A*.
Pinion Side Movement – On the side of the inner bearing cone rib and at the small end of the pinion teeth. Indicators Nos. 3 & 3*A*.
Ring-Gear End Movement – Ring-gear movement at pinion mesh and at 180 deg. from the pinion mesh. Indicators Nos. 4 & 5.
Ring-Gear Lift – Movement on top of ring-gear. Indicator No. 10.
Ring-Gear Fore and Aft Movement – Movement of ring-gear in line with pinion axis. Indicator No. 11.
Differential pedestal and cap movements both vertically and on ring-gear axis.
Pinion-housing deflection in relation to carrier flange.

If these deflections are not within normal limits, other deflections are checked until a clear picture of the general movements is obtained. One object in making these tests is to determine what effect the change in axle torque produces on the tooth contacts. In order to obtain both maximum fatigue resistance and quietness of operation, the tooth contacts must not depart from conditions which provide for the lowest tooth stresses and sufficient tooth overlap to insure as near uniform transfer of torque from one tooth to another. The tooth contacts under "no load" are obtained first by painting the teeth with powdered red-lead and oil, and the indicators are all set to zero. Four sets of indicator readings are usually taken at 25 per cent, 50 per cent, 75 per cent, and 100 per cent of the maximum torque in low gear in both forward and reverse. This loading is done by tightening the prony brake bands and weighing the torque reaction on standard platform scales. After the maximum torque has been applied, the load is removed and the deflections and tooth contacts again noted in order to see if any permanent set has occurred due to some particular weakness. If such is found to be the case, investigations are made to determine the cause, and recommendations are made to improve the axle so as to bring the deflections within limits found to be satisfactory in practice.

The recommendations made by the Gleason Works for maximum gear movements are shown in Fig. 4. If movements can be controlled within these limits, successful operation of the gears is assured, providing the gears are made to accepted standards of manufacture.

The various types and arrangements of antifriction bearings made to support the ring-gear and pinion in general have characteristics of their own which determine the type and arrangement for any particular application. Space limitations and bearing sizes which control, in many instances, the economic design of an axle, influence the choice of bearings in most instances.

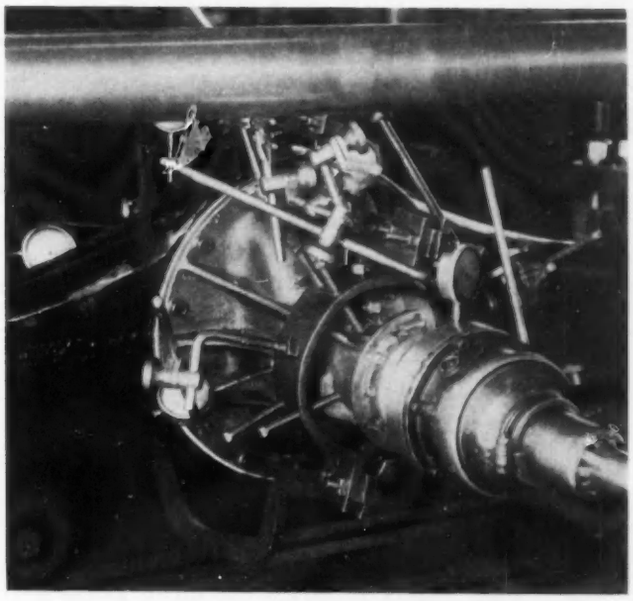


Fig. 2 – Arrangement of Front Dial Indicators

Fig. 5 shows the set-up for checking the deflections on a combined transmission and axle for a tractor. In this case the torque was taken through a chain drive from the sprocket wheel on the axle final drive assembly. A sprocket wheel was fixed to the brake drum cross-shaft of the test machine, which replaced the gear used when testing a conventional axle, and the torque absorbed on both the prony brakes. In this application, the pinion shaft forms the bottom shaft of the transmission. The ring-gear shaft is connected to the chain-gear sprocket shaft through a final spur-gear drive. A total of 75 separate indicator readings was taken, 18 being on the transmission and pinion shaft, 14 on the ring-gear shaft assembly, 28 on the final drive assembly, and 14 on the transmission and axle housing. Because the ring-gear shaft contained two plate clutches, one connecting each final drive assembly, which were used for steering purposes, the total torque was taken through one final drive which would correspond to the worst possible conditions which could be met in service. In this case all indicators were mounted from a bar on the top of the transmission case at the rear as shown, this being considered as the stiffest part of the assembly, so that all deflections are relative to this point. The test was conducted in low gear, in third gear, and in reverse, being the worst conditions for deflections in the transmission.

In tests of a combined transmission and axle for passenger cars, we found that the torque when in second and low gear of the transmission produced a twisting of the transmission case, which twisting naturally is detrimental to its operation. It is necessary, therefore, that the case should be stiffened to resist this twisting action if trouble from jumping-out-of-gear and early fatigue failure is to be eliminated.

By far the greatest number of tests that we have conducted have been on passenger-car and truck axles. In early designs this same twisting action created by the torque on the pinion and ring-gear teeth was responsible for some of the early troubles experienced.

A common feature of many axles under reverse conditions is the sidewise movement of the pinion housing in relation

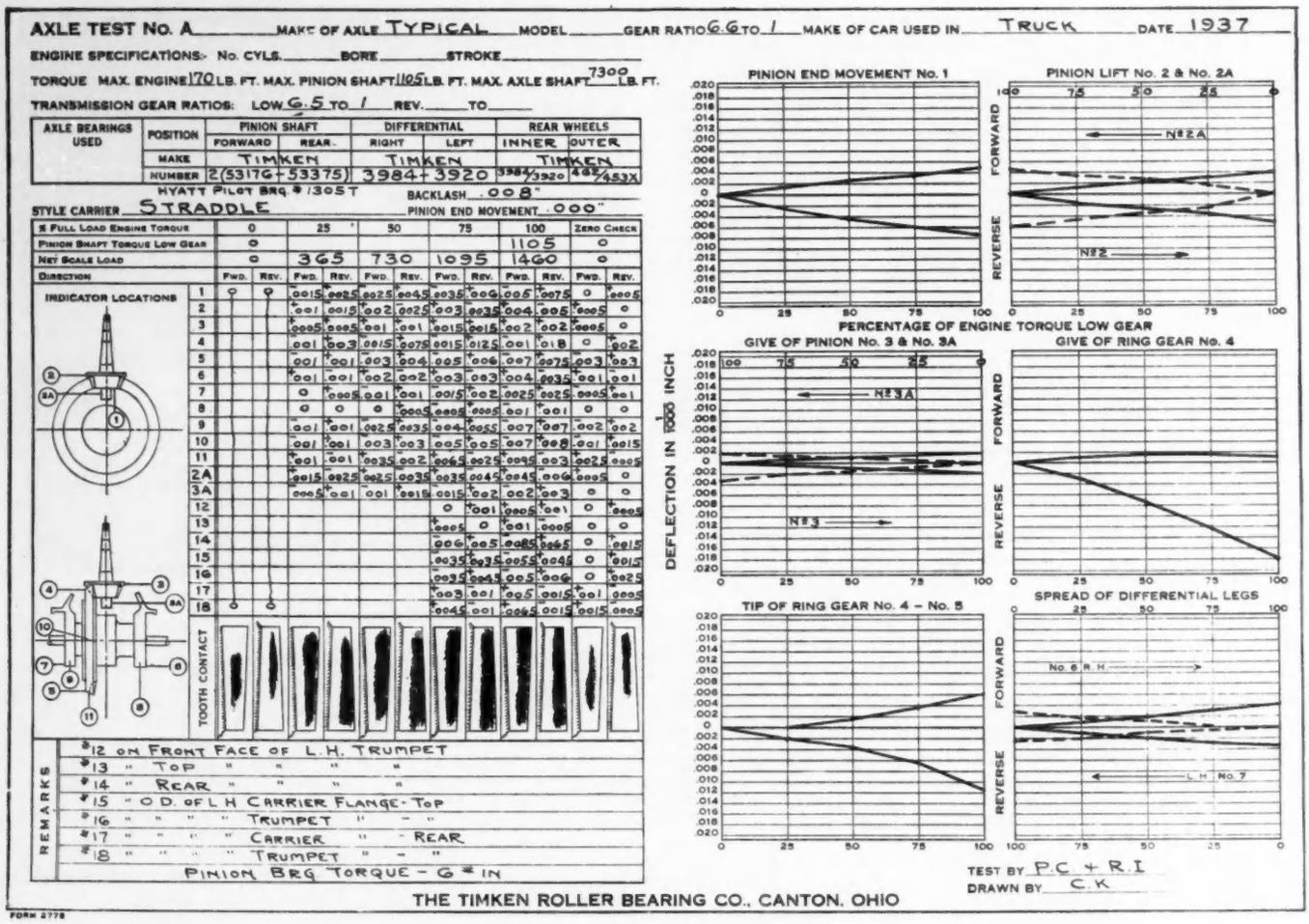


Fig. 3 – Typical Data of Deflections and Tooth Contacts

to the left-hand pedestal leg and carrier flange. Fortunately the loads encountered under these conditions, and under coasting against the motor, are of short duration in the majority of instances. Where abnormally long periods of operation under these conditions are experienced, gear trouble is likely to be encountered. In passenger cars, especially with hypoid gears where pressure angles on the coast side are around 25 deg., the utmost rigidity should be obtained because of the inherent tendency for gear scuffing due to the increased tooth sliding under coast and reverse conditions. Although it is necessary even with the best hypoid gears today to use special E.P. lubricants, the influence of excessive deflections increases the necessity for further protection from this source, which should be taken care of by design and not by lubrication.

A second weakness found on many axles for both passenger cars and commercial vehicles is the deflection of the left-hand pedestal leg, due to the separating force between the ring-gear and pinion. It has been found effective to provide substantial horizontal ribbing between the base of the pedestal legs and the pinion housing, and stiffening of the pedestal legs to the carrier in order to avoid excessive leg spread. Although the carrier flange itself is simply a means of holding the carrier to the banjo, or axle housing, it is found desirable in many instances to make this flange stiff enough to resist the external loads produced by axle-housing deflections from interfering with the rigidity of the carrier itself. It must be borne in mind that we are interested in maintaining the true relation between the pinion bore and the differential bore of the carrier. If sufficient rigidity cannot be built into the carrier itself, assistance from the axle housing can be obtained only when this member is sufficiently rigid to act as a stiffening to the pedestal legs. Early attempts to brace the differential legs against press-steel banjos were a failure because of the inherent weakness of the banjo itself to maintain its shape under combined gear and external loads, both spring loading and the components of road wheel torque. In the case of cast housings without coverplates, the use of large dowels, particularly on the left-hand pedestal, is effective.

Fig. 6 shows the attempts made to stiffen the nose of the pinion and the differential legs by means of ribs extending from the carrier flange to the forward end of the pinion housing. In order to improve the rigidity of the pedestal legs,

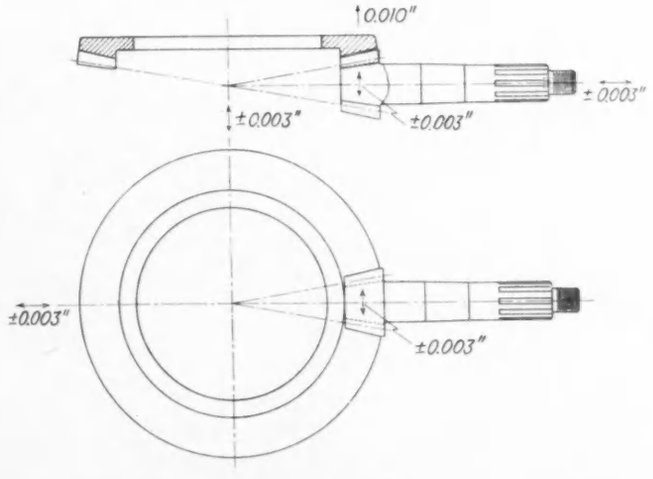


Fig. 4 – Recommendations for Maximum Gear Movements Made by the Gleason Works

they are carried into the main body of the carrier as well as can be done considering the space requirement for the ring-gear. In this connection some axles have external horizontal ribs approximately in line with the top of the pedestal legs, as shown at *A*. The loads on the right-hand differential pedestal are not so high as on the left one, particularly when the ring-gear is very close to the left-hand differential bearing, which is usually the case when the centerline of the pinion falls on the centerline of the axle. In order to reduce the bearing loading and provide sufficient stiffness for the ring-gear flange, many axles now have the pinion offset to the right to provide more space for ring-gear flange stiffness and the reduction of the left-hand bearing radial loads. This construction improves the general ring-gear deflections, and is to be recommended whenever advantage can be taken of this arrangement. Usually the vertical rigidity of the pinion housing is quite satisfactory due to pinion oil passages acting as reinforcing members, but occasionally, particularly in truck and bus applications, vertical webs are required between the pinion housing and the carrier flange to keep the relative movement between the pinion housing and the ring-gear from becoming excessive.

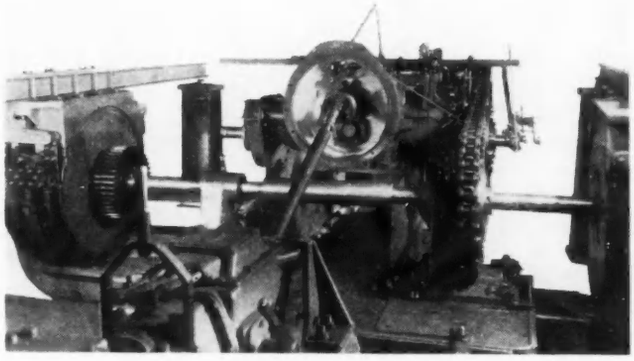


Fig. 5 – Set-Up for Checking the Deflections on a Combined Transmission and Axle for a Tractor

If the differential supports are not stiff enough in a vertical direction, the relative displacement between the pinion and the ring-gear in this direction will produce heavy heel bearing tooth contacts, which are productive of noise and ultimate tooth breakage.

Besides the endwise ring-gear movement and the lift, a tipping action of the gear is found due to leg spread and general differential case flexibility. For this reason it is a common practice in heavy-duty applications to provide a ring-gear stop to limit the movement of the ring gear at the point of gear tooth mesh, particularly on reverse and coast loads. This stop is usually a bronze pad or button, and sometimes it is provided with a screw adjustable feature to maintain a minimum clearance due to inevitable wear. This is not a desirable feature, but is resorted to due to the difficulty in economically providing sufficient rigidity in the differential case and bearing supports to withstand the tremendous torque moment acting on the ring-gear.

With housings of substantial rigidity, pads on the differential pedestal caps are sometimes used which are machined to fit into the housing. These pads are usually straight faces which fit into machined faces in the axle housing, providing a bracing effect in a horizontal direction, as shown at *B* in Fig. 6. Some axles have a pilot turned on the pedestal caps to fit a machined diameter in the axle housing as shown at *C*. This pilot provides stiffness in the horizontal and to some extent in the vertical direction. To obtain the maximum amount of improvement in both vertical and horizontal direc-

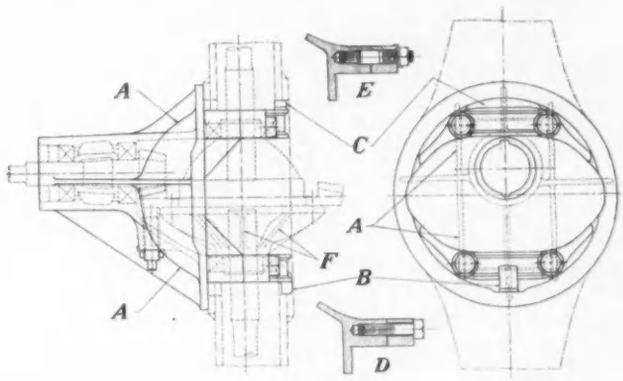


Fig. 6 – Typical Carrier

tions, the fitting of large-size dowels into the pedestal caps – with a cast housing without inspection coverplate – has proved very effective in reducing ring-gear movements.

As the ring-gear radial load is taken on the differential pedestal caps for the majority of its life, care must be taken to see that the caps are bolted securely to the pedestal legs, giving sufficient rigidity so as not to increase ring-gear movements from this source. The usual method of attaching the pedestal caps to the pedestal is shown at *D* in Fig. 6, the cap bolt hole having a clearance of 1/64 to 1/32 in. on the diam-

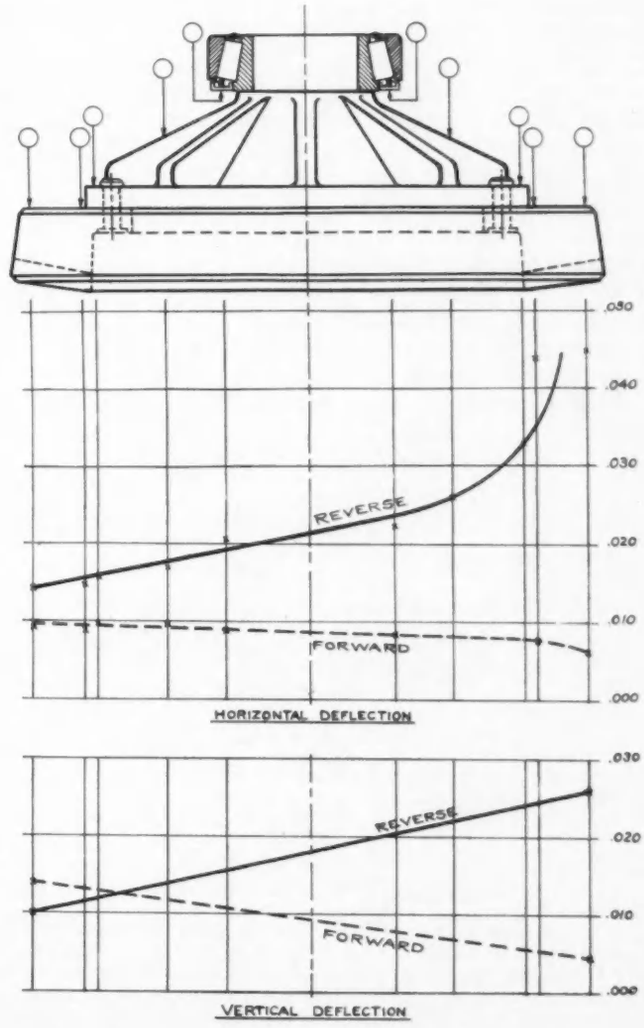


Fig. 7 – Horizontal and Vertical Deflections

eter. Better rigidity would be assured by using a dowel-type stud as shown at *E*. In most cases the bearing backing for the differential is by means of a threaded ring in a split pedestal. If these threads are not a good fit, the ring-gear movement from the combination of thread fit and bearing-cap bolt-hole clearance will be increased. It is obviously necessary that the rigidity of the differential pedestal cap in relation to the pedestal should be a maximum in order to maintain minimum ring-gear movements.

Because of the difficulty in holding the ring-gear movement to reasonable limits under maximum loads, recourse is made to preloading of the differential assembly by spreading the pedestal legs, thus putting an initial compression in the bearings. This method is effective in reducing the movements slightly but, due to the displacement of the differential bores, serious results are caused by excessively spreading the pedestal legs. It is obvious that the antifriction bearings are working at a great disadvantage if the housing bores are thrown out of alignment, and this practice of spreading the pedestal legs by preloading is limited by the stiffness of the pedestal assembly. The experience obtained by the bearing manufacturers should be taken advantage of, and their recommendations of maximum pedestal-leg spread for each individual axle followed.

The rigidity of the ring-gear mounting to the differential case is another point which has shown weakness in the past. The best practice is to back up the ring-gear with a flange well ribbed to the differential bearing hub as shown at *F* in Fig. 6. The deflection of the ring-gear at the point of pinion mesh has been improved considerably in many axles in the past few years by this construction, and it is highly recommended.

Fig. 7 shows the rapid increase in deflection at the point where the ring-gear is fastened to the differential case flange. A construction which is becoming popular is to extend the differential case flange to back up the ring-gear, eliminating the relatively thin ring-gear flange, which ordinarily either is riveted or bolted to the differential case. The back-face of the ring-gear is tapped, and the ring-gear is held to the differential case by cap screws. This construction is satisfactory for small gears, but is not yet popular on large gears for trucks due to difficulties in lining up the cap-screw holes. Distortion in hardening the large gears interferes with the use of close-fitting cap-screw holes.

With a tendency in heavy-duty axle construction to obtain low ratios with single reductions, considerable trouble has been experienced when attempts have been made to make reductions more than approximately 6:1. In attempting to limit the ring-gear size, the diameter of the pinion and the number of teeth become quite small with consequent high tooth loading. The diameter of the shaft also is governed by the pinion size and, in many cases, the shaft is weakened still further by the tooth-cutters removing metal at the bearing supports. Even though the bearings are adequate for the imposed loads, the deflection of the shaft itself, especially on the overhung mountings, becomes excessive, and no bearing can compensate for lack of pinion rigidity.

In some cases, where the forward pinion bearing is butted against a shoulder formed by reducing the forward end of the pinion-shaft diameter, trouble has been experienced due to the weakness caused by the reduction in stiffness of the shaft by this design. Lack of pinion stiffness results in concentration of the load on the large end of the teeth, due to the pinion deflecting away from the ring-gear teeth.

An example of lack of pinion stiffness is shown in Fig. 8, where an analysis is made of the relative deflections due to the pinion shaft and also the pinion bearings. The actual combined deflections found in test are also shown. Considering that the calculated values are for static load conditions, the check between the two sets of figures is considered to be quite good. It is seen that, although the stiffness of the pinion bearings is very important, the stiffness of the pinion shaft itself is equally important, and increasing the stiffness of the bearings above a critical point without increase in pinion-shaft stiffness would not result in decreased pinion-head movements. The limitations of straddle-mounted pinions is also due to lack of stiffness in the pilot of the pinion shaft, which is necessarily small, caused by the tooth-cutter clearance required. The space limitations for installing a suitable bearing also govern the maximum possible rigidity that can be obtained by such a mounting.

The use of hypoid gears offers a means of increasing the

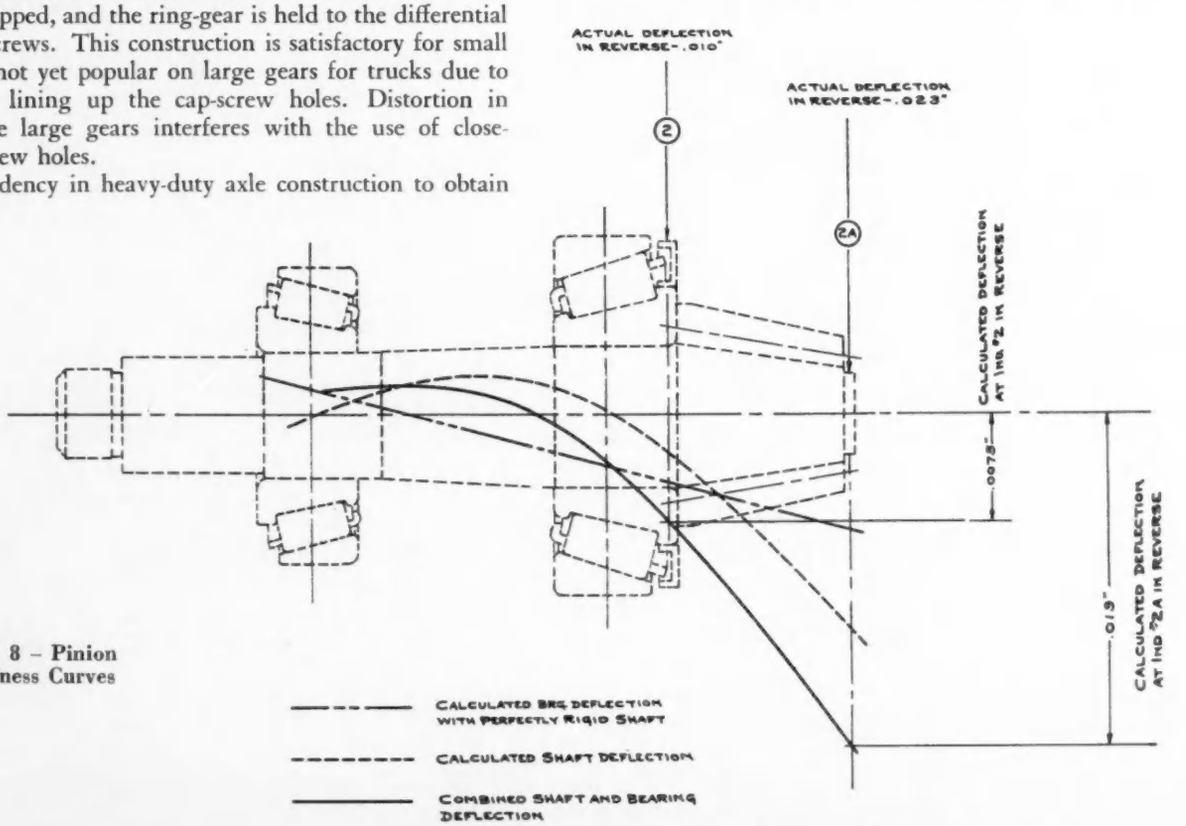


Fig. 8 – Pinion Stiffness Curves

pinion-shaft stiffness because, for a given ratio and ring-gear size, the pinion is increased considerably in diameter with consequent increase in pinion-shaft size. It would appear that a good solution to the lower ratios for single-reduction axles would lie in the use of hypoid gears because of the increased pinion stiffness and increased fatigue resistance of the pinion teeth. Adequate bearing sizes, however, should be used because of the increased bearing loads.

If attention is paid to the weak points in axle design as brought to light by the results of deflection tests as described, considerable improvement in fatigue life of axles will result.

Discussion

Points Out Importance of Stress Concentration

—L. Jacobi

Allis-Chalmers Mfg. Co.

IN the first paragraph of the paper under discussion, we find the statement that we do not obtain maximum fatigue life of rear axle and transmission gears in commercial installations due to economic considerations. Further, that it is more economical to renew parts than to design them so that they have a fatigue life equal to that of the whole machine. This logic is sound and has its place.

To tractor transmissions in general, however, we hardly can apply it. If a gear goes out, or a shaft, it is usually disastrous to the whole transmission.

We may allow an occasional failure of a part as long as it remains the exception. We must govern our design so that a machine has given economical service in its life. The duration of such life should not be difficult to determine. At the end of such a period, the transmission should be worn out without having any broken parts.

We may obtain maximum fatigue life on gears when we apply the term to the tooth surface but, when it comes to shafts, I believe it is a misnomer. We have not reached maximum fatigue life as long as a shaft has not broken, and we cannot have it as long as we do not wish this to happen during the life of the machine.

Deflection tests of a transmission as outlined certainly should be of much value, but such dynamic deflection tests, even under full torque, will not tell the whole story. A correct picture we will have only when we can measure the deflection at full torque under the normal r.p.m. This is quite an order. Until such equipment is available, we will have to do the best with the means at our command. By judicious design a number of things may be done that will tell us beforehand how the transmission will turn out.

In a final analysis deflection governs. If we calculate the deflection carefully, we lay good ground work and come close to actual deflections. With involved design of transmission cases, we will not be so successful with our calculations, especially when we try to build them light. Then, even a static load test will be of great help to find the weak spots. If we adhere to such fundamental rules as minimum bearing spread, overhung loads close to bearings, gears placed so as to cause the least deflection, bearings adequate and supported solidly, avoiding flat walls and making certain that in the case of cast-iron transmission cases the fibers under tension will be favored, then we will not have to be concerned very much about the outcome.

Better materials, unfortunately, will not give us decreased deflection but, by employing them, we can often condense the design and achieve our purpose indirectly.

There are, of course, cases when means may be employed to neutralize the effects of deflection successfully, for instance, the ring-gear stops mentioned by the author, or by making corrections in the tooth form. When such means are worked out correctly, they will not be makeshifts but must be termed excellent engineering. An interesting example is the piston-rod as we make them for our large Allis-Chalmers gas engines. One piston is 60 in. in diameter, weighing approximately 6000 lb. and is supported by a piston-rod 25 ft. long, weighing about 18,000 lb. This rod is supported on both ends, the piston floats. The deflection is calculated, then each end of the rod from the piston out is finished from a fulcrum off center, the distance of which equals calculated deflections. When piston and rod are in place, the deflection will straighten the rod to a horizontal position. This construction results in effective working stuffing boxes.

One important point that does not receive the attention it should is stress concentration. Little real effort is made to avoid its causes, especially in mass production. In consequence, when a part breaks due to such a cause, we call it a fatigue break but, in reality, it is plainly a dose of overload under such conditions. The part likely would have endured had stress concentration been avoided. We know the bad effect of it but, until we avoid it, we will not be able to design most economically. Surface and flexional endurance limits lose their meaning when they cannot be utilized through premature failure of the action of stress concentration.

Contributes Experience with Transmission Problems

—L. A. Bixby

Clark Equipment Co.

IN general, Mr. Boden has not discussed a great deal regarding transmission problems. However, inasmuch as I am concerned primarily with the transmission, my remarks will be pointed to that particular unit.

There is one basic law that we all recognize, that is, that deflections exist whenever we apply load to any piece of mechanism. Therefore, good engineering dictates that the design shall be such as to limit this deflection to an amount that is satisfactory for the service in which the unit is to be subjected. We should bear in mind that the unit must meet a commercial need. Therefore, we have to determine the point in any design to which it is permissible to go in order to eliminate deflections and still keep the product so that it is commercial and falls within the point of giving satisfactory service.

In connection with transmissions, the proper design of the case can be a great aid in limiting deflections and holding them to a minimum. This statement obviously does not necessarily mean that we must put excessive weight in the job but, after having recognized the lines of force causing the deflection, we can distribute the metal properly so as to hold the deflections to a minimum.

It is well at this point also to mention that care should be exercised in a design to provide a uniformity of deflection in the case and not allow a portion to have great rigidity and then permit some other section to deflect excessively.

There are many deflections that occur within a unit which we can not measure or calculate. The gear itself, its webs and hubs, distort a certain amount, as well as torsional deflection that occurs in shafting, all of which can be accounted as a contributing element in the life expectancy of the unit.

In transmission constructions effort should be exercised to keep the design as short and compact as possible.

It is extremely difficult to establish a fixed amount for the deflections permissible without knowing any of the conditions under which the unit shall be operated, such as the impact loads that it will encounter, the duration of load in the various ratios, what material is used, as well as the type of mountings and surrounding elements. For this reason all factors should be analyzed carefully before one is in a position to determine what deflections can be allowed.

All of us have certain units with which we have had experience, and which provide us with some form of a yardstick. Although this may not be strictly an engineering method, I believe that we all recognize that past performance gives us valuable experience upon which to base our future designs.

Claims Gears Blamed for Deflections of Other Parts

—B. W. Keese

Wisconsin Axle Division, Timken-Detroit Axle Co.

IT must be admitted that, in this present era of engineering, gears, shafts, and bearings are being stressed to their utmost and there is a constant tendency on the part of all manufacturers to increase input torque without opportunity to change the driving members accordingly.

However, it is believed that there has been some laxity in the study of deflection which erroneously has been blamed not only on bearings, but gears. Without a doubt, gear strength and gear changes to coarser teeth or wider faces have been made in units to increase the life, whereas actually undue deflection in shafts or supporting members has been almost solely responsible for the failures.

Of what avail is the 3 in. face of gear if the shaft or housing deflects to a point which actually results in ½ in. face contact?

With our present gear-cutting equipment and precision measuring devices, we are still inclined to permit excessive deflection to creep in to destroy most of the accomplishments. It is a common fault when speaking of deflection to associate it immediately with shafts, whereas actually housing designs, bearing installation clearances, and bearings themselves are all guilty offenders and should receive the same consideration and study as does the shaft.

Unfortunately, although deflection can be calculated, it is usually diffi-

cult to analyze and chart the results of the deflection on the gear teeth, the resulting bearing-path wear in the antifriction bearings, and the effect on the other miscellaneous component parts. Therefore, it is almost essential that an experimental unit be made up and the unit thoroughly tested by bluing the tooth contacts as well as the raceways of the antifriction bearings, then operating under a full torque load, disassembling and checking in order to learn the true results of the deflection and whether it works in favor of the design, as is sometimes the case, or the opposite, which is most frequent.

After determining the results of the deflection, the next step would be to analyze the component parts of the unit by means of using indicators in the manner described by Mr. Boden, and then correcting the offending members by additional ribbing of the housing and strengthening the weak parts.

Those who manufacture a unit which will fit into Timken Roller Bearing testing equipment are fortunate and will receive a most comprehensive test and recommendation for improvements.

An analytical study of the deflection charts usually indicates a very sharp increase of deflection as the loads pass through the 100 per cent loading zone into the overload zone. Therefore, if the unit is subjected to any overload, it is well to increase the input torques from 125 to 150 per cent of the actual loading and arrange the design accordingly.

Although many believe that measures in corrective designing to the limit or *control deflection* are expensive, it is often surprising how the increase to heavier shafts and more rigid housings actually reduces costs by permitting the use of not only smaller gears, but the choice of smaller bearings.

In instances where testing equipment is not available, often much can be learned from a static deflection test by arranging a beam and scale on the *input shaft* of the unit set to the exact torque required and the load applied at the *output shaft* by means of levers and jacks. Usually the biggest problem is to find a method of anchoring the unit.

Although this static test is not as accurate as the live test described by Mr. Boden, it is indicative of weak members and greatly assists in making corrective measures.

Relative Harmful Value of Deflections

—Thomas Barish

Assistant Chief Engineer, Marlin-Rockwell Corp.

TWO important interpretations should be added in evaluating data presented. All of the deflections are harmful only in disturbing the uniformity of tooth loading.

The ideal would be if one contacting gear face were taken as a reference point and all deflections translated as relative movement of the mating gear with respect to the reference tooth. The author's present method approaches this ideal as closely as possible by using the pinion housing as a reference point which automatically should give pinion tilt as an independent reading. The effectiveness of this compromise depends on whether the reference point, the pinion housing, moves at all with respect to the gear support. It, therefore, seems desirable to check such deflection by having indicators read on the larger part of the banjo at some point with relatively little stressing, such reading being both to check lift and side tilt of the reference point. Perhaps the author already has checked in this way and found the deflection negligible.

The second interpretation is the relative harmful value of each of the deflections. Movements in direction *A* of Fig. *A* do not interfere with uniformity of load distribution or smoothness of gear action. They only make the tooth contact angle steeper. Hence the only possible harmful effect is a small increase in separating forces and bearing loads. It would be interesting to know how much deflections of the magnitude indicated by Mr. Boden affect the tooth contact angle.

Likewise movements in direction *B* are not necessarily harmful unless they change the uniformity of load distribution or the smooth meshing of the gears.

For spur gears or helical gears both of these deflections are harmless if they are equal at *C* and *D*. Trouble arises when these movements are unequal, and the important reading to note is not the deflection at *C* and *D*, but the difference between these two movements, both in direction *A*, and direction *B*.

Unequal deflections in direction *B* are more harmful than those in direction *A* in the ratio of cos t/ sin t or cot t (from 3.9/1 to 2.7/1 for common pressure angles).

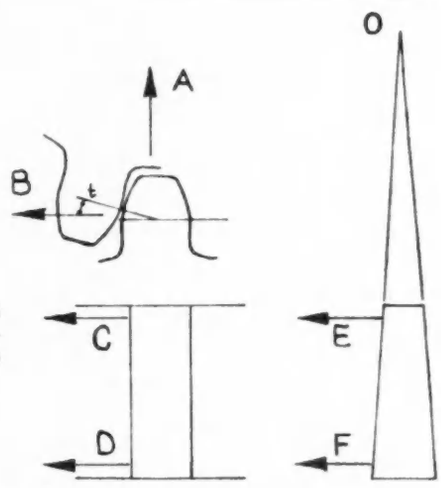


Fig. *A* – (Barish Discussion) – Direction of Various Movements

A spur gear half way between bearings on a uniform shaft could withstand relatively large shaft and bearing deflections because the teeth of the two mating gears would remain parallel.

A bevel or spiral-bevel gear would have undisturbed load distribution if the deflections varied approximately with the cone center distances, *OE*/*OF* on Fig. *A*; I say approximately because this ratio should be modified somewhat for the relative rigidity and strength of the tooth itself.

This ratio has direct bearing in applying Mr. Boden's test results on the question of overhung versus straddle-mounted pinions.

Figs. *B* and *C* reproduce two illustrations from Mr. Griswold's paper on hypoid gears presented at the January, 1937, Annual Meeting of the Society.[a] The straddle pinion deflections, if they occurred as shown, would be practically harmless, whereas the overhung pinion deflections produce the maximum tilting effect.

[a] See S.A.E. TRANSACTIONS, Vol. 32, May, 1937, pp. 194-205: "Hypoid Rear-Axle Design and Lubrication," by W. R. Griswold.

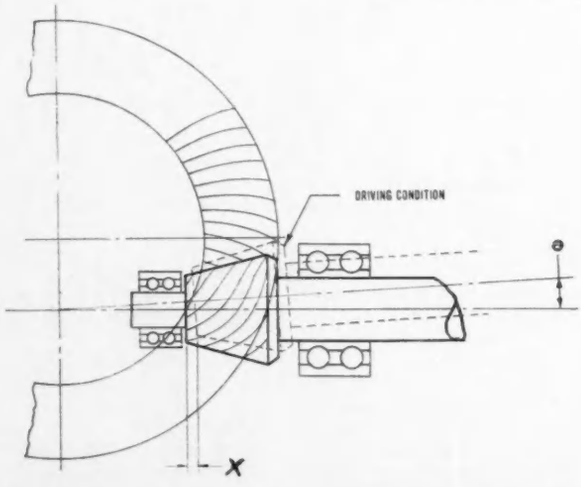


Fig. *B* (Barish Discussion) – Exaggerated Displacement of a Straddle-Mounted Pinion under Load

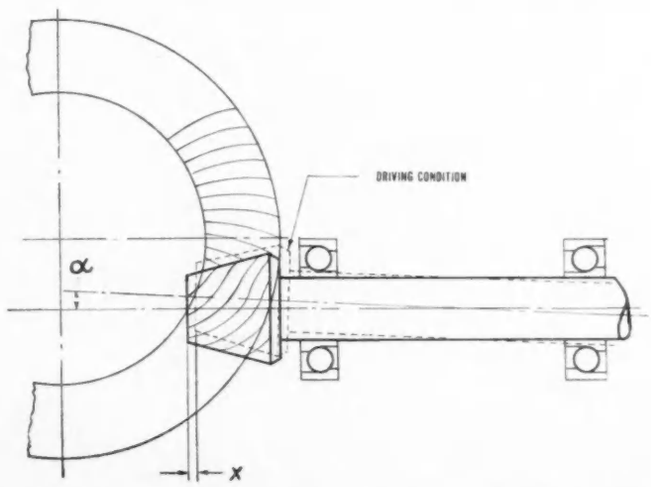


Fig. *C* (Barish Discussion) – Exaggerated Displacement of an Overhung Pinion under Load

Some Factors Controlling Part-Load Economy

By Hector Rabezzana

AC Spark Plug Division, General Motors Corp.

AUTOMOBILE engines, Mr. Rabezzana declares, have reached a point where it is very difficult, by a single improvement, to render any decidedly better performance, adding that, in his opinion, any further worthwhile improvements in engine economy on a conventional engine will be made only by several well-directed moves concerning small details.

He points out that improvements in part-load economy can be obtained by using a homogeneous mixture and efficient ignition.

Multicylinder engines are the ones most affected by low economy at light loads, he maintains, and lists the following as the main disturbing factors inherent in the majority of them: non-uniform mixture distribution among cylinders; low mixture speed in the manifold at light loads; poor turbulence in the cylinder-head; stratification of the mixture; dilution of the mixture due to the valve timing overlapping the high exhaust back-pressure; poor scavenging around the spark-plug gap position; and improper mixture at the spark-plug gap.

IT is logical to think that the best economy in fuel that can be obtained under different loads and speeds in an engine is when the mixture is absolutely homogeneous and the air-fuel ratio the closest to the optimum, compatible with a good and steady ignition.

Multicylinder engines are the ones most affected by low economy at light loads. The following are the main disturbing factors inherent to the majority of them:

1. Non-uniform mixture distribution among cylinders.
2. Low mixture speed in the manifold on light loads.
3. Poor turbulence in the cylinder-head.
4. Stratification of the mixture.
5. Dilution of the mixture due to the valve timing overlapping and high exhaust back-pressure.
6. Poor scavenging around the spark-plug gap position.
7. Improper mixture at the spark-plug gap.

In regard to non-uniform mixture distribution in different cylinders we already have pointed out in an article previously published how to find out faulty cylinder distribution by using a spark-plug temperature method. Present-day engines, in this regard, are far better than many engines of several years ago, but we are still far from an ideal condition. More work done along this line, no doubt, would improve matters for the future.

Figs. 1 and 2 – Mixture Distribution with Carbureted and Gaseous Fuel – In the six-cylinder engine having a manifold

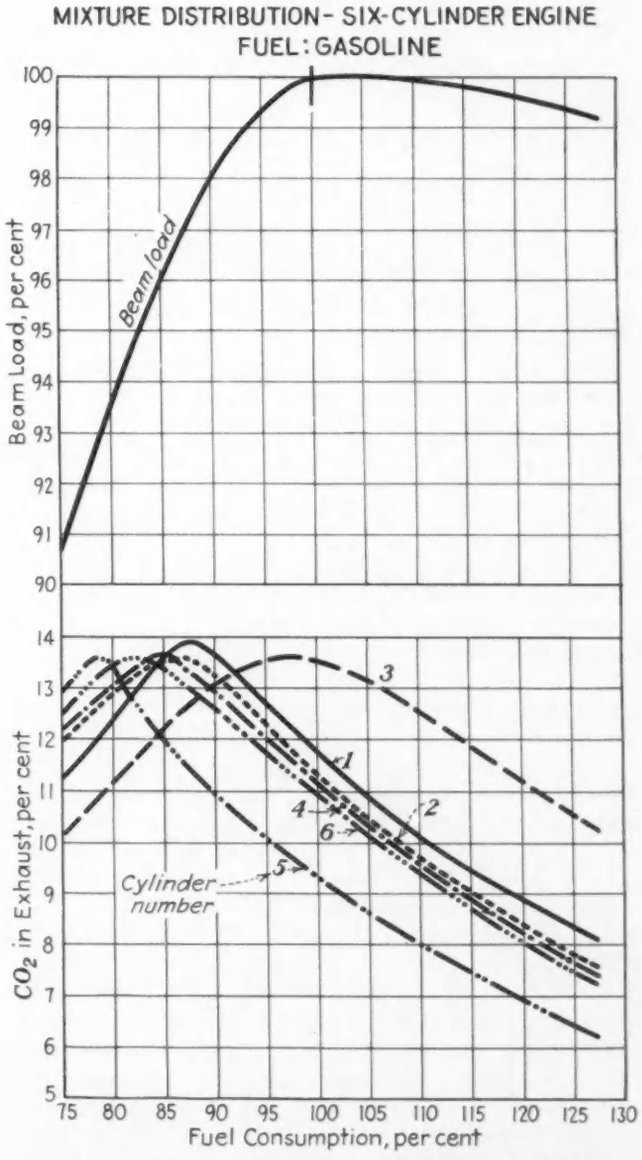


Fig. 1 – Mixture Distribution with Carbureted Fuel

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 15, 1938.]

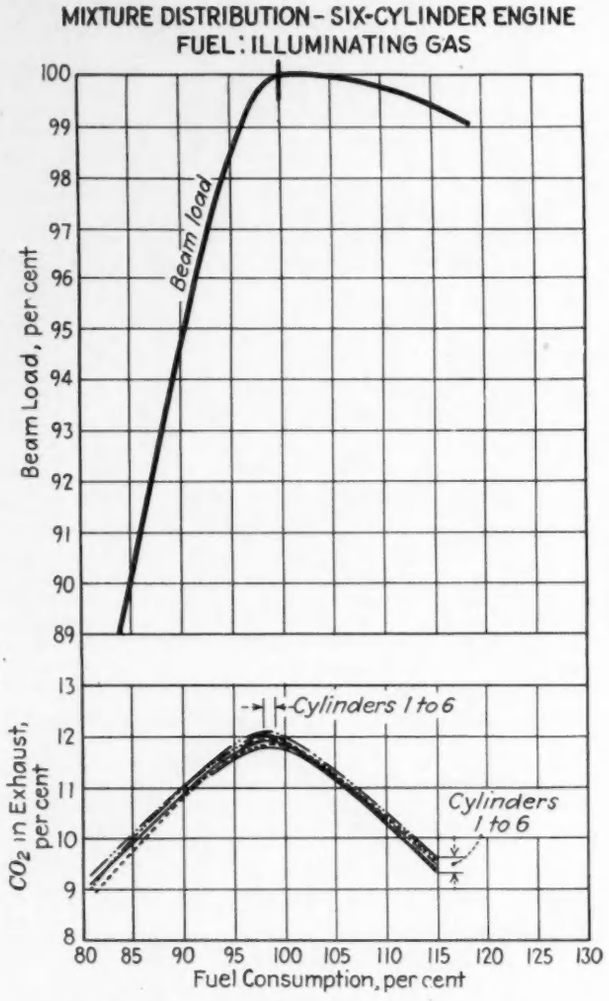


Fig. 2 – Mixture Distribution with Gaseous Fuel

of conventional design and operated with gasoline, an exhaust-gas analysis of the individual cylinders shows non-uniform mixture distribution (See CO_2 curve on Fig. 1). If the same engine is operated with gaseous fuel, the mixture distribution is uniform (See Fig. 2).

Low mixture speed at light loads in the induction system is a result of the trend of getting maximum engine power at top speed. This operation entails large manifolds and down-draft carburetion. Probably the next trend will be towards a compromise for economy and flexibility rather than top speed, using smaller manifolds to increase the mixture speed at part load.

Turbulence is proportional to mixture speed in the induction system and in the cylinder – the greater the turbulence, the more homogeneous the mixture.

Mixture stratification is caused mainly by centrifugal forces present in the mixture stream, which cause the heavy part of the mixture to condense in strata during the turns which it is compelled to take before reaching the top of the piston. The ramming effect originated by the engine firing sequence, the condensation of the fuel because of sudden expansion, and sudden temperature changes are other sources of stratification.

Stratification can be corrected either by improving the induction-system design, by using a supercharger between the carburetor and the cylinders, or by using mechanical mixers to keep fuel atomized as much as possible.

Dilution of the mixture happens during the overlapping of the valve timing, which causes exhaust gases to be introduced into the intake manifold.

Fig. 3 – The Effect of Exhaust-Gas Dilution – By using a valve timing in which the exhaust and intake do not overlap, part-load economy can be increased. Likewise, by decreasing the exhaust back-pressure, a better mileage can be had. This curve shows the effect of exhaust-gas dilution.

It should not be a difficult problem to develop a variable valve timing, which will be correct at all speeds. Such a system will not only give better fuel economy but it will give better car flexibility and better idling performance, especially if helped by an automatic lash adjuster.

Good scavenging around the spark-plug can be helped by using smaller size plugs with a small dead space between the insulator and the shell, and by locating the plug in such a place that the incoming gases have a chance to push out the combusted ones.

In analyzing the problem of mixture incendivity, we see that the most important thing is to have the right mixture ratio at the spark-plug gap. The mixture may be, on an average, quite good in the cylinders, but it may be stratified and very, very poor at the gap of the spark-plug. In that case, if we want to ignite the mixture, we will have to put the gap in richer strata or to resort either to a larger cap or to a further advance of the ignition, or to have more energy in the spark. We already have published an article covering this subject. Any of these systems, however, shifts the effect and does not remove the trouble's cause. Larger gaps cause shorter spark-plug life between re-gappings, and shorten the life of the plug when the insulator of the plug becomes covered with combustion residuals. Increasing the advance makes the engine more critical to different types of fuels; the reasons which cause poor mixture at the spark-plug gap are also the causes

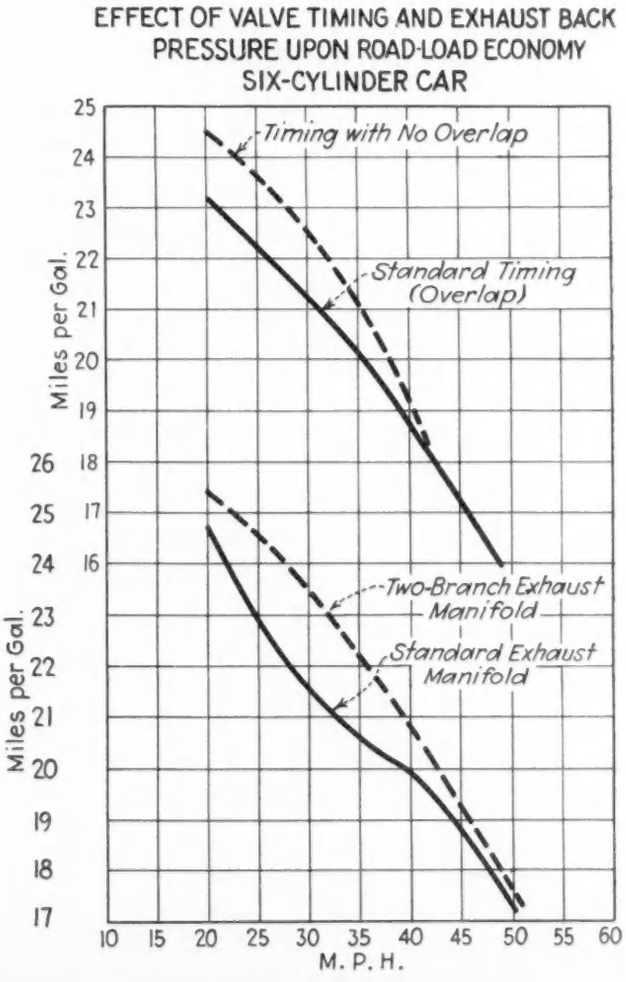


Fig. 3 – Effect of Exhaust-Gas Dilution

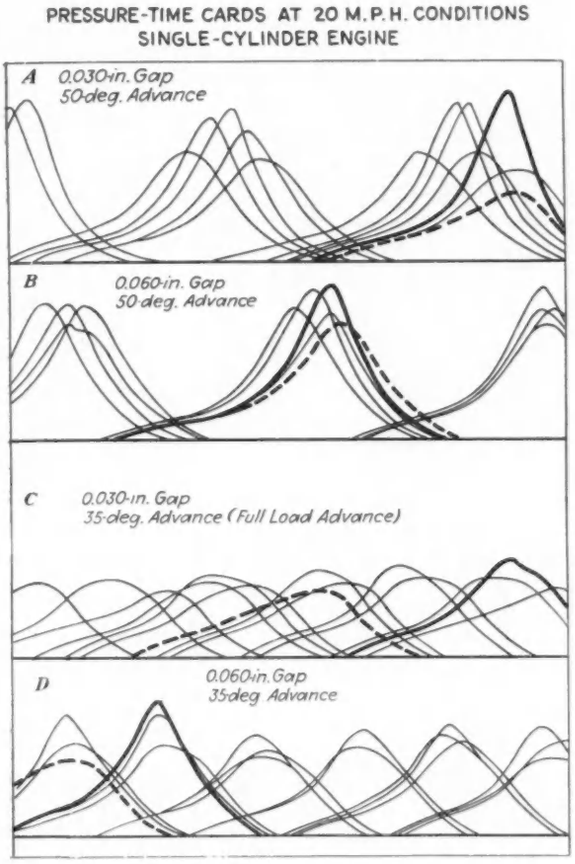


Fig. 4

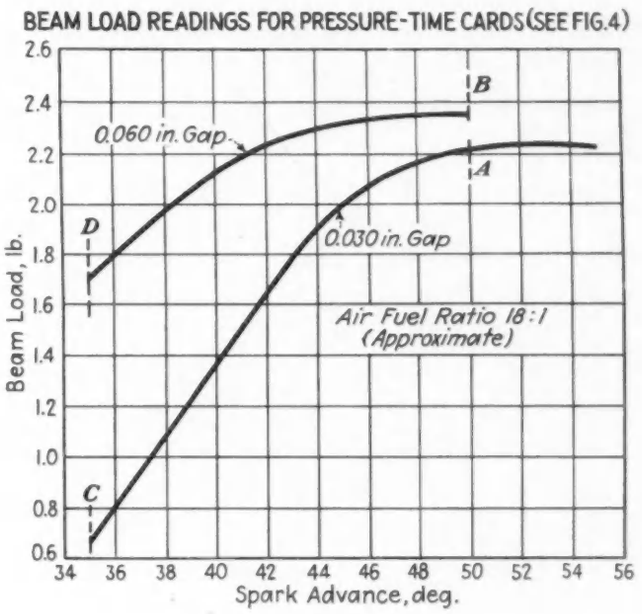


Fig. 5

Figs. 4 and 5 – Effect of Large Spark-Plug Gaps

of mixture stratification which, in turn, controls the flame propagation speed. The rate of combustion of the mixture is proportional to the density of the mixture and the air-fuel ratio. When we have stratification, the flame propagation is not uniform and, consequently, the explosion pressures vary continuously, and so does the engine efficiency. I would say that a poor mixture at the gap is a true indication that there is room for improvement in part-load economy.

Figs. 4 and 5 – Effect of Large Spark-Plug Gaps – On Fig. 4 actual pressure-time indicator records are retraced, taken in a single-cylinder test engine in which the conditions existing

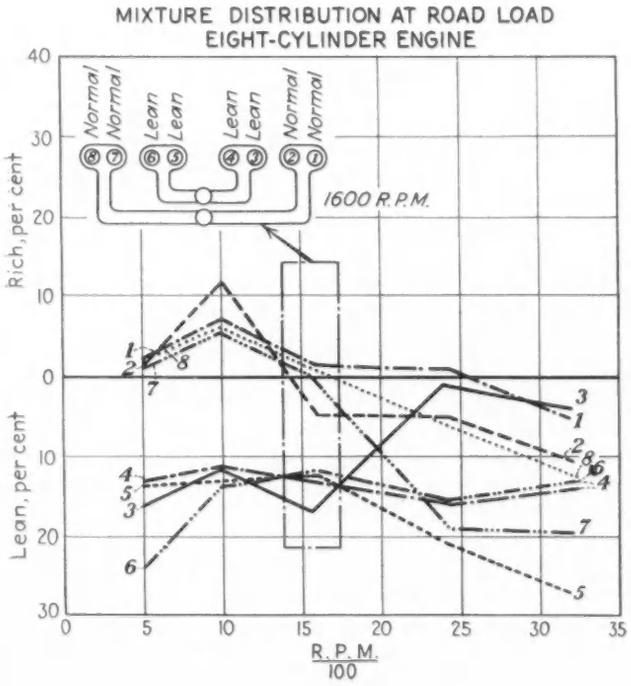


Fig. 6

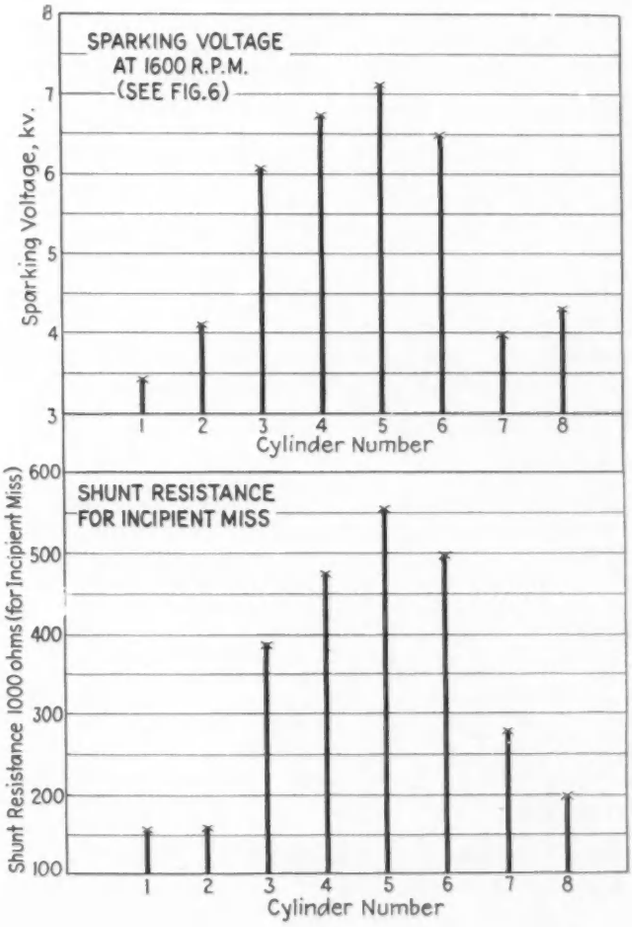


Fig. 7

Figs. 6 and 7 – Effect of Mixture Distribution upon Sparking Voltage and Shunt Resistance

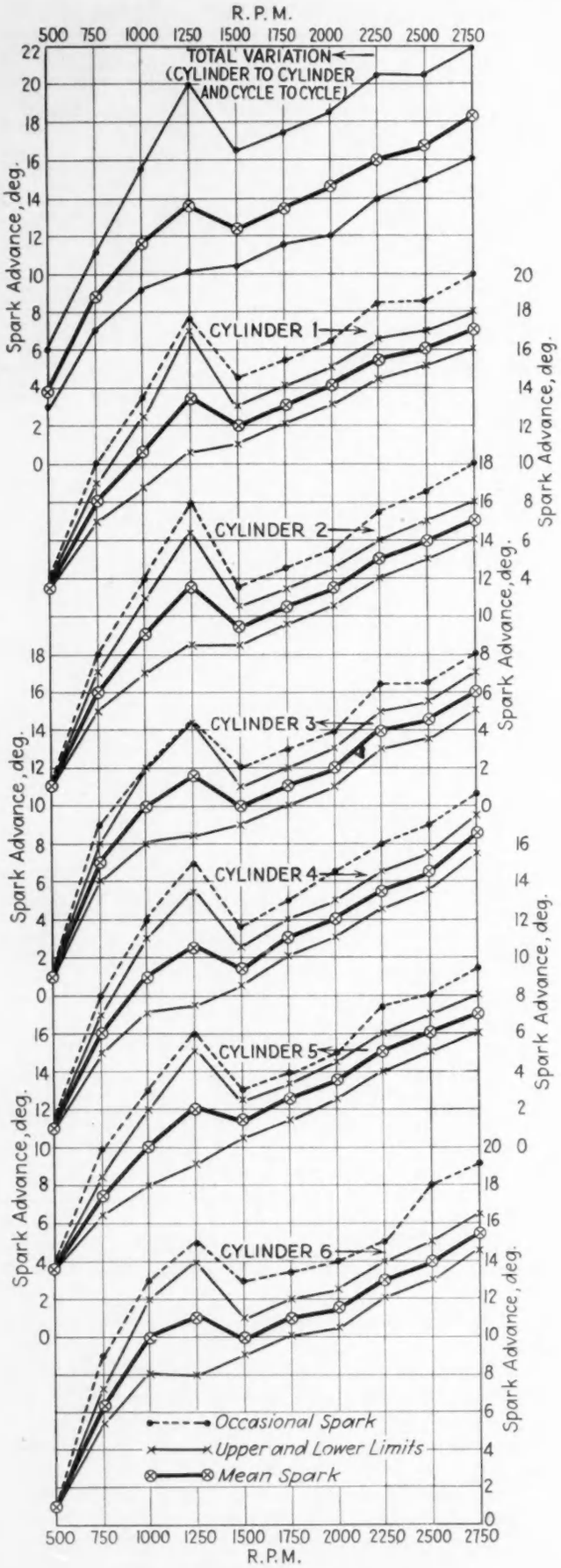


Fig. 8 – Ignition Timing Variations

in a six-cylinder engine at road load were simulated. Fig. 5 represents the beam load readings. In the pressure-time records the heavy full and dotted lines represent the optimum and the worst conditions, respectively. At full-load advance (35 deg.) and normal gap (*C*) most of the pressure cycles show delayed, slow combustion. To compensate, the spark advance can be increased (*A*), or the discharge characteristics and duration of the spark modified by increasing the gap length (*D*) or both (*B*). This effect of the large gap can be utilized only in lean, highly diluted mixtures.

Another factor that has quite a good deal of importance in fuel economy is the amount of combustion deposits over the spark-plug insulators. These deposits have several origins: from metals surrounding the spark-plug gap; metals in suspension in the fuel or oils; road dust; rust; and so on. As we have stated previously, these deposits may become, under appropriate conditions of temperature, good conductors of electricity and, consequently, may shunt some of the spark's energy. This shunting may be of such magnitude that there will not be enough energy left to form a spark, thus causing missing in the ignition or, if the spark becomes very weak, a delay in ignition. This missing, in turn, cools the spark-plug to a point where the conductivity drops enough so the ignition becomes effective again for a few more cycles, until the insulator coating becomes hotter, and so on. This is a type of missing which cuts down the mileage at top speed, both in regard to the car speed and miles per gallon. Cooler plugs and correct spark-plug location may go a long way toward bringing longer life between spark-plug cleanings.

There is another line of attack for better fuel economy and that is from the ignition standpoint. We know that spark-plug life is limited. If the right plug is chosen, this limit is governed by two factors: the first, the amount of deposit that accumulates on the insulator and the other by the excess of energy available in the ignition system to overcome the conductivity of this deposit.

Figs. 6 and 7 – Effect of Mixture Distribution upon Sparking Voltage and Shunt Resistance – Fig. 6 shows the results of a mixture-distribution test on an eight-cylinder engine at road-load conditions. At 1600 r.p.m., for example, the cylinders Nos. 1, 2, 7, and 8 are normal, while Nos. 3, 4, 5, and 6 are lean. On Fig. 7 is shown how this condition affects the ignition requirements. The plugs in the lean cylinders require a much higher sparking voltage than the ones in the normal ones and, therefore, can stand much less shunting (coating on the spark-plug insulators) than the ones in the normal cylinders.

Many multicylinder engines of today require long spark-plug leads. The distributor and ignition coil sometimes are located so that they also require long leads. Long leads mean a loss of spark energy and, in many cases, misfiring or cross-firing, due to the transfer of energy from one cable to another; consequently, a good part of the current, which should be available for the ignition, is squandered.

The factors affecting ignition are capacity losses in secondary cables, coupling losses between the secondary cables, and instability of the timing. The first and second factors can be minimized by using low-capacity cables, very short leads, and by keeping the cables separate without touching each other. Cross-firing is caused by an induced ignition of the mixture in one cylinder during the intake cycle. Sometimes it has been corrected by another wrong – by shielding one or more cables. This shielding introduces more capacity losses. A better way would be to install the distributor head in the center of the cylinder-block and drive it by a flexible shaft. The distributor head could have the coil included in the same casing and, in this way, secondary leads from the coil to the distributor could be done away with. This type of distributor drive will permit the installation of the breaker points on the

(*Concluded on page* 528)

Cylinder Cooling and Drag of Radial Engine Installations

By Kenneth Campbell

Project Engineer, Wright Aeronautical Corp.

REDUCTION in the baffle pressure drop required to cool promises a great reward from every standpoint, Mr. Campbell asserts, and further progress in this direction is theoretically possible through both fin and baffle development. Present aluminum-alloy head fins, he continues, are still at least twice too thick for their height, and the limiting factor in fin design is no longer engineering knowledge but the manufacturing art.

The cooling problem presents a "potential limitation" to the engine manufacturer, Mr. Campbell points out, explaining that two conflicting requirements must be met, each of which grows more severe as engine powers and airplane speeds increase. The first requirement dictates that, for the improved engine, the rate of waste-heat transfer must be increased whereas the second requires that, for the improved airplane, the cooling drag must be decreased.

The engine manufacturer can improve the rate of heat transfer either by increasing the velocity of the cooling air past the surfaces, or by increasing the area of the useful heat-transfer surface, the author states. He then shows that the latter method is far less costly, especially when done by adding to the height of existing fins and increasing the cross-sectional area of the heat-transfer passages.

Data are presented to enable the cowl designer to provide the effective cowl exit-slot width necessary at his particular indicated air speed to meet the baffle pressure drop requirements specified for the conditions.

Three main avenues of approach to the problem of increasing the low baffle pressure drops existing at very low airplane speeds in order to obtain the velocity of cooling air necessary for increased heat rejection, are studied:

1. Use of the propeller slipstream to induce a depression after the baffle passages.

2. Use of the propeller slipstream to build up a pressure before the baffle passages.

3. Use of a fan for either purpose.

The research data presented indicate that the progress to date in cylinder cooling of air-cooled radial aircraft engines, though considerable, has not yet reached the point of diminishing returns.

FOR the engine manufacturer, in the present period of rapid progress in increased horsepower outputs, few design problems are permanently solved. He continually is conducting research, design, and tests to overcome potential limitations to his next advance. The subject known as "cooling" might well be considered as one of the most persistent of all such potential limitations, and is therefore one to which he gives continuous attention. The reason for this condition is the necessity for meeting two apparently conflicting requirements, each becoming more severe as engine powers and airplane speeds increase from year to year. These requirements are, in simplest form:

1. For the improved engine: Increase the rate of waste heat transfer.

2. For the improved airplane: Decrease the cooling drag.

The objective of the engine manufacturer with regard to cooling development has, therefore, always been not the improved engine alone but the improved airplane installation. Because of the conflicting nature of the foregoing requirements, the main avenues along which he usefully may direct his cooling-development efforts must be carefully chosen. In determining what these avenues are, a very condensed discussion of the present status of cooling practice is in order.

The drag chargeable to cooling is due to:

A. The energy theoretically required to force the cooling air through the internal resistance of the cooling air passages. This energy is subsequently referred to as "internal air drag horsepower."

B. Additional energy losses chargeable to the cooling air passages, principally due to collecting the air and returning it to the slipstream.

C. The energy to overcome that portion of the form or external drag of the engine installation in excess of the drag due to a perfect streamline shape of the same diameter, in so

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 14, 1938.]

far as such deviation from streamline form has been undertaken because of cooling considerations.

With regard to Item A, the internal air drag horsepower, this is probably the most important factor on which cooling drag depends. Its calculation is dependent on an exceedingly simple relation, which is that the energy required to force the cooling air through a resistance is directly proportional to the volume of flow times the drop in total pressure across the system. The relation may be expressed in numerical form as follows:

$$\text{Internal Air Drag Hp.} = \frac{\text{Volume (ft.}^3\text{/min.)} \times \text{Pressure Drop (lb./ft.}^2\text{)}}{33{,}000}$$

$$= (\text{ft.}^3\text{/min.}) \times (\text{in. of } H_2O) \times 0.000157$$

From this relation it appears that the energy to cool could be kept to a low value if either the volume or the total pressure drop across the system were kept close to zero, and provided good heat-transfer performance were obtainable at the same time. However, it was found several years ago[1] that the combination of large volume and low total pressure drop was difficult to attain because of high turbulence losses at many points throughout the cooling system, and that velocities obtainable over the rear heat-transfer surfaces using this open internal design were not satisfactory for good heat transfer. Hence, the practice in this country became that of limiting the volume to a low value by means of pressure baffles, the flow being confined solely to that through the useful heat-transfer passages. An appreciable pressure differential across the fin-baffle passages exists, creating the velocity required for adequate heat transfer. For any given design of cylinder finning it is believed that, by this method, the product of volume times pressure drop across the system represents the lowest energy expenditure throughout the cowling system to provide a given required rate of heat transfer.

Baffle Pressure Drop

The pressure drop across the heat-transfer passages, or "baffle pressure drop," is the engine manufacturer's most useful criterion of performance in all cooling and drag problems, both in design and application. Being easily observed in actual installation, it has become the independent variable to which, for a given engine, he may relate cooling performance, air volume requirements, and internal air drag horsepower.

By restricting the cowl exit opening, adjustably or otherwise, a second major resistance to the flow of cooling air is placed in series with that of the heat-transfer passages of the baffled engine, and the available pressure drop across the cooling system divides itself principally between these two resistances in proportion to their relative values. However, if the cowl exit slot is designed properly, accelerating the air smoothly in a rearward direction, little energy loss is represented by the static pressure drop across this opening. The drop in total pressure across the slot is negligible if a smooth design has been accomplished, and the rearward acceleration of the air as it leaves the airplane at this point results in a forward thrust approximating the product of mass flow times the relative velocity.

With the drop across the exit slot eliminated as an important component of wasted energy, the energy loss across the engine becomes of primary importance in the reduction of cooling drag. For this reason the engine manufacturer, for many years, has been making every effort to attain his necessary heat-transfer improvement at the lowest possible value of the product of volume times baffle pressure drop.

The airplane manufacturer, for making his most advantageous installation of a given engine, must be supplied with quantitative data as to what this value is and the best means of obtaining it over the range of air speeds in which his projected operation lies.

In practice, of course, the losses in the cooling system actually cannot be confined to those represented solely by the product of engine baffle pressure drop times volume. Turbulence takes place in front of the engine, which represents energy loss and must be paid for. Turbulence after the cowl exit slot usually cannot be avoided because the velocity leaving the exit slot usually cannot equal that of the slipstream, the pressure drop available across the slot creating this velocity being never equal to the dynamic pressure corresponding to the airplane speed. (An exception to this statement is theoretically possible at very high air speed low baffle pressure drop operation when exit velocity may equal or exceed the slipstream speed by virtue of the added heat energy[2]). Nevertheless the N.A.C.A.[3] has demonstrated that it is possible to design cowlings in which the exit turbulence loss is of relatively small importance.

Despite progress made in reducing baffle pressure drop necessary for a given heat-transfer rate, the increase in engine ratings and the progress in large airplane development, particularly of flying boats, necessitate continued efforts to augment baffle pressure drop available at the low air speeds of taxiing, take-off, and climbing.

Cooling at very low air speeds is still being accomplished by the use of conventional flat open-front nacelles for obtaining a higher degree of baffle pressure drop and also turbulence for cooling the fronts of the cylinders. The total pressure entering the baffle passages always is observed to be somewhat less than the dynamic pressure corresponding to the air speed, the difference possibly being a measure of the turbulence energy losses. It may well be that turbulence-cooling is unnecessarily expensive at very high air speeds, and evaluation of methods of elimination of turbulence-cooling of the front portion of cylinders is important. The subject appears intimately associated with form or external drag.

Form drag generally is not a cooling charge but one factor in the selection of a general type of engine design, which selection is determined by a large number of other considerations. With the trend toward greatly increased horsepower of radial engines without increase in diameter, and with the tendency to build ever larger ships, the external-drag charge against the diameter of the radial engine has diminished greatly. Nevertheless, with regard to shape, if published wind-tunnel comparisons of nacelle shapes made with no propeller[4] were to be interpreted without qualification, the flat open-front nacelle would appear to have 35 per cent to 40 per cent greater external drag than a well-streamlined shape of the same diameter. Unlike internal drag horsepower, external drag horsepower cannot be maintained at a constant low value for all air speeds but increases approximately as the cube of the speed. Although, at present high airplane speeds, the foregoing penalty for imperfect shape is comparable with properly minimized internal cooling air drag, the possibility of its becoming seriously large at 400 m.p.h. makes a careful evaluation of this factor very important for future cooling and cowling development for very high-speed installations. Further wind-tunnel laboratory investigation of this point appears in order since it would seem that propulsive efficiency and effect of the propeller on drag are closely related to the problem. With modern high-powered large-diameter low-speed propellers where blade shanks contribute no thrust, better propulsive efficiency may result from blocking

[1] See S.A.E. TRANSACTIONS, Vol. 29, May, 1934, pp. 147-166: "The Cooling and Cowling of Radial Air-Cooled Aircraft Engines," by Rex B. Beisel, A. Lewis MacClain, and F. M. Thomas.

[2] See *Aircraft Engineering*, Vol. VIII, August, 1936, pp. 218-220: "Ducted Radiators," by F. W. Meredith.

[3] See N.A.C.A. Technical Report No. 592, 1937: "Full-Scale Tests of N.A.C.A. Cowlings," by Theodore Theodorsen, M. J. Brevoort, and G. W. Stickle.

[4] See S.A.E. TRANSACTIONS, Vol. 32, December, 1937, pp. 581-595: "Design of Cowlings for Air-Cooled Aircraft Engines," by Donald H. Wood.

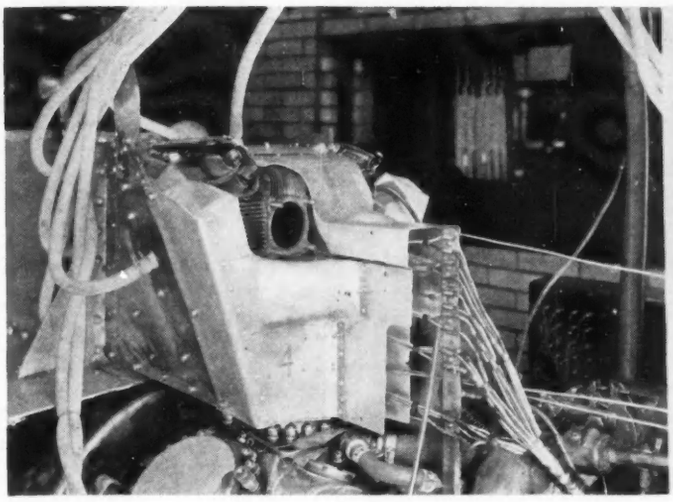


Fig. 1 – Cyclone Cylinder Assembly Equipped with 36 Thermocouples

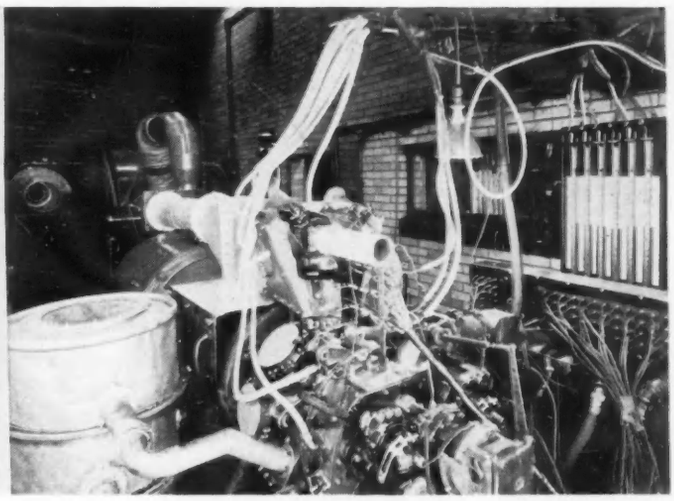


Fig. 2 – Single-Cylinder Test Engine

off the center portion of the propeller disc, as is now accomplished by conventional radial-engine cowlings.

Further substantial progress in reduction of baffle pressure drop required for a given heat-transfer rate should make possible the reduction of the large frontal opening and a much closer approach to the streamline form of nacelle.

With regard to reduction of internal air drag horsepower by reducing to a minimum value the product of volume times baffle pressure drop required for a given heat transfer, it is interesting to examine which design policy the engine manufacturer should pursue to attain this end, that is, should he reduce volume or pressure drop? For given atmospheric conditions of pressure and temperature, the rate of heat transfer from the cylinder walls is a function of:

a. The velocity of cooling air past the surfaces.

b. The area of useful heat-transfer surface.

Obviously, the manufacturer theoretically can move toward either of two extremes in improving his rate of heat transfer. He may leave his useful heat-transfer area alone and increase his cooling-air velocity (and corresponding baffle-pressure drop) or he may continue with the same or lower baffle-pressure drop but increase his useful heat-transfer surface. A very simple example shows that the latter method, that of increasing the heat-transfer surface, is a far less costly method of improving cooling as far as drag is concerned. Let us suppose that it is desired to improve heat-transfer 50 per cent under given atmospheric conditions:

a. By increasing the velocity only as required, or

b. By adding to the height of the existing fins, increasing both the useful surface slightly less than proportionally (for head fins of current design) and also the cross-sectional area of the heat-transfer passages.

The baffle-pressure drop increases approximately as the square of the velocity through the passages and the velocity in turn must be increased approximately as the 1.67 power of the rate of heat-transfer desired. The figures shown in the following columns for the a and b methods of improvement represent the ratios of increase of the various values contributing to the increased cooling power consumption resulting from the change.

Ratio of values after increasing heat transfer, to values before increase	Method a	Method b	
Heat-transfer ratio (specified)	1.5	1.5	1.0
Required surface area ratio	1.0	1.5+	1.5+
Required velocity ratio	1.97	1.0	0.5
Required volume ratio	1.97	1.5+	0.75+
Required pressure-drop ratio	3.86	1.0	0.25
Resulting power consumption ratio	7.60	1.5+	0.20+

Comparing the first two columns, the great superiority in power consumption of improvement by Method b is of course attributable to the fact that, although the volume is increased, the baffle pressure drop is not, whereas both must be increased substantially in the case of Method a. The advantage of Method b is brought out more forcibly if we calculate the effect of increased fin area on minimum baffle pressure drop required for the same heat transfer. Assuming a 50 per cent increase in useful fin area but seeking no further increase in heat transfer, it will be observed from the third column of the table that the required velocity is reduced to one-half its original value, the corresponding baffle pressure drop to one-fourth, and power consumption to one-fifth. The proper computation of optimum fin design is not nearly so simple as this, partly due to a declining temperature gradient of the fins, but these examples serve to illustrate that great reductions in cooling drag are possible by achieving relatively small reductions in baffle pressure drop required for given rate of heat transfer.

Referring again to the subject of pressure baffles, it is thus evident that, although these baffles represent the greatest economy in that they restrict the flow of cooling air for a given fin design, this statement must not be interpreted to mean that flow should be reduced at the expense of useful fin area. Larger useful fin areas giving the same heat transfer at lower velocity and baffle-pressure drop bring about substantial drag reduction even though the passages are increased in area.

Fin Developments Promising

It is thus clear that reduction in baffle-pressure drop required to cool promises a great reward from every standpoint. Further progress in this direction is theoretically possible through both fin and baffle development. Of the two, fin development still appears to promise the greater return. The research conducted by the N.A.C.A. over the past 8 to 10 years has now resulted in fairly complete instructions to the engine manufacturer as to which way to go in his fin design and how far it is worth while to carry his improvement if he can. As a result, the limiting factor in fin design is no longer engineering knowledge but the manufacturing art. Methods of manufacturing fins are continually being scrutinized, and it is probable that the future will see useful fin areas greatly increased just as, in the past, we have seen them doubled and doubled again.

Present aluminum-alloy head fins, which really represent a beautiful foundry job, are still probably at least twice too

thick for their height and should be spaced more closely; yet they are only 1/16 in. thick, 1½ in. high, with 5/32 in. clear space between surfaces. It is true that very thin fins which are useful at low air velocity would, when subjected to high air velocities, lose most of their heat to the air through their inner surfaces and the outer parts would be useless added weight under this condition but, as outlined before, the minimum drag condition is obtained by operating at relatively low baffle-pressure drops in high-speed flight as well as in low-speed flight, and the high-air-velocity condition, therefore, should not exist.

Further improvement of baffles over present-day design for obtaining a given heat transfer at a substantially reduced baffle-pressure drop appears rather less promising. It has been pointed out[5] that the flow energy wasted by the practice of discharging the high-velocity air directly from the baffles instead of through diffusers constitutes about 80 per cent of the total energy loss chargeable to the baffle passages. However, a radial-engine cylinder assembly of current design with its associated parts, including intake pipe, exhaust stack, spark-plug and cable, push-rods, and fins converging at right angles at the rear of the head, forms a structure apparently rather unsuited to taking advantage of the foregoing perfectly sound theory. Tests made on a Cyclone cylinder assembly equipped with 36 thermocouples (Fig. 1) on a single-cylinder engine (Fig. 2) with about 12 different modifications of baffle design, failed to show any appreciable variation either in volume or temperature distribution. The entering cooling air was made turbulent by passing it through a spoiler or flat-plate orifice of approximately rectangular proportions. Although further flow tests may demonstrate a suitable method of application, the practicability of diffuser-type baffles, from a standpoint of endurance and engine servicing, would have to be given consideration.

[5] See N.A.C.A. Technical Note No. 620, October, 1937: "Energy Loss, Velocity Distribution, and Temperature Distribution for a Baffled Cylinder Model," by M. J. Brevoort.

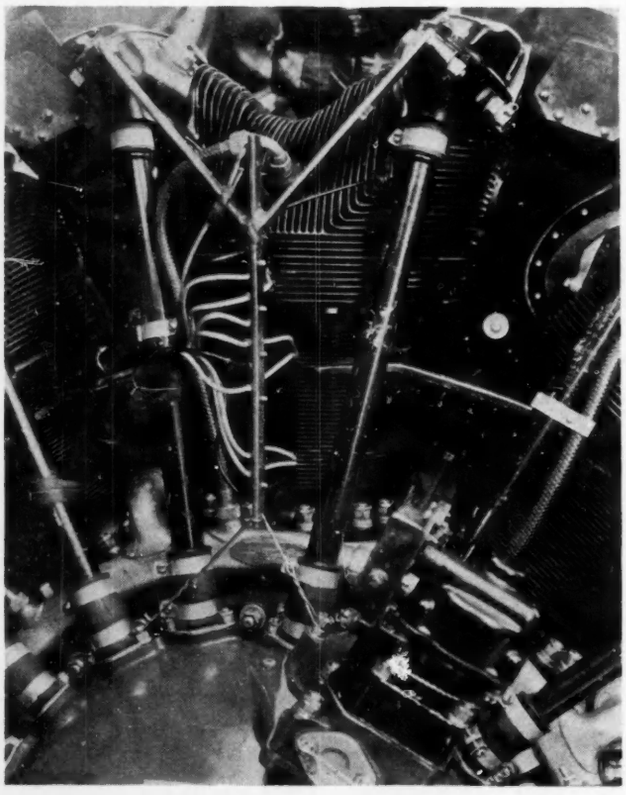


Fig. 3 – Cylinder Installation in Airplane with Standard Cyclone G-100 Baffles

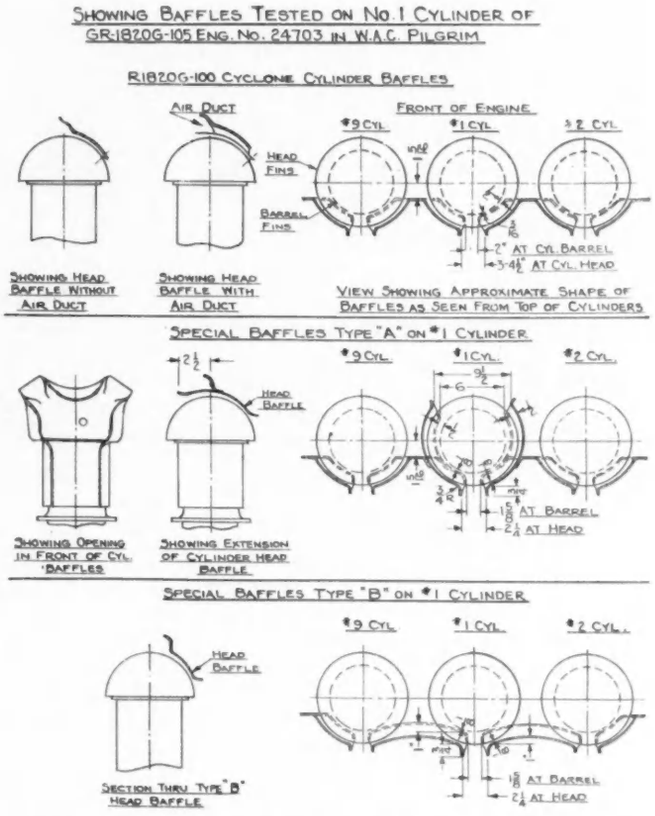


Fig. 4 – Design Details of Three Sets of Baffles Used in Observation of Temperature Distribution over a Cylinder in Flight

Flight tests also have been made with three widely differing types of baffles applied to a cylinder similarly equipped with 36 thermocouples to determine the actual distribution of temperature and the effect of extremes in baffle design in actual airplane flight. Fig. 3 shows the cylinder installation in the airplane with standard Cyclone G-100 baffles, and Fig. 4 gives design details of the three sets of baffles tested. Data were taken for numerous flight and engine operating conditions, Figs. 5 *a*, *b*, and *c* (representing 120 m.p.h. climb) being fairly representative. All engine and flight conditions, including fuel-air ratio, were carefully controlled. Generally speaking, with front turbulence cooling, the design of pressure-type baffles for good temperature distribution in flight does not appear very critical. However, investigation of the more completely surrounding Type *A*, considered as a possibility for submerged or ducted installations where front turbulence would be eliminated, showed that these baffles improved front cooling but materially raised rear temperatures. Use of this type of baffle would require an increased effective baffle passage area and volume of flow. Some such increased cooling power cost, of course, is to be expected if turbulence were eliminated and the same overall heat transfer achieved. Temperatures observed at several points over the upper portion of the cylinder-head showed no variation with changes in baffle design.

Of incidental interest with regard to the effect of turbulence on cooling is the frequent comment that engines cool better in airplane installations than when installed on blower-cooled dynamometer stands where strong turbulence does not exist. Experiments to check this condition and to approach airplane cooling conditions on a blower-cooled stand were conducted by noting the effects of various types of turbulence-producing devices installed in front of a full-scale engine. Although almost any spoiler had some effect, the most practical device was a large sheet-metal orifice, or annular spoiler, placed in front

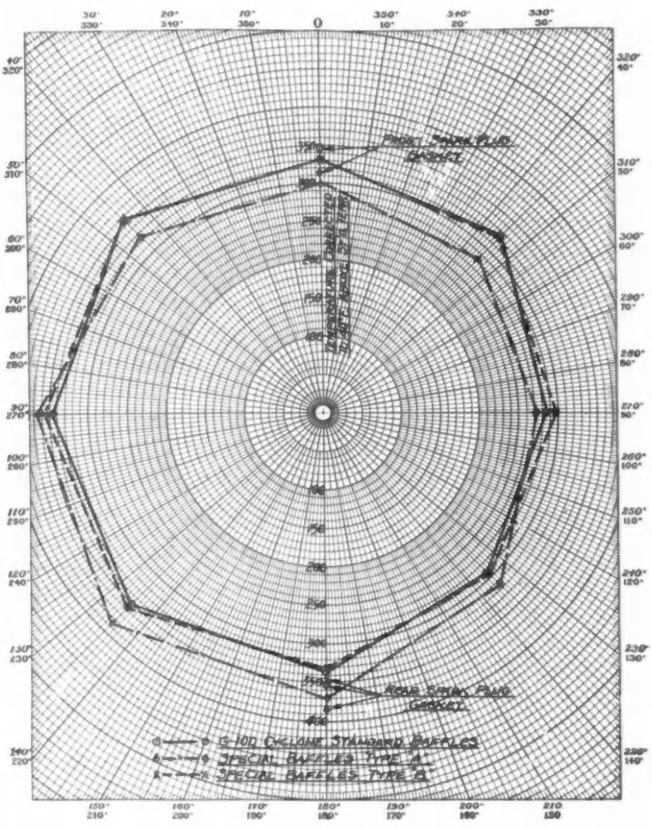


Fig. 5*a* – Temperature Distribution around Cylinder-Head

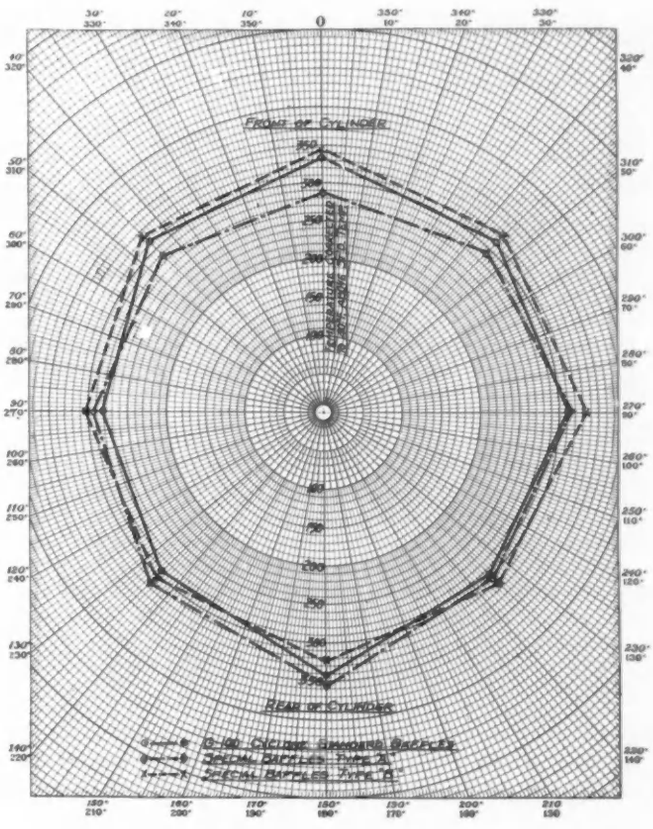


Fig. 5*b* – Temperature Distribution around Top of Barrel

Figs. 5*a*, 5*b*, and 5*c* – Observed Temperature Distribution around No. 1 Cyclone Cylinder in Wright Pilgrim Airplane in Climb at 120 M.P.H. (Ind.), 1900 R.P.M., 30 In. M.A.P., 0.095 Fuel-Air Ratio Maintained Constant

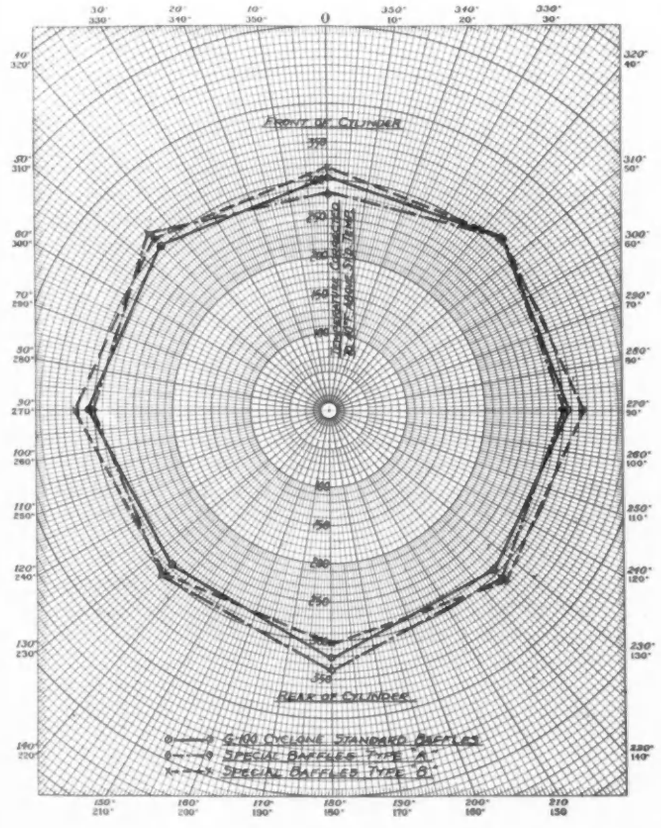


Fig. 5c – Temperature Distribution around Bottom of Barrel

of the engine, through which the cooling air had to pass and then expand, somewhat similar to cowl nose action. Fig. 6 illustrates the cylinder temperatures obtained with and without the turbulence orifice. Note that rear head temperatures as well as front show improvement using the orifice, though to a lesser degree, presumably due to the conductivity of the head.

Generally speaking, it appears that, although further revision of baffle design may not greatly reduce internal air drag horsepower, there is going to be a great further improvement in this direction when the art of fin manufacture has overtaken the science of fin design. In the meantime the airplane manufacturer demands cowl design data for making the most of established progress in his installation.

Fig. 7 gives, for a Cyclone engine with modern finning and baffling, the relation between baffle pressure drop and volume of cooling air flow under standard atmospheric conditions, determined by actual flow test on a single cylinder. These data supersede a similar curve for an earlier model of the same engine published three years ago[6]. The air volume-pressure drop relation shown is that which would prevail if there were substituted for the engine a perfect nozzle or orifice of 1.6 sq. ft. of area. This is the "effective baffle passage area" of the engine. The flow-pressure drop curve corresponding to a more powerful engine of the same diameter, assuming an effective baffle passage area of 2.5 sq. ft. also is plotted.

The pressure drop across the cooling system, measured from total at the baffle entrances to static after the cowl exit slot, is divided between the baffles and cowl exit slot in proportion

[6] See S.A.E. TRANSACTIONS, Vol. 30, November, 1935, pp. 401-411: "Evaluation of Variables Influencing Air Cooling of Engines," by Kenneth Campbell.

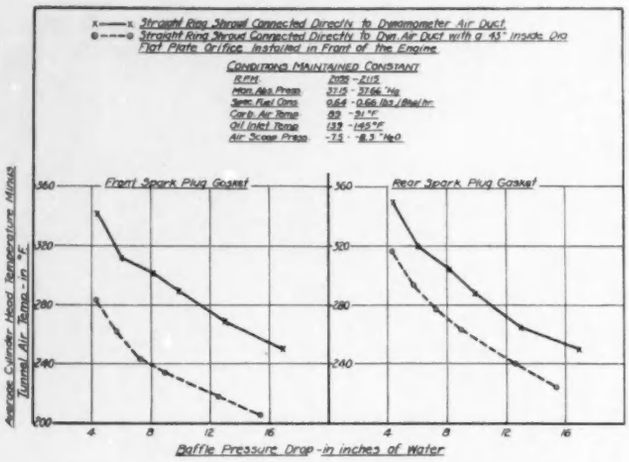


Fig. 6 – Observed Effect of Turbulence on Front and Rear Cylinder-Head Temperatures

to their relative resistances, which, in turn, are inversely proportional to the squares of their effective areas. By assuming various effective areas of the cowl exit opening and calculating for each area the ratio of baffle pressure drop to total drop available, the curves of Fig. 8 are obtained showing this relation for the two engines in question, assuming a cowl of engine diameter, and expressing the exit area in terms of "effective exit slot width." Although the shape of the curves is, of course, characteristic of the mathematical relation shown, the abscissae become larger and larger for engines of higher effective baffle passage area. It will be observed that there is a region of diminishing returns beyond the slot width required to obtain across the baffles more than 75 per cent or 80 per cent of the total drop across the system. It is also indicated that, in order ideally to utilize this proportion of total drop for cooling, it is necessary to have an effective exit slot

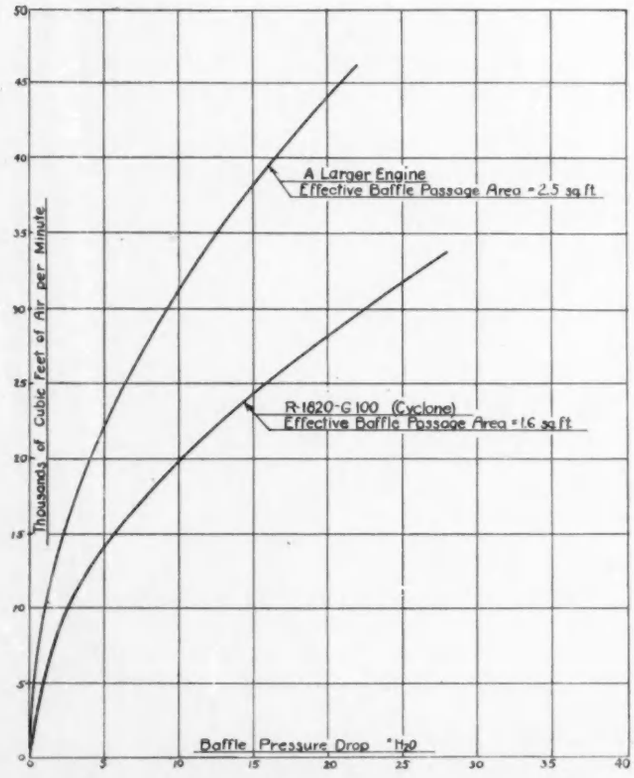


Fig. 7 – Observed Relation of Cooling Air Volume to Baffle Pressure Drop for the Cyclone Engine and the Corresponding Relation for an Engine of Higher Output

width of about 3 in. for the Cyclone, and 4 to 5 in. for the engine of larger assumed baffle passage area.

For level flight at any specified indicated air speed, one may apply representative overall cowl drop values to the curves of Fig. 8 and thus approximate numerical values of baffle pressure drop expected with various effective exit slot widths. Flight test data on cowls with various locations and designs of exit slot and published research data indicate static pressures in the region of conventionally located exit slots of from atmospheric to two- or three-tenths of dynamic pressure below atmospheric. On the other hand, the total pressure above atmospheric entering the baffle passages is also found to be reduced one- to two-tenths of dynamic head, as previously mentioned. For well-designed cowl exit slots it therefore appears that a good approximation of total drop across

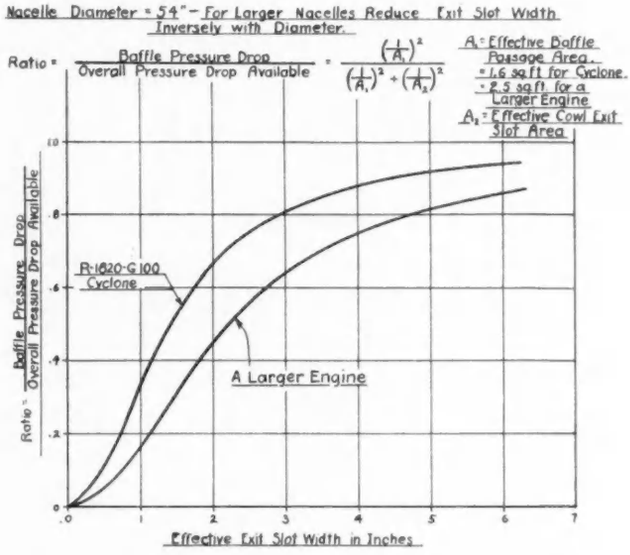


Fig. 8 – Relation of Effective Cowl Exit Slot Width to Proportion of Total Existing Cowl Drop Available for Baffle Pressure Drop

the baffle-exit-slot system is the dynamic pressure corresponding to the indicated air speed. Figs. 9*a* and 9*b* show, for the two engines, the curves of Fig. 8 each expanded into a family of curves, relating effective exit slot width to baffle pressure drop for various constant indicated air speeds. Since the internal air drag horsepower is proportional to the product of baffle pressure drop times the volume of flow, the volume data of Fig. 7 may also be applied to give a scale of internal air drag horsepower as shown in Fig. 9.

By referring to these curves, the cowl designer may provide the effective cowl exit slot width necessary at his particular indicated air speed to meet the baffle pressure drop requirements which the engine manufacturer specifies for the conditions. For exampe, if 6 in. of water baffle pressure drop is specified by the manufacturer for a given level flight operation of the Cyclone, and the airplane is to fly at an indicated air speed of 200 m.p.h. at 10,000 ft., a required effective exit slot width in the streamline position of approximately 1 in. is indicated (for a cowl of engine diameter).

For engines of large effective baffle passage area the great drag cost of an excessive exit slot width in high-speed flight is at once apparent. On the other hand, it was pointed out before that, for these engines, in order to take advantage of whatever small cowl drop exists in taxiing, take-off, and climbing, a very large opening is required for adequate cooling unless more positive means of increasing baffle pressure drop under these conditions can be provided. It appears, therefore, that controlled cooling is essential for good per-

formance with the larger engines. The same principle also applies to more moderate sized engines such as the Cyclone if the extreme in high-output airplane performance is desired, such as in military application. However, it has been practicable, by virtue of the good cooling characteristics of this engine, to restrict the cowl exit slot to quite a low compromise effective width and satisfactorily avoid the less desirable considerations involved in controlled cooling as currently developed. A factor which further makes this method practicable is the fact that the operator defeats his purpose if he seeks such a low value of baffle pressure drop as to maintain excessive head temperatures. Tests on blower-cooled dynamometer stands in which baffle pressure drop is reduced in successive stages while manifold pressure is maintained constant show that, in rated power operation, the engine output declines only slightly with increase in cylinder temperatures if these increases are moderate, but the horsepower declines more rapidly as the engine temperatures reach excessive values. It is thus possible to carry reduction of internal air drag past the point of maximum net thrust. For this reason, the engine manufacturer specifies a minimum baffle pressure drop for protracted level-flight operation such that cooling air drag is greatly reduced but, under normal conditions, no severe penalty in engine output due to excessive temperatures is experienced.

It should be mentioned that, in the curves of Fig. 9, no allowance has been included for the effect of the heat added to the cooling air on effective slot width required or on net internal air drag horsepower. At baffle pressure drops of over 6 in. of water and up to 300 m.p.h., both effects are small even under rated power operation, and are probably within the error of other assumptions made. Observation of the rise in cooling-air temperature of single and full-scale engines indicates a heat rejection to cooling air at rated power of from 12 to 14 B.t.u. per min. per b.hp. This value means a rise in temperature of the cooling air at a flow corresponding to 6 in. of baffle pressure drop at rated power of not over 50 deg. fahr. Although the resulting decrease in density would tend to restrict the flow through a given size of exit slot, it has a similar though somewhat less restricting effect on flow through the baffle passages also, since the temperature rise in question takes place entirely within those passages. Thus the relation between the two resistances does not change rapidly with rise in cooling air temperature due to heat absorption from the cylinders.

It has been pointed out[2] that the heat added to the cooling air will provide an increase of velocity leaving the exit slot and produce increased thrust. At very high air speeds, with baffle pressure drop maintained at a low value, a large drop and relative velocity exist at the cowl exit slot, which velocity will increase inversely with the square root of the density if the drop be maintained constant. This condition results in a greater value of MV, or thrust, for a given weight flow of air if heat is added than if it is not. However, it appears that ideally only about 8 thrust hp. could be recovered for a Cyclone engine at rated power at 300 m.p.h. at 6 in. of water baffle pressure drop. These calculations are exclusive of heat added from the exhaust manifold, which should be removed largely from the cooling-air passage. (Use of the exhaust energy for thrust is a separate problem.)

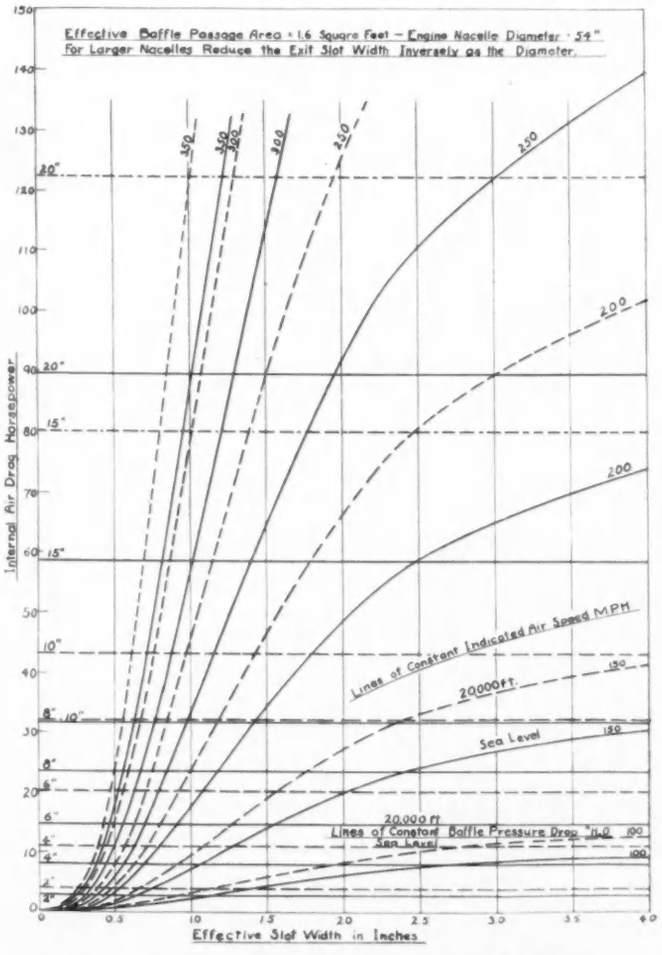


Fig. 9*a* – Relation of Effective Slot Width to Baffle Pressure Drop and Internal Air Drag Horsepower for Cyclone Installations

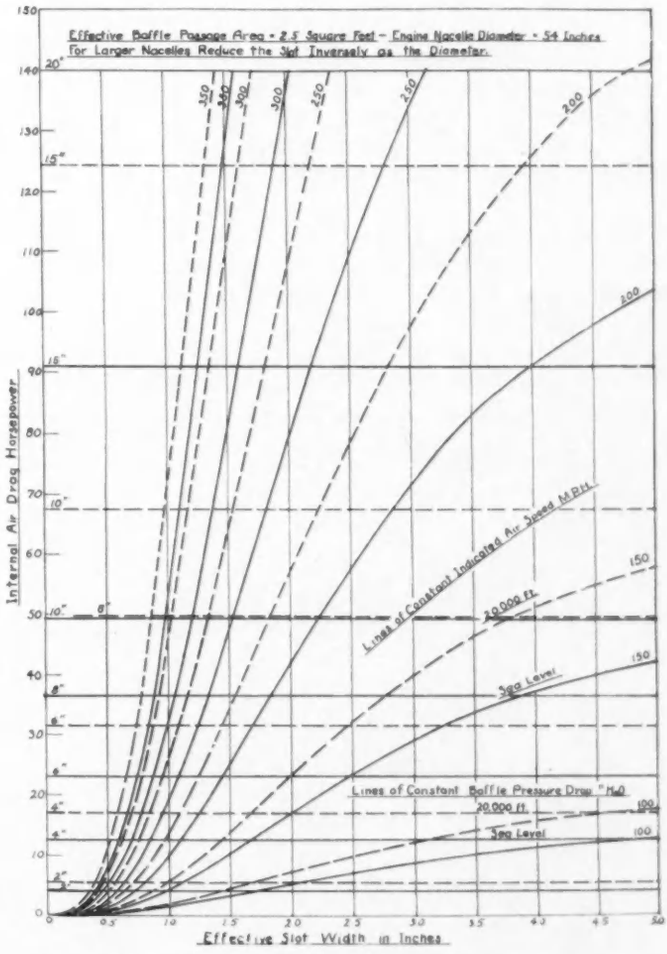


Fig. 9*b* – Relations of Fig. 9*a* for a Higher Output Engine of 2.5 Sq. Ft. Effective Orifice Area

For several modern Cyclone high-performance installations with cowling designs based on the data of Fig. 9*a*, the observed relation between baffle pressure drop and measured slot width in high-speed flight has checked the effective slot width chart of Fig. 9*a* in a satisfactory manner. On the other hand, it is very important to point out that, in the case of most cowlings existing today, the effective slot width is far less than the measured minimum value.

A surprisingly great restriction to effective exit slot width, aside from the obvious one of lack of smooth internal design, appears to be directly attributable to what has heretofore been fairly universally adopted slot design for high-speed cowlings, namely, the practice of allowing the line of the trailing edge of the cowl, produced, to be tangent to the nacelle shoulder instead of falling outside of it. Some data taken in connection with flight tests of the reverse flow and other

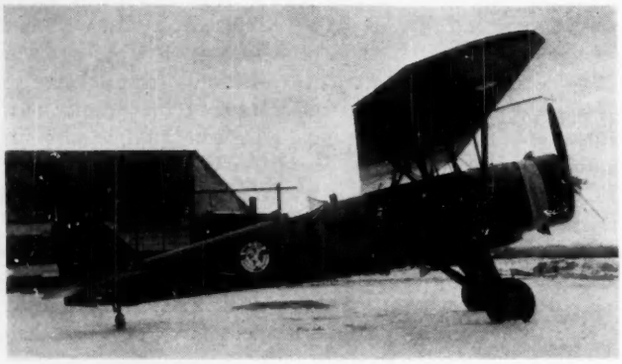


Fig. 10 – Wright Cyclone-Powered General Airplane Used for Flight Tests of the Reverse-Flow and Other Cowls

cowls on the Wright Cyclone-powered General airplane (Fig. 10) are interesting in this connection. The reverse-flow cowl data can be referred to more logically later.

Tests were made of a conventional installation in which the cowl skirt was of the same diameter as the shoulder, as

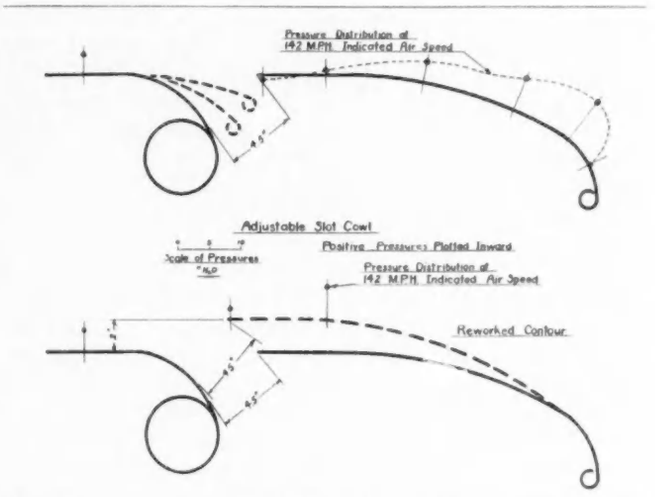


Fig. 11 – Outlines of Adjustable Slot Cowl Tested on Wright General Airplane – (*above*) with Equal Skirt and Shoulder Diameters (*below*) Same Slot Width and Shoulder, Enlarged Skirt Diameter

just described, but in which the slot opening could be varied from the inside by an adjustable shoulder consisting of overlapping strips of spring steel. Fig. 11, top, gives the outline of the cowl. Fig. 12 shows the baffle pressure drops and the pressures behind the engine observed for three slot sizes, 1½ in., 3 in. and 4½ in. over a range of indicated air speed in level flight. The reduction in pressure behind the engine and in pressure drop across the slot with increase of measured width from 1½ in. to 4½ in. is relatively small. Applying the baffle pressure drops to the chart of Fig. 9*a*, the effective slot width in the 4½-in. position is still only about 1½ in. By reworking the cowl as indicated by the dotted lines at the bottom of Fig. 11, still retaining a measured slot width of 4½ in., the effective slot width was doubled, apparently by giving the exhausting air a better opportunity to parallel the

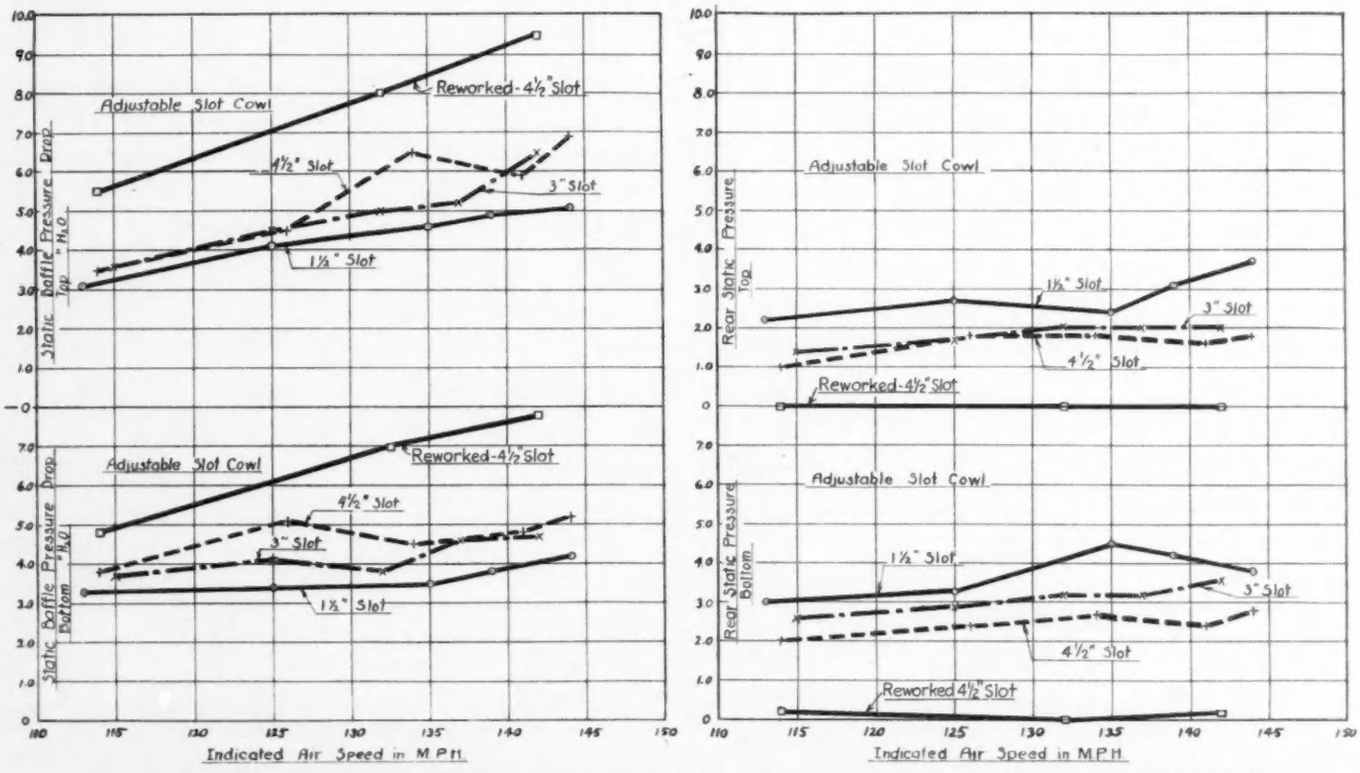


Fig. 12 – Comparison of Baffle Pressure Drops and Pressures behind the Engine Observed in Level Flight with Exit Slots of Fig. 11

slipstream instead of being forced to intersect it. In spite of the reworked cowl being 4 in. larger in diameter, carefully conducted speed checks showed no measurable difference in speed within the limited maximum speed of test (143 m.p.h. indicated) which was unfortunately a very low value for such testing.

The exit-slot designs just discussed are obviously not representative of recommended practice and are shown only to demonstrate a particular point applicable in varying degree to many present-day installations. An easier fairing of the shoulder into the line of the fuselage or nacelle, and extension of the overhanging cowl skirt further to the rear, bring measured and effective widths closer together and should further reduce drag. Unless this work is done, a rational preselection of exit-slot width for any specified speed is very speculative.

In continuation of the tests just discussed, extension of the large-diameter skirt to the rearward reducing the measured slot width to 2 in. brought the effective and measured slot widths together. This result thus gave a greater effective slot width than that obtained using the original 4½ in. slot with skirt diameter equal to shoulder diameter (1½ in. effective width).

Through the work of L. T. Miller, engineer in charge of experimental flight testing, an effort is being made to develop a suitable technique of recording the flow pattern at the cowl exit slot in flight. Fig. 13 shows flow patterns obtained for the external flow by spraying a mixture of carbon tetrachloride, oil, and lamp black just outside the surface of the cowl nose and depositing it on longitudinal white plates projecting radially from the slot region at top and bottom of the cowl. Patterns have been taken thus for ground, climb, and level flight at 5000 ft., the photos shown representing the latter condition for slots similar to the three designs just discussed. The restricting effect of the faired in 4½-in. slot is indicated clearly in the figure.

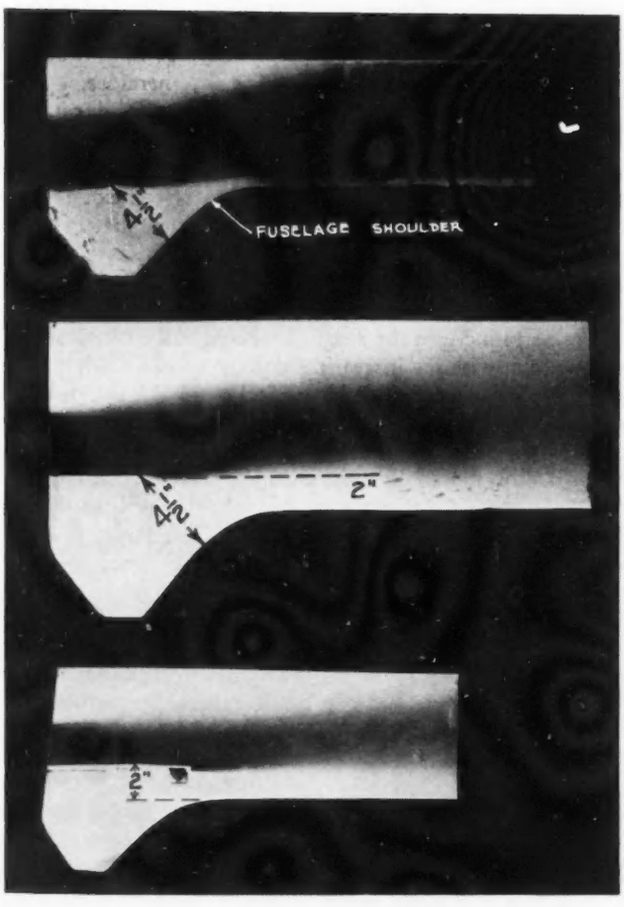


Fig. 13 –External Flow Patterns

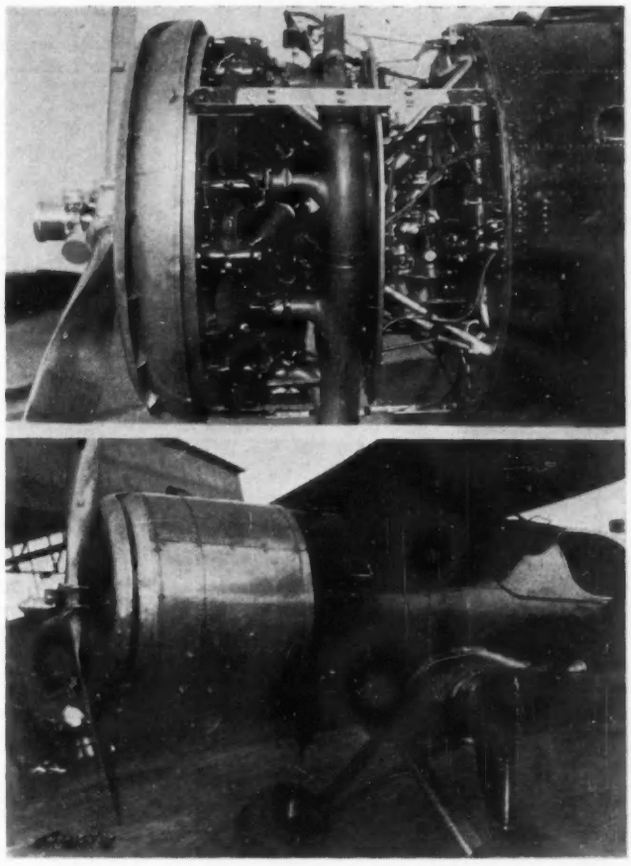


Fig. 14 – Two Modifications, *A* and *B*, of the "Reverse-Flow" Cowl, Installed

Efforts to record the flow from inside the cowl have met with more difficulty. The slot being fairly close to the engine, conditions vary around its periphery depending upon whether a location is chosen behind a baffle passage, behind a baffle itself, behind an exhaust stack, and so on. Such flow recordings do show, however, that turbulence outside the exit slot is such as to cause the apparent recorded boundaries of the external and internal stream flow to overlap. Thus it appears that the present photos must be taken qualitatively rather than quantitatively.

The desire to increase the low baffle pressure drops existing at very low airplane speeds in order to obtain the velocity of cooling air necessary for increased heat rejection has challenged the ingenuity of engine and aircraft investigators for many years. There are three main avenues of approach to this problem:

a. Use of the propeller slipstream to induce a depression after the baffle passages.

b. Use of the propeller slipstream to build up a pressure before the baffle passages.

c. Use of a fan for either purpose.

Use of the first method does not appear to promise further progress at the very low air speeds of taxiing and take-off than that already accomplished by application of the flared trailing-edge principle, the chief advantage of which is the creation of a very large effective exit slot area.

One device intended to accomplish this purpose which has caused considerable speculation as to its soundness, is the so-called "reverse-flow" or forward-exit-slot cowl. The Wright Aeronautical Corp. conducted a series of tests on this type of cowling beginning in early 1936. The cowlings were installed on a Cyclone G engine in the WAC General airplane.

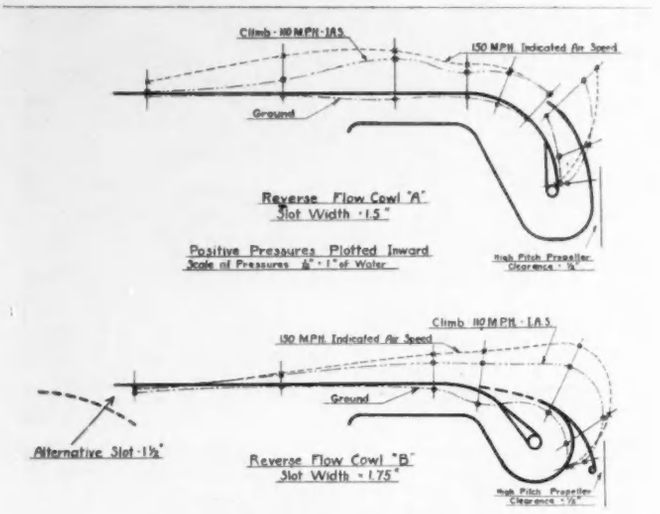


Fig. 15 – Outlines and Observed Pressure Distributions of Two Reverse-Flow Cowls, *A* and *B*, Tested on WAC General Airplane

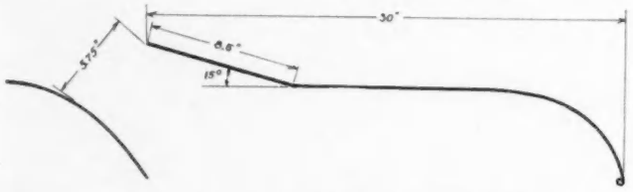


Fig. 16 – Outline of DC-2 Cowl with a 15-Deg. Flare Compared in Flight Tests with Reverse-Flow Cowl *A*

Fig. 14 shows two modifications of this cowling, *A* and *B*, installed. Fig. 15 shows the outlines of these two cowls on which considerable test data in ground, climb, and level flight operation were obtained.

The air was passed one way through the entire fin-baffle passage system of the engine, and then reversed into an exit passage concentric with the engine as indicated. Marked on the outline drawings of these cowls are the pressure distributions obtained on the ground, in 110 m.p.h. indicated air speed climb and in 150 m.p.h. indicated air speed level flight. Contrary to popular understanding, it will be observed immediately that, with neither type of cowl shown, was any negative pressure obtainable over the exterior of the cowl surface under stand-still conditions. This finding eliminates one of the principal reasons for interest in the forward slot, namely, utilization of a high value of external negative pressure for ground and taxiing cooling. It, therefore, appears that, at low air speeds, just as large an effective exit-slot width is required with this type of cowling as with the slot in the conventional location behind the engine. The ground-cooling performance of the two cowls tested showed that the one with the discontinuous or overhanging slot contour gave the better cooling, although both were quite inferior in this case to a conventional cowling with large exit slot and trailing flare because of the high resistance of the exit passage, reducing the effective exit slot width to a low value. The same may be said of climbing performance.

The cowl of the upper sketch of Fig. 15 was compared in flight with a Douglas DC-2 cowl having a 15-deg. trailing-edge flare added and a very large exit slot as sketched in Fig. 16. In both cases the airplane was operated under carefully controlled engine operating conditions to maintain comparability of power output, speeds and pressures being observed over a range of manifold pressures. Plotted against indicated air speed, the pressures at the rear of the engine and also the baffle pressure drops obtained with these two cowls are included in Fig. 17. The high resistance of the reverse-flow cowl exit passage is at once evident since, in spite of the high negative pressure outside of the exit slot, a substantial positive pressure still existed behind the engine, a large portion of the available pressure drop across the cowl being required to force the air through the exit passage.

From these results and the data on pressures over the cowl nose, a rough comparison can be made, sufficiently accurate for our purpose, of the internal air drag horsepowers at a given air speed:

	DC-2 Cowl with Flare	Reverse-Flow Cowl *A*	Reverse-Flow Cowl *B*
Pressure in Front of Engine, in. H_2O	7	8	7.5
Pressure at Rear of Engine, in. H_2O	— 1	3	4.25
Baffle Pressure Drop, in. H_2O	8	5	3.25
Volume, cu. ft. per min.	18,000	14,000	11,000
Static at Exit Slot, in. H_2O	—	— 7	— 7
Drop Across Exit Passage, in. H_2O	—	10	11.25
Baffle Plus Exit Drop, in. H_2O	8 (minimum)	15	14.50
Internal Air-Drag Horsepower	22.5	33	25
Horsepower Difference	11.5	8	

Assuming very conservatively that all the pressure drop across the exit passage of the reverse-flow cowl represents wasted internal energy, whereas only that across the baffles represents wasted energy in the case of the flared rear exit cowl, there is, at this low air speed, an excess of only 12 internal air drag hp. in the case of the reverse-flow cowl. Comparing the air speeds of the two cowls, on the other hand, from the upper curves of Fig. 18, a difference of from 4 to 6.5 m.p.h. in speed at 5000 ft. is indicated, and an increase in manifold pressure for the reverse-flow cowl of about 2 in. of mercury or 60 hp. is required to equalize the speed. The engine was operating in the region of 540 hp. for the flared cowl and 600 hp. for the reverse-flow cowl at 135 m.p.h. The results were checked with two different types of air-speed indicator in two entirely separate series of comparison flights. It, therefore, appears that the contour of the reverse-flow cowl nose *A* caused a tremendous external-drag disadvantage.

The cowl was then reworked to form *B* (Fig. 15) to reduce the drag and to improve the smoothness of the exit passage. The external contour was changed to simulate that of a conventional cowl, and the exit slot was faired in carefully and widened from the original 1½ in. to 1¾ in. Unfortunately, space was too limited by the proximity of the propeller to the engine to increase the slot further and still make a smooth nose contour without increasing diameter. The exit passage was smoothed up considerably as indicated. An annular insert also was provided with which the cowl could be converted into a conventional type, the forward slot blocked off, as shown by the dotted lines in Fig. 15, and for use with this nose a smooth 1½-in. exit slot was provided well to the rear of the engine as indicated. The speed performances of the cowl with and without the forward exit slot were compared as indicated by the lower curves of Fig. 18 and, as far as could be determined with the slow-speed equipment, the speeds were not identical but fairly comparable. However, examining the pressures of Figs. 15 and 17, it is evident that, at the same air speeds as the first cowling, the resistance of the exit passage of the modified cowl was now substantially *increased* instead of reduced as had been intended, passing only about three-quarters of the flow of the cowl *A* at the same pressure drop. As the exit passage was improved somewhat internally, this restriction again appears to be due to the fact that the exit slot, even though slightly wider, was carefully faired into the nose contour instead of being of the overhanging type.

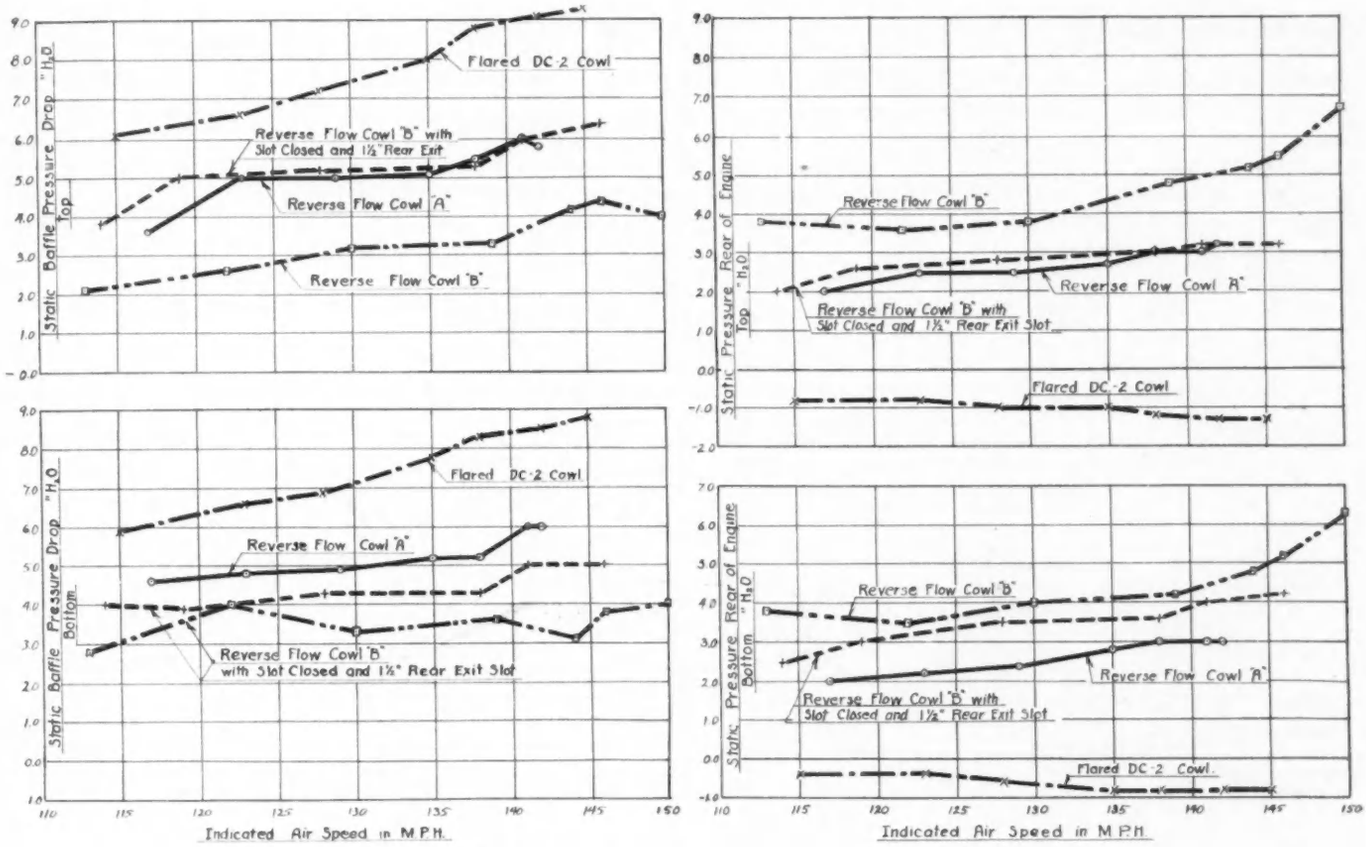


Fig. 17 – Comparison of Baffle Pressure Drops and Pressures behind the Engine Observed in Level-Flight, Reverse-Flow Cowl Tests

The test results would indicate that:

1. For ground, taxiing, and take-off cooling with engines of large effective baffle passage area, the forward slot is not likely to be more effective than a conventionally located slot with cowl trailing-edge flare, and should equal that performance only if the forward slot is large and of discontinuous contour which, if allowed to exist in high speed, probably would cause a prohibitive drag. If an exit opening of the type described for ground-cooling is not feasible, other means of augmenting baffle pressure drop on the ground should be provided.

2. For high-speed flight the forward slot must be faired carefully into the anti-drag nose contour.

3. For climbing, the forward slot can help augment baffle pressure drop, provided the resistance of the exit passage and of the air path through the engine is kept low. For engines of the type under discussion, this arrangement appears impracticable for a forward-entrance-forward-exit type of cowl. If the air is brought through part of the baffle passages in one direction and then reversed and passed through the remainder of the baffle passages on the return path, the tortuous approach to the cowl exit slot in the nose can be avoided, but the baffle resistance has then been increased and whatever drop is created by the cowl must be divided between the two sets of baffle passages in series. If, on the other hand, the diameter of the cowling nose is increased, a return path concentric to the engine having reduced resistance can be constructed but, in order to provide a low-resistance passage, the shape of the cowl nose becomes such that the slot could just as well be located behind the engine instead, and would still be in the low-pressure region of the curving cowl contour. Such a rear slot probably would be superior because this arrangement would represent the lowest resistance of path possible, and would certainly be more practicable.

4. For these reasons the promising outlook for the forward-slot cowl appears to lie not in the forward entrance type but in the one-way-flow type wherein the air enters from the rear, passes forward through all the baffle passages in parallel, and thence out a forward slot, controllable or otherwise, the con-

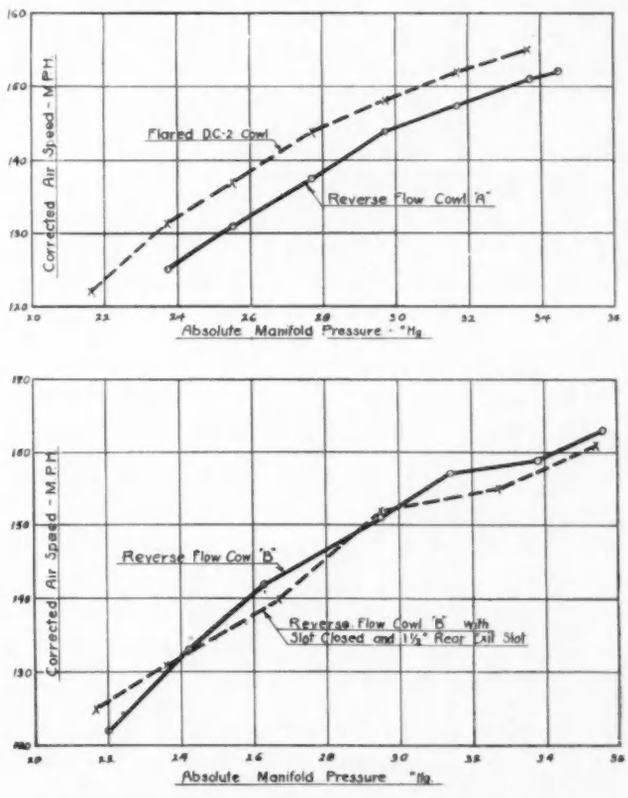


Fig. 18 – Level-Flight Speed Comparisons from Reverse-Flow Cowl Tests

ventional frontal opening being blocked off and even streamlined with a spinner if considered desirable. Such a cowl could be limited to engine diameter. The resistance, at least as far as the baffle-exit slot system is concerned, could be made comparable to that of conventional cowlings, if the slot were sufficiently large. This structure was disclosed for radiator cooling in a U. S. Patent issued to Gilmore in 1924. A British patent to Ellor and Paravacini in April, 1936, disclosed cooling-air entrance openings in the high-velocity portion of the propeller slipstream in the leading edges of the airplane wings to get the increased baffle pressure drop desired at very low air speeds. It appears possible that such a device could utilize a well-faired moderately sized low-drag nose slot under all conditions and achieve both good cooling and good drag performance, subject to careful scrutiny of and provision against excessive losses in the wing ducts. It is understood that a British installation already has been constructed on this principle, but the writer has no information as to requirements, dimensions, or performance. Such a cowling appears as an attractive speculation for very high-speed installations for radial engines of present baffle-pressure-drop characteristics. A streamlined nacelle shape could be approached, and any possible high-speed drag penalty for the conventional deviation from this shape thereby eliminated. Suitable reversed baffles to cool all portions of the cylinders without turbulence and to pass a somewhat higher flow than at present, as discussed earlier, would have to be provided.

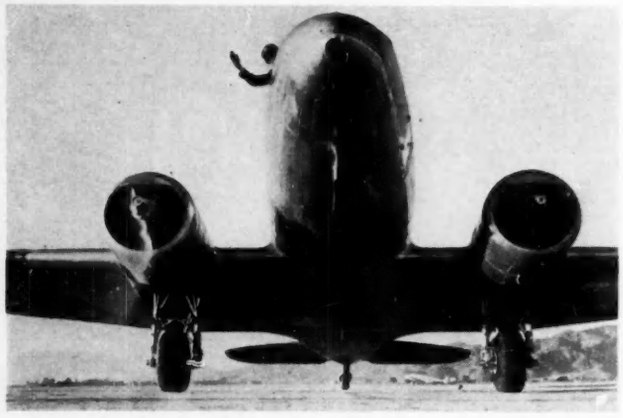


Fig. 19 – Douglas DC-3 Showing Segmental "Eyelid" across the Top of the Cowl Opening

Aside from the just-described one-way-flow forward-slot cowl principle, use of the slipstream to build up a pressure before the engine at low air speeds always has involved use of the conventional large-diameter front opening. The effect of the propeller in building up pressure on the engine is well known. In direct-drive installations with smaller propellers and higher blade velocities, we have observed repeatedly that, in climb, pressures in front of the engine greater than the dynamic pressure corresponding to the air speed are obtainable. This condition is for Cyclone engine installations with cowl entrance openings of about 80 per cent of engine diameter. With geared installations, on the other hand, no such high pressures are obtainable, only fractional baffle pressure drops being obtainable on the ground and, in 100 m.p.h. climb, only 3 to 4 in. of water drop are obtained with the lowest values of baffle pressure drop at the top of the engine. Improvement in ground- and climb-cooling is observable for each increment of enlargement of the entrance opening to

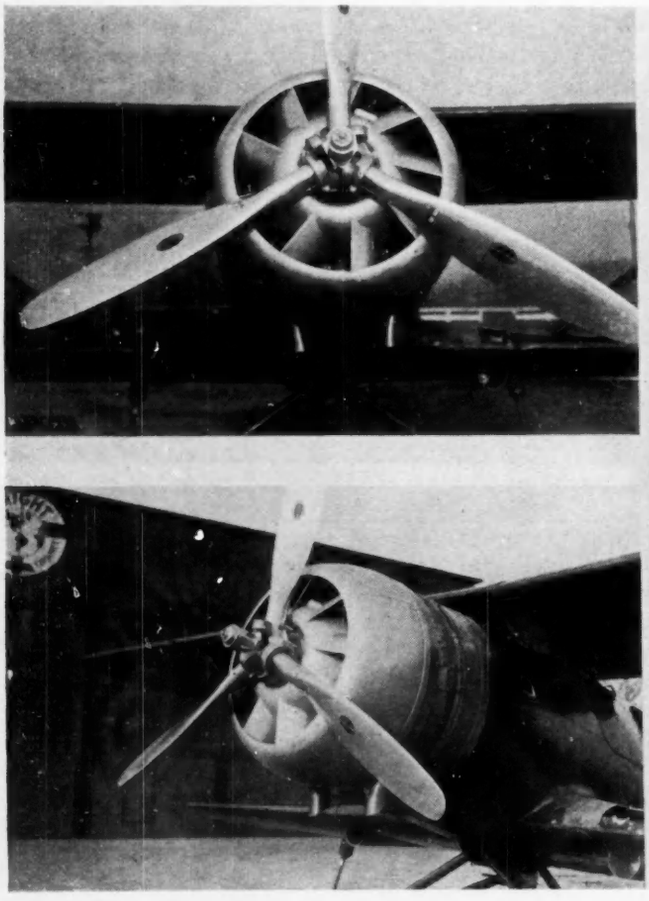


Fig. 20 – Several Attempts at Converting to Pressure the Velocity Energy of the Air Displaced in a Tangential Direction by the Propeller Shanks Applied to a Cyclone Engine on the WAC General Airplane

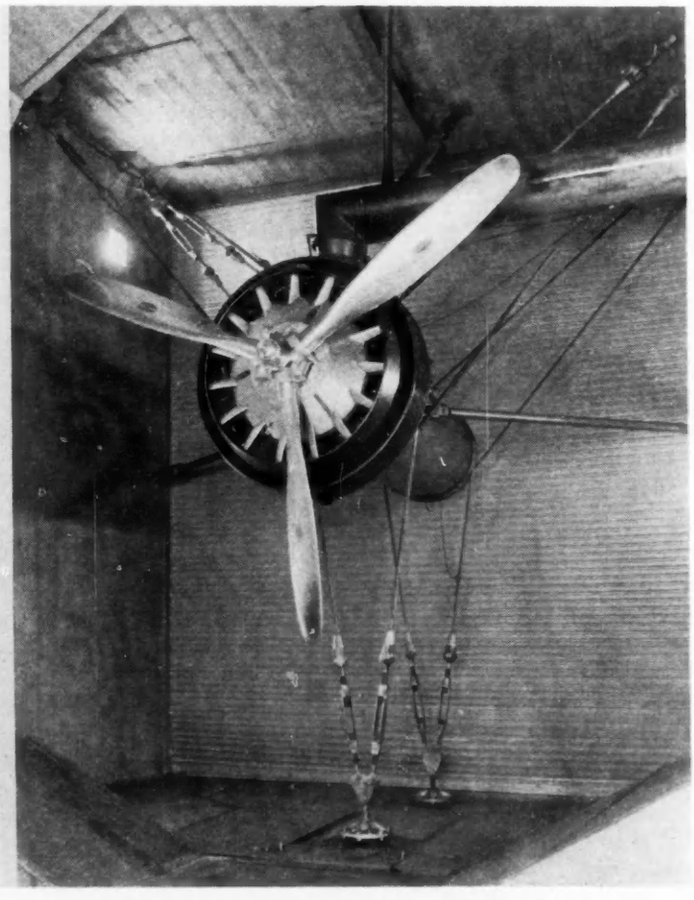


Fig. 21 – Set-Up for Fan-Cooling Tests on 16:11 Gear Ratio Cyclone

values as high as 85 per cent of engine diameter. This condition is probably due to the very poor propeller-blade shape near the hub with propellers of current design.

The phenomenon of reduced baffle pressure drop at the top of the engine in climbing angles of attack is one which the engine manufacturer has appreciated for several years. The cooling air escapes out over the nose of the cowl at the top, apparently due to the shifting forward of the negative pressure region on the outside of the nose contour, and the local velocities in front of the engine are such that the static pressure in front of the engine at the top is reduced severely. In the case of cowls with an extreme opening diameter this outflow has been observed to be quite large, resulting in inferior cooling of the upper cylinders and also in a tendency for the escaping warm air to enter the carburetor air scoop and further reduce engine output in climbing. In the belief that the benefits of the large entrance opening could be retained with the outflow reduced to a normal amount by application of a segmental "eyelid" across the top of the cowl opening, flight tests were conducted on such a device and its use recommended by the Wright Aeronautical Corp. Tests also indicated, however, that this type of correction can be overdone, resulting in higher front and rear temperatures of the upper cylinders instead of improvement. The first commercial application of these results was on the Cyclone powered Douglas DC-3 with which nearly every one is now familiar, as shown on Fig. 19.

Another interesting possibility for building up pressure in front of the engine and equalizing pressures at the top and bottom is to convert to pressure the velocity energy of the air displaced in a tangential direction by the propeller shanks. Even with geared engines, if we consider the peripheral velocity of the propeller shank in front of a cowl opening of normal diameter, it would appear that several inches of water would be obtainable if the tangential velocity energy could be converted to pressure without too terrible an efficiency. Fig. 20 shows the latest design of several recent attempts in this direction applied to a Cyclone engine on the W.A.C. General airplane using the same cowl and exit slot discussed in connection with Fig. 11 top. The propeller is geared, 16:11 ratio. Initially, a set of radial vanes was installed with not very spectacular results. Subsequently, it was realized that the vanes should be offset from the radial position in the direction of rotation. Although highly turbulent, the general air movement is described by an axial and a tangential component. By the time it reaches the plane of the vane edges along a tangential path the revised-type vanes shown are more effective in turning the air axially against the engine disc, with a minimum of other components remaining. The vanes shown in the photograph were attached to the cowl by machine screws and elastic stop nuts. Comparative observations of baffle pressure drop and cooling performance were made with and without the vanes on the ground, in climb, and in level flight under carefully controlled engine operating conditions. At the same mixture strength (as indicated on a Cambridge exhaust-gas analyzer), cylinder temperatures on the ground with the vanes were reduced about 70 deg. fahr. at 1800 engine r.p.m. Baffle pressure drops on the ground were increased from fractional to about 1.8 in. of water. Pressure readings also were taken just as the ship left the ground at 60 m.p.h. take-off, 1000 hp., 2200 engine r.p.m. Baffle pressure drops from 4 to 6 in. of water were obtained.

In climb at 110 m.p.h. no benefit was derived from the vanes either in baffle pressure drop or in reduced cylinder temperatures.

A careful check of the speed in level flight with and without the vanes at several manifold pressures and at closely identical engine operating conditions including fuel-air ratio, indicated no change in speed performance and no change in baffle pressure drop. For some reason, apparently related to turbulence, the head temperatures were reduced about 30 deg. fahr. at the higher powers. Maximum indicated air speed was 144 m.p.h.

It thus appears that the device as tested on the Cyclone constituted a form of automatic controlled cooling with no moving parts in that, with the restricted fixed exit slot, baffle pressure drops in level flight were moderate, but some additional pressure boost was provided for the ground and take-off operation without the requirement of any movable mechanism. The weight of the vanes and nose disc was 12½ lb. Whether such a device will provide similar improved ground-cooling and automatic control on more powerful engines of the same diameter but having slower propeller speeds and larger effective baffle passage areas has not yet been established. Such an application is in preparation for test.

With regard to fan-cooling for raising baffle pressure drops at very low air speeds, this method has not met with general favor or interest for use in conventional installations. However, if submerged or full-streamlined installations in high-speed airplanes come into consideration, fan-cooling may become a very satisfactory type of automatic controlled cooling. Full-scale engine laboratory tests with a fan attached to the propeller of a 16:11 gear ratio Cyclone (Fig. 21), have demonstrated that at least 3 in. of water boost were available at take-off r.p.m. with the blades designed to "float" or produce no boost at the higher flows resulting naturally from high-speed cruising flight. The propeller manufacturer is the logical one to develop the propeller-driven type of fan, as he is in a position to foresee the changes in hub design and to standardize on methods of fan attachment. Up to the present the rapid obsolescence of propeller hub designs also has obsoleted the fan by the time its endurance characteristics have been safely developed.

For submerged or streamlined engine installations a variety of engine-driven fan-cooling applications is, of course, possible, though in all cases the fan should be designed to provide reduced boost at the higher air flows corresponding to high-speed flight.

In conclusion, it appears that the progress to date in cylinder cooling of air-cooled radial aircraft engines, though considerable, has not yet reached the point of diminishing returns. As in the past, improved installation will result from a high degree of cooperation between the independent investigator, the airplane manufacturer, and the engine manufacturer.

I should not like to close without expressing sincere appreciation of several years of invaluable assistance from H. V. Shebat, our test pilot, without whose long experience and meticulous care in maintaining comparable flight test conditions a large portion of our research efforts would be without value.

Discussion

Compares British and American Practice

—F. M. Owner

Bristol Aeroplane Co., Ltd.

WE at Bristol are very happy to corroborate many of the views that Mr. Campbell puts forward. As a proof of this agreement, we may perhaps be allowed to cite a cowling diagram given by A. H. R. Fedden in a paper read before the Royal Aeronautical Society around 1929 or 1930, in which the engine nose was extended and the exit-slot contours conformed to Mr. Campbell's suggestions.

It is, however, a little astonishing to us that the American engine

manufacturer is prepared to leave the responsibility for cowling design to the airplane builder. British practice is tending rapidly towards the supply of powerplants complete with exhaust system and engine cowling, and the aircraft manufacturer's collaboration in the design of the remainder of the nacelle fairing is sought carefully.

A further point of difference lies in the basic dissimilarity between the typical American cylinder and the "Bristol" design in the matter of exhaust-port location, the forward-facing "Bristol" ports tending to lessen the circumferential temperature gradient, and to alter the optimum baffle characteristics.

With this reservation in mind, I should like to strike a more cordial note by agreeing heartily with Mr. Campbell on the following points:

1. Our opinion of the "reverse-flow cowl" is rather similar, in that the discharge of appreciable quantities of air at the nose of the cowling may alter greatly the pressure distribution.

2. We agree entirely with his views on the extension of fin area, and would point out that the relative insensitivity of the air-cooled fin to mass flow, whilst rendering increase of cooling-air velocity uneconomical, also plays a very useful role in permitting very great retention of cooling ability at high altitudes. Practical proof of this condition has been obtained on the "Bristol" high-altitude-record aircraft.

3. As just mentioned, we agree that the cowl tangent should lie outside the afterbody by an amount sufficient to accommodate the cooling airstream, and would like to suggest in this connection that tests of efflux angle (by indicator vanes or similar means) might furnish a useful guide to exit-slot-contour design.

4. Improvement of the sections of propeller blade roots is most desirable.

As regards Mr. Campbell's reference to British wing-entry installations, I regret that I have no data to offer on these types, but have every reason to believe that their aerodynamic efficiency is satisfactory.

Awaits Further Data on Turbulence Drag

—Weldon Worth
U.S. Army Air Corps, Wright Field

I BELIEVE that Mr. Campbell's paper is very important in pointing out the present deficiencies and the basic factors that suggest the desirable trends of future development. One of the most important factors appears to be the 35 to 40 per cent of the nacelle drag that results from the turbulence in front of the engines, and I am sure the industry and certainly the Air Corps are waiting with interest the test results that should be forthcoming if Mr. Campbell reduces to practice some of the obvious solutions stimulated by his analysis.

One question on which I would like to have an opinion is the reliability, consistency, and accuracy that Mr. Campbell thinks can be obtained in measuring baffle pressure drops as an index of cooling. In other words, the turbulence of the orifice plate decreases the pressure drop required to maintain a constant rear spark-plug temperature by 2 to 4 in. How closely will flight conditions duplicate the simulated turbulent conditions?

Exit-Slot Location on In-Line Engines

—A. T. Gregory
Ranger Engineering Corp.

IN Fig. 11, Mr. Campbell sheds some most interesting light on the design of the cowl exit slot. Increasing the efficiency of the slot should increase the overall efficiency of the cooling system and thus reduce drag for a given degree of cooling. The airplane speed, therefore, should not have been reduced by the re-worked cowl unless cooling air flow was actually raised.

The location of the exit slot appears to be less of a problem for the in-line engine than for the radial. If the slot is at the rear of the engine as in most installations, the cooling air follows a direct path from the cylinders to the outside of the cowl; if it is at the front, a single turn is required which need not be restricted in size. If it is of advantage to have the path of the cooling air as short as possible from the cylinders to the cowl exit, the slot may be placed at about the center of the engine.

On the other hand, the slot location may be governed by other considerations rather than by accessibility around the engine. The fuselage may have a perfectly streamlined form with an in-line engine installation. In such cases it has been found most desirable to locate the exit slot slightly forward of the maximum cross-section of the fuselage.

Because of the shape of the in-line engine, the cowl exit slot need not extend completely around the fuselage or nacelle. Better results may be obtained with a relatively wider gill adjacent to the cylinders.

It is a little surprising to see the suggestion to take air in at the leading edge of the wing and use the forward exit slot. Duct sizes probably would have to be large to reduce flow losses to a minimum. When this arrangement is accomplished, however, there is the further question of the effect of removing large quantities of air from the region in front of the wing on the lift characteristics. It is doubtful if this removal can be done without affecting seriously the performance of the airplane.

The use of vanes to build up pressure at the front of the engine is most interesting. Similar results have been obtained in in-line engine installations.

With regard to the use of a fan, it may be necessary at some future date to resort to such cooling means. Much has been said against the fan, but its use would place in the hands of the engine manufacturer the means of insuring the satisfactory operation of the engine under all operating conditions independent of the installation. If, as, and when the cooling problem becomes really critical on the air-cooled engine using all other means without the fan, it seems logical to expect such a development, and possibly even before such a time.

Some Factors Controlling Part-Load Economy

(*Concluded from page* 514)

crankshaft, which will prevent wandering of the ignition timing due to elasticity and vibration of the engine parts. Timing wandering does not permit the use of the maximum advance for economy.

Fig. 8 – Ignition Timing Variations – Due to excessive clearance in the fitting of the distributor drive, variations in spark timing may reach prohibitive proportions. In this particular job, matters were aggravated by a resonance phenomenon in the range of 1250 r.p.m.

Today automobile engines have reached a point where it is very difficult, by a single improvement, to render any decidedly better performance. In a general way we have adhered religiously to the idea that improvements can be had only by checking one change at a time – and upon the results obtained deciding whether the change is worth the expense or not. However, when we are dealing with very fine points, such a procedure stops further improvement because any suggested change points out that it is not worth the chances taken in making it. But, if we make several changes at the same time, each change being based on a logical thought and with logical reasons why the change should be made, the final result will be very apparent and worth while. In my opinion we have reached that point today, and further improvements in engine economy on a conventional engine will be made only by several well-directed moves concerning small details.

Discussion

Fuel Economy During Idling Conditions

—V. C. Young
Wilcox-Rich Division, Eaton Manufacturing Co.

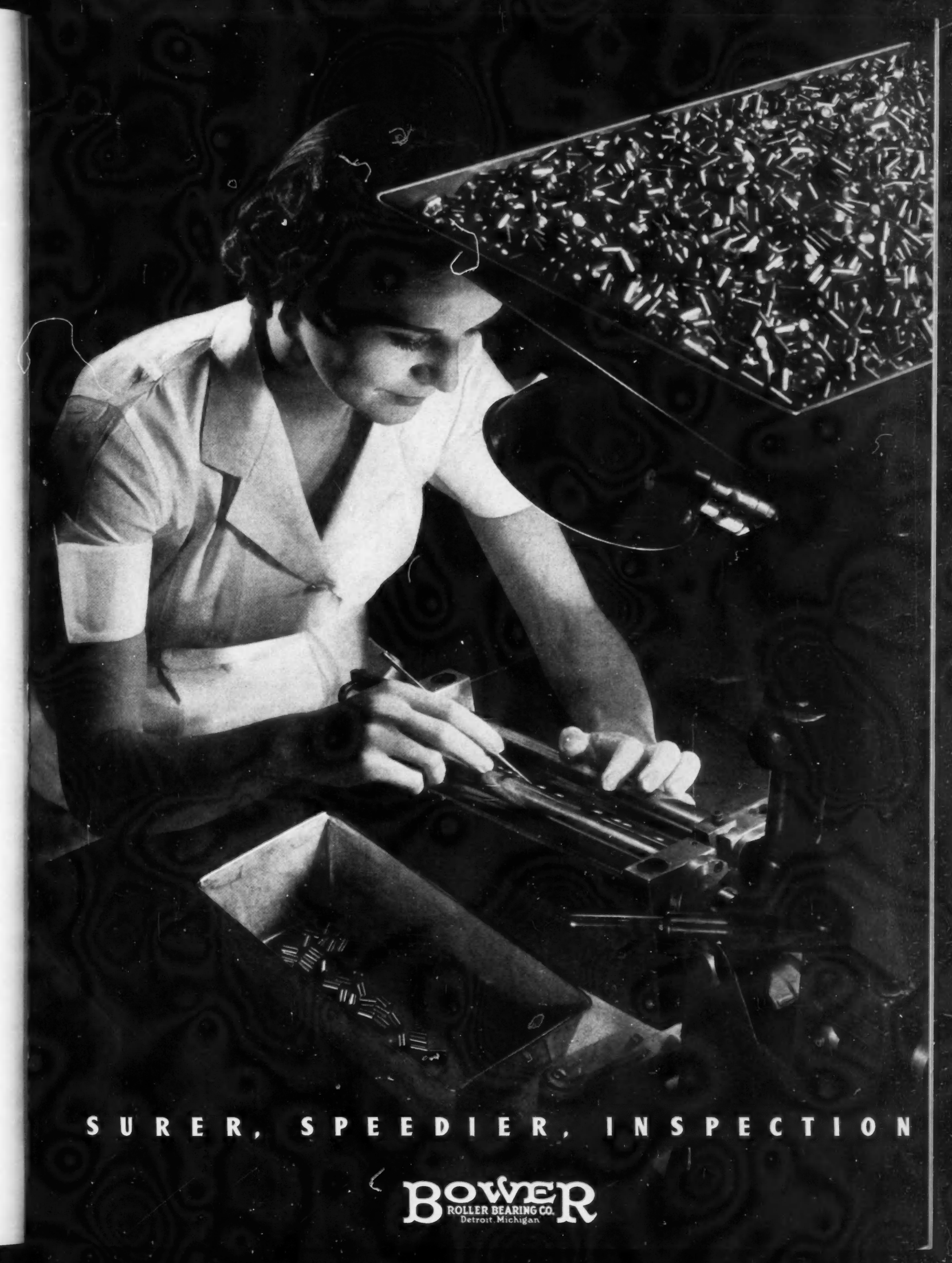
IT is suggested that more study be given to idle or just-off-idle conditions, although this effect is only under certain types of operation.

The fact is that the engine operates under closed-throttle or what we would normally call idling conditions more often than is realized, since every time the gear shift is made, or the throttle is closed to coast to the stop, or an intermediate slow-down occurs, the engine immediately is operating under a fuel consumption that is proportional to that obtained when idling normally.

Under commercial operation where these conditions were exaggerated by encountering a great many gearshifts per mile of operation, actual fuel economy was improved some 6 per cent over a period of time by careful attention to idle settings.

Buick "41" Spring Rate Quoted in Wrong Terms

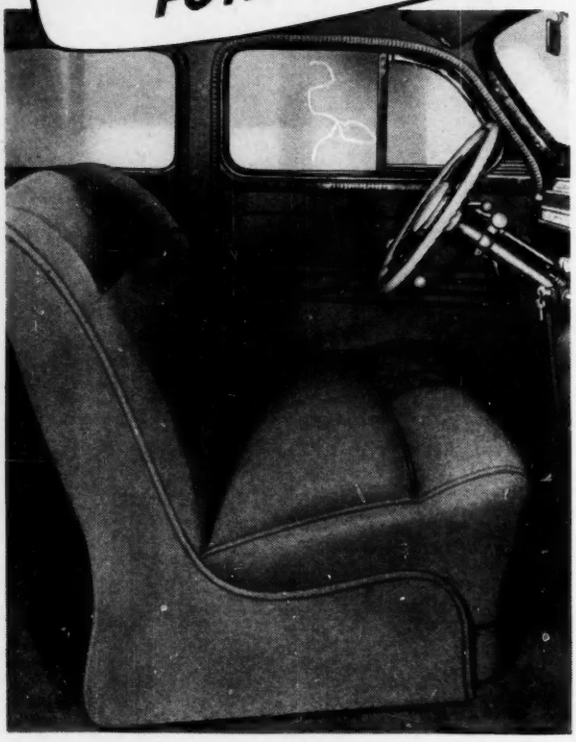
The rear spring rate of the Buick "41" was quoted as "being 129 against 150 *per min.* in 1938," in the first sentence on p. 448 of Austin Wolf's article, "Trends in Design of 1939 Cars," published in the November, 1938, TRANSACTIONS SECTION of the S.A.E. JOURNAL. The rate, of course, should have been given as being 129 against 150 *lb. per in. deflection* in 1938.


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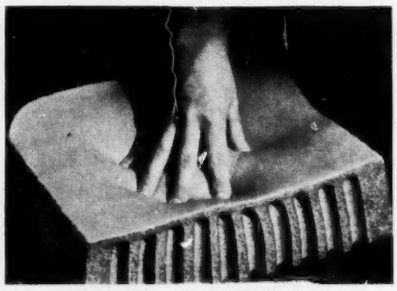
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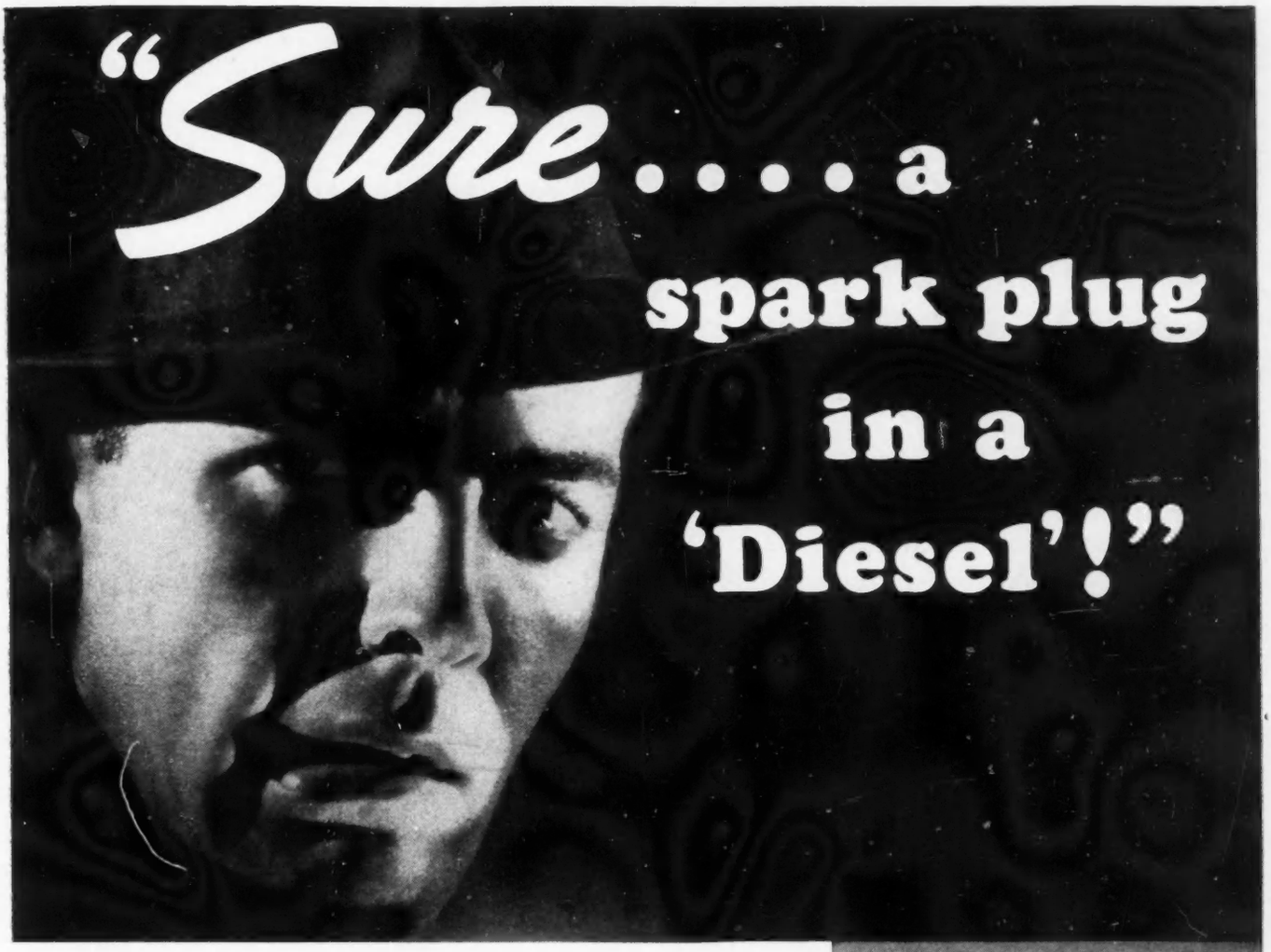
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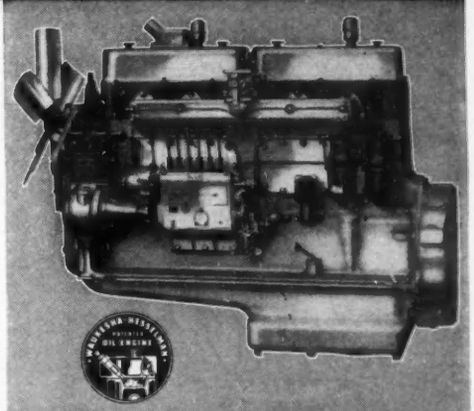
Spark plug ignition starts a Hesselman Engine easier than a gasoline engine—compression is moderate and gasoline is sprayed directly into the intake.

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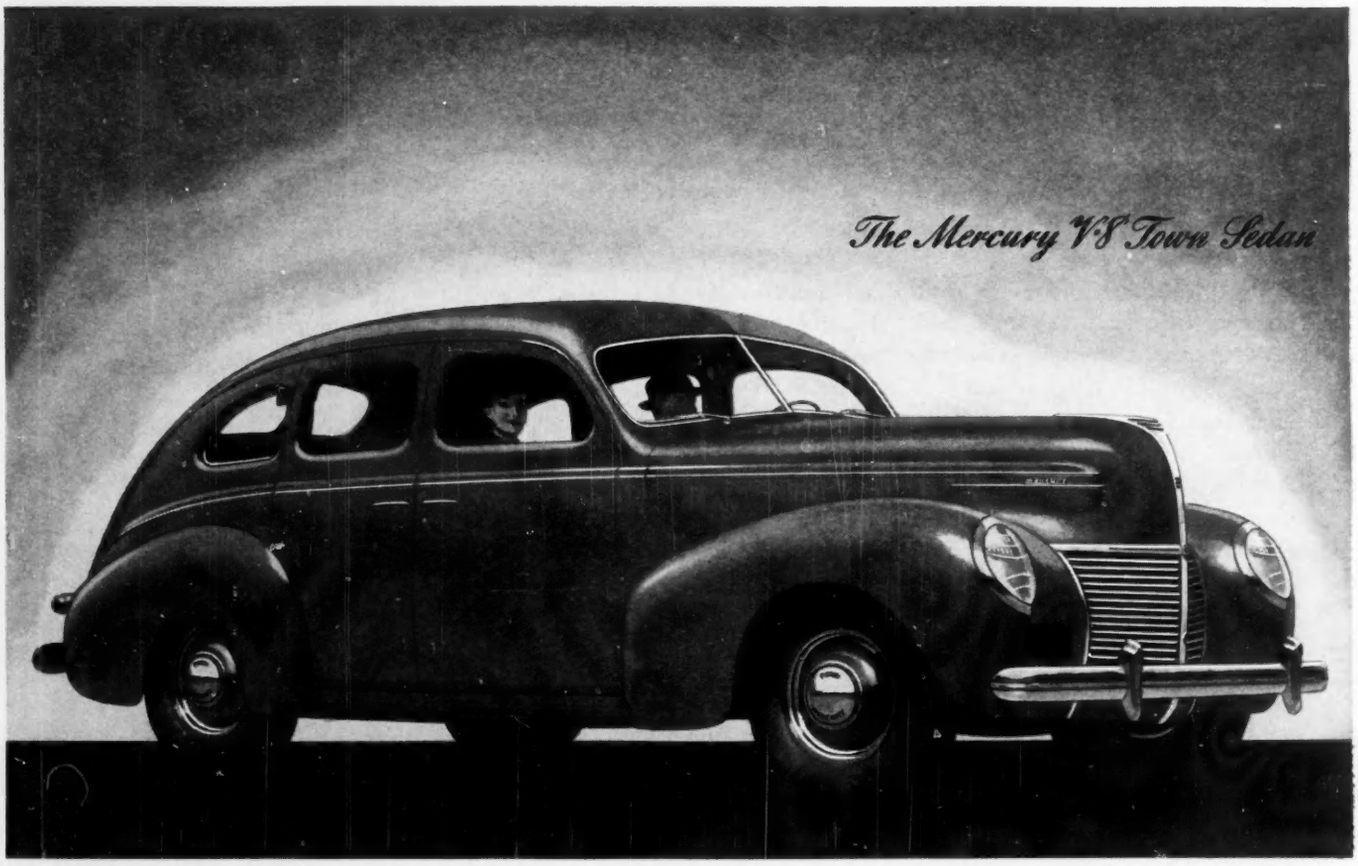
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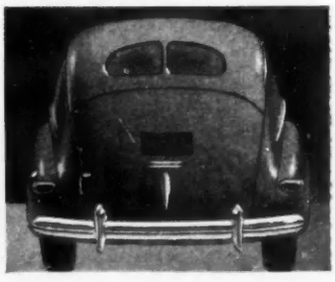


The Mercury V-8 Town Sedan



SOME OF THE PLUS-VALUE FEATURES OF THE NEW MERCURY 8

Large bodies • 116-inch wheelbase • New powerful V-type 95-hp. 8-cylinder engine • Positive, easy-action *hydraulic brakes* • Luxurious interior appointments, including new soft seat construction • Modern streamline styling derived from the Lincoln-Zephyr • Evenly distributed weight and center-poise design give "cradled" riding comfort • Scientific noise control that insures extremely quiet operation • Four beautiful body types and a choice of eight colors.



ONE WILL REACH THE HIGHEST RANK



EWING GALLOWAY

. . . for one has all the qualities of a leader

Ball bearings, too, look very much alike. Yet actually they differ in respects that affect their future performance and that of the machines they will serve.

New Departures have *all* of the important attributes that make them leaders . . . superior design, forged alloy steels for great endurance—precision dimension—and equally important: *creative engineering* to apply the bearings effectively and economically.

It pays to design New Departure Ball Bearings into your machines.

Please do not forget that, whatever your bearing problems, New Departure engineers are always ready to help you save and improve.

New Departure, Division General Motors Corporation, Bristol, Conn., or Detroit, Chicago, San Francisco, Los Angeles, Boston, Pittsburgh.



Nothing Rolls Like a Ball

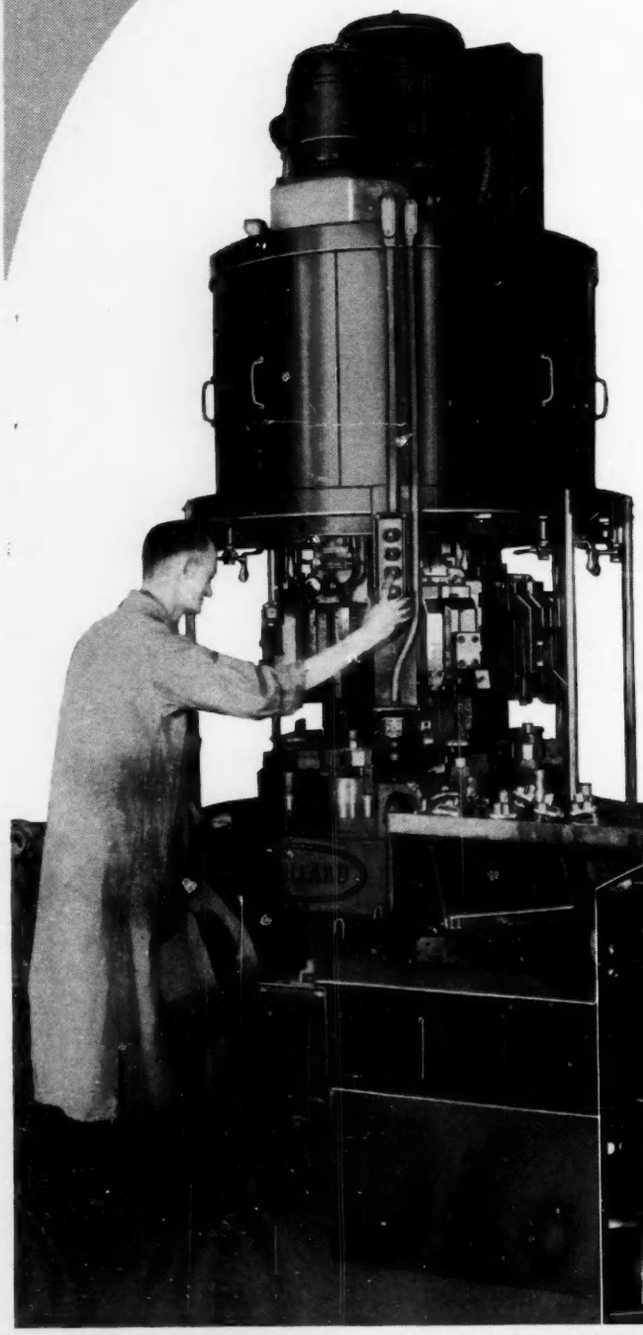
NEW DEPARTURE

WORLD'S NO.1 BALL BEARING

2716

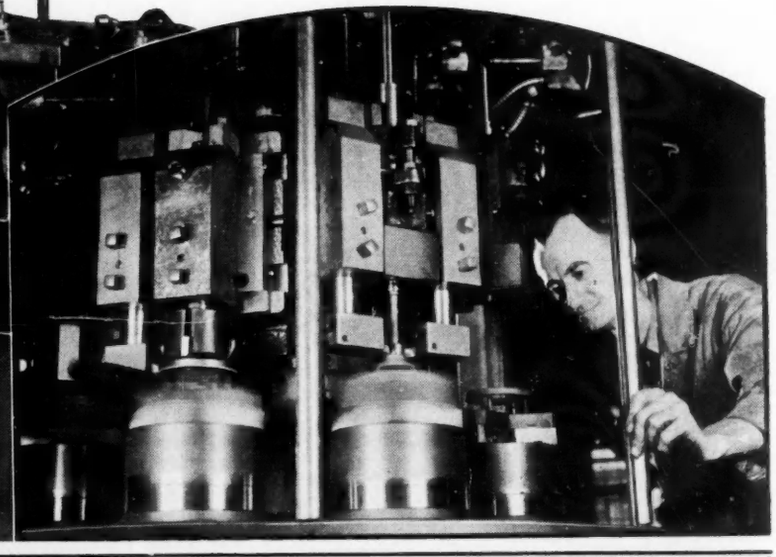
SATISFIED USERS ··

tell the story

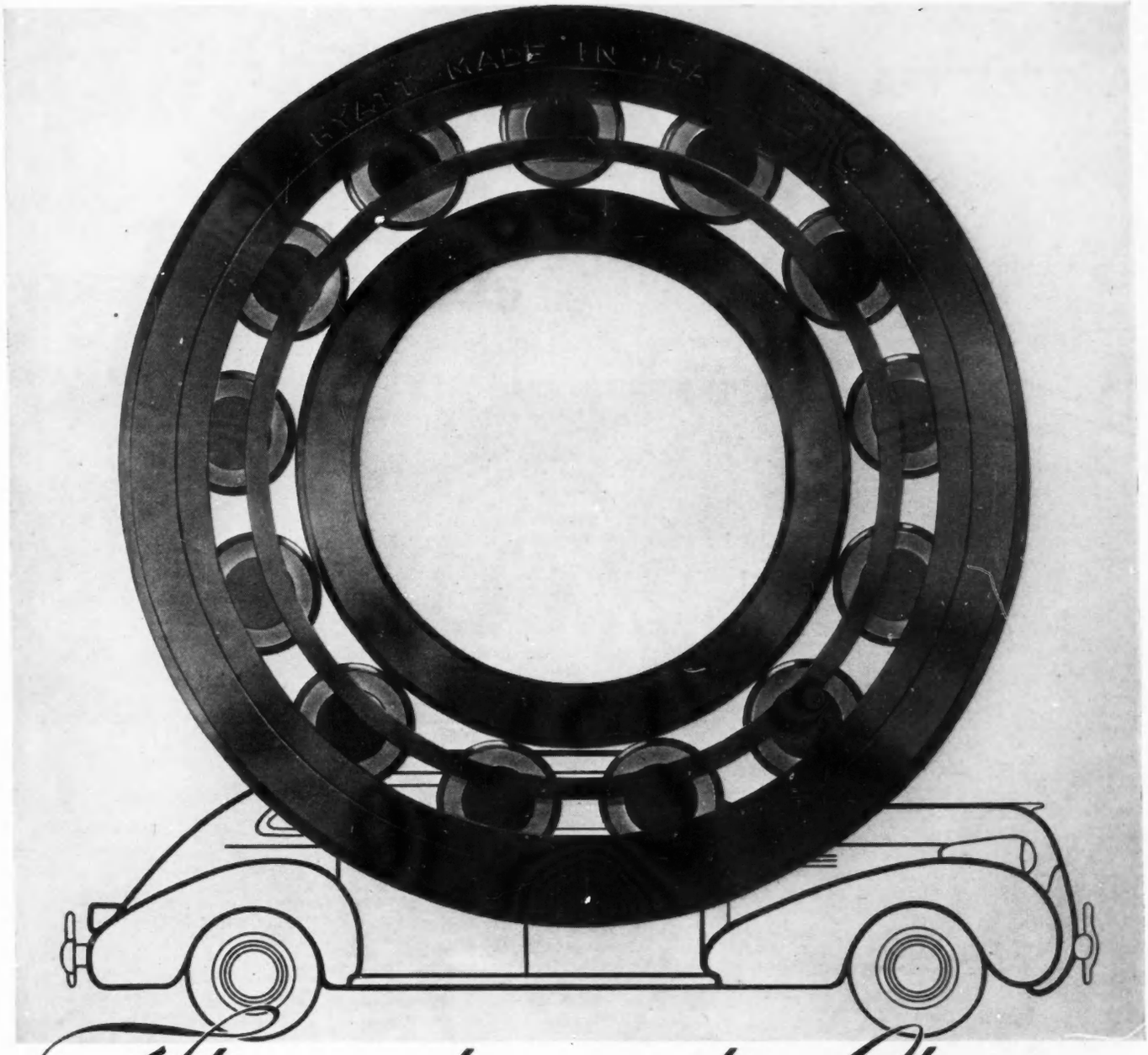


These new Type "J" Mult-Au-Matics in the field have equalled, if not bettered, their expected Performance. Some of these machines are scheduled for single jobs on long runs. Others are tooled and scheduled for short run variety work. Whatever the case may be, Bullard "J" Series 7-inch and 11-inch are producing to dimension, quality, and Profit specifications.

For proof of the statement, "Bullard 'J-7' and 'J-11' Mult-Au-Matics can Profitably improve manufacturing operations," ask Bullard Engineers to submit estimates on these smaller, faster Mult-Au-Matics as applied to your work. Let us give you facts as to what these machines are doing for others.



THE BULLARD COMPANY
BRIDGEPORT, CONNECTICUT



Throughout the Chassis

...DEPEND ON HYATTS

Hidden away in the chassis of millions of motor cars, trucks, and buses, Hyatt Quiet Roller Bearings run easily, accurately, silently... giving many extra years and miles of bearing satisfaction... with least attention. Turning wheels, shafts, and gears owe their permanent alignment to Hyatt design, precision, stamina, and performance. For these reasons car builders use Hyatts extensively, and seasoned motorists prefer them. Hyatt Bearings Division, General Motors Corporation, Harrison, N. J., and Detroit, Mich.

HYATT ROLLER BEARINGS

S. A. E. JOURNAL, December, 1938, Vol. 43, No. 6. Published monthly by the Society of Automotive Engineers, Inc. Publication office at 56th and Chestnut Streets, Philadelphia, Pa. Editorial and advertising departments at the headquarters of the Society, 29 West 39th Street, New York, N. Y. $1 per number, $10 per year; foreign $12 per year; to members 50 cents per number, $5 per year. Entered as second-class matter, Feb. 16, 1933, at the Post Office at Philadelphia, Pa., under the Act of Aug. 12, 1912. Acceptance for mailing at special rate of postage provided for in Section 1103, Act of Oct. 3, 1917, authorized on Jan. 14, 1928.

A GEAR MATERIAL THAT MAKES THE GRADE.. FOR EVERY REQUIRED PROPERTY



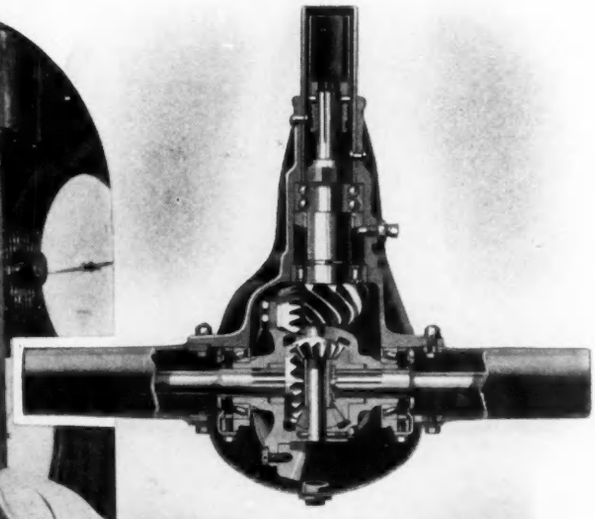


Photo courtesy of Chevrolet Motor Division
(above) *Cut-away view of typical rear axle, showing ring gear, pinion and shafts of Nickel alloy steels.*

Photo courtesy of Chevrolet Motor Car Company
(left) *Pressure test to determine the strength of a Nickel alloy steel ring gear.*

● Rear-end requirements, especially in bus and truck applications, are unusually severe. In rear axle gears there must be high resistance to wear and fatigue and minimum distortion in heat treatment.

Since the early days of the automotive industry, the Nickel alloy steels have been preferred for these highly stressed parts. In the quenched and slightly tempered condition in which they are applied in gears they have conclusively demonstrated that they possess the high strength, wear resistance, fatigue resistance and toughness necessary to insure dependability and a wide margin of safety in service.

The Nickel alloy steels also offer decided manufacturing advantages. They forge easily. They respond readily to annealing for easy machining with a high finish and they may be hardened by mild quenching in oil with a minimum of distortion.

There are many other places where the Nickel alloy steels play an indispensable role in modern motor car construction. Consultation on any of your problems involving the use of Nickel is invited.

FOR GREATER DEPENDABILITY USE
NICKEL ALLOY STEELS

THE INTERNATIONAL NICKEL COMPANY, INC., 67 WALL ST., NEW YORK, N. Y.

Announces

AND TWO NEW FORDS FOR 1939

and out. Both have new *hydraulic brakes*, built to strict Ford standards. And both . . . by reason of scientific soundproofing and new seat-construction . . . invite you to enjoy a really restful ride.

The Ford V-8 offers a choice of the 60-horsepower engine, for which owners report 22 to 27 miles a gallon — or the 85-horsepower engine, famous around the world for its fine performance. The De Luxe Ford V-8 has the 85-horsepower engine only.

THE FORD QUALITY GROUP: Beside these two new Ford cars, and the new Mercury, the Ford Motor Company also presents the 12-cylinder Lincoln-Zephyr and Lincoln — five quality cars — each representing the remarkable value and performance for which the Ford name stands.

With this broad range of cars and prices, our dealers are in a position to serve your individual needs better than ever before.



1939 FORD FEATURES

INTERIORS: spacious, pleasant, well appointed. Mohair or broadcloth upholstery.

QUIET: thorough, scientific soundproofing provides unusually quiet operation.

COMFORT: new seat construction, soft cantilever transverse springs, big double-acting hydraulic shock absorbers.

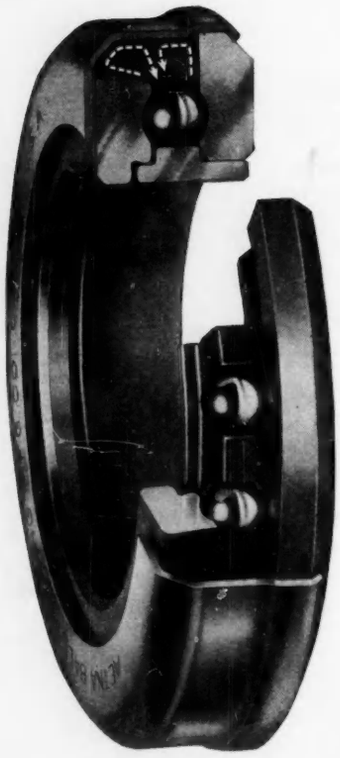
V-8 ENGINE: thrifty 60 horsepower or brilliant 85 horsepower in Ford V-8. 85 horsepower only in De Luxe Ford V-8.

HYDRAULIC BRAKES: precision-built to meet strict Ford standards of safety. Even, easy pedal action. Big drums and large lining areas for fast stops and long life.

ALL-STEEL BODIES • LOW PRICES



STANDARD ON MORE THAN 50% OF THE GREAT NEW 1939 MODELS



● **Only after exhaustive tests do automotive engineers change standard equipment. Yet, in the four years since Aetna "T" Type Clutch Release Bearings were introduced over 50% of the manufacturers using clutch release bearings have switched *TO* Aetnas, none have since switched *FROM* Aetnas.**

There's ample reason for such a marked preference. Aside from being the ultimate in metallurgical and mechanical excellence Aetna "T" Type Clutch Release Bearings offer new, revolutionary advantages and definitely end all the troubles characteristic of conventional type bearings. (see below)

If your cars and trucks are not Aetna-equipped why not investigate? Laboratory "torture" tests, time and millions of miles of usage under the severest heat, cold and load conditions have proven them worthy of your consideration. AETNA BALL BEARING MANUFACTURING COMPANY, 4608 SCHUBERT AVENUE, CHICAGO, ILLINOIS. Detroit Office: 7310 Woodward Avenue. Cleveland Office: 402 Swetland Bldg.

Cut-away view shows Aetna's famous* ONE-PIECE, *"T"-shaped, oil-cushioned bronze ball retainer that assures true alignment—abolishes the noise, wear and tear of eccentric thrust. Arrows show how beveled surfaces force centrifugally thrown lubricant back to vital working parts. Only Aetnas have these features. Only with these features can you be sure of positive, escape-proof, carefree lubrication.

Aetna "T" Type CLUTCH RELEASE BEARINGS

THE "T" TAMES TROUBLE

Pack up your Troubles in *his* Old Kit Bag



U·S·S CARILLOY ALLOY STEELS
To strengthen vital parts such as springs, bearings, gears, etc. Special analyses for every purpose.

U·S·S STAINLESS STEELS
To permanently resist corrosion. To assure lasting beauty in hub caps, window frames, radiator grilles, and interior trim. To simplify cleaning and polishing—to add a selling advantage that will help dealers sell more cars.

U·S·S CONTROLLED STEELS
Carbon steels for forging, forming, heat treating and machining in which *all* quality *factors* are predetermined.

U·S·S HIGH TENSILE STEELS
To carry high unit stresses and to reduce weight to a minimum at low cost. (U·S·S COR-TEN has resistance to atmospheric corrosion 4 to 6 times that of plain steel).

U·S·S ABRASION-RESISTING STEEL
To reduce abrasive wear wherever earth, sand, gravel, waste, etc. flow over, through or against your equipment. Costs little more than plain carbon steel.

SPECIAL PURPOSE STEELS
USS
STEELS FOR EVERY PURPOSE

WHEN we say "Bring your steel problems to us"—it is not an empty phrase. It is a definite offer of assistance.

Today with more than 10,000 different alloys available to the steel user, steel making has become immeasurably more complicated. And so has steel buying.

To get the most value from your steel dollar you must match the steel you use to the special need for it. And that's quite a job. It presupposes a knowledge of the infinite variations in physical properties and chemical compositions of these special steels—of how differently they act in fabrication and perform in service—and most important of all, how they differ in ultimate cost.

That knowledge we are prepared to offer you.

Our corps of metallurgical contact men is trained to know all there is to know about custom-made steels. When one of them steps into your plant his first duty is to find out what you want the steel to do, both in your shop and in your product—then to select the exact grade of steel that will do the best job for you at lowest cost. His job is to analyze your steel requirements and to interpret them to our steel makers in such a way that you get not only the right steel but get it exactly right again and again. Let *him* pack your troubles in his old kit bag.

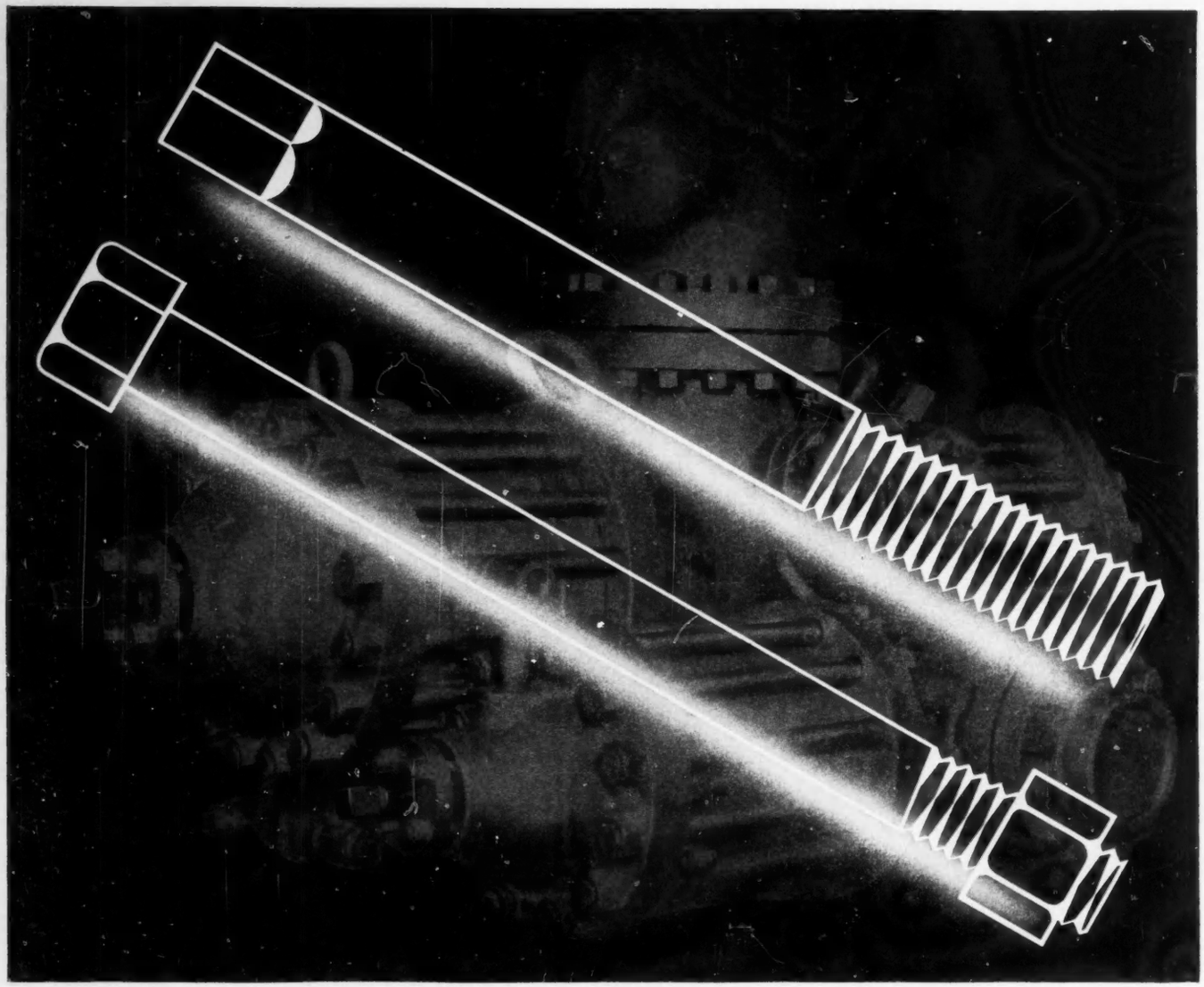
CARNEGIE-ILLINOIS STEEL CORPORATION

Pittsburgh *Chicago*

Columbia Steel Company, San Francisco, *Pacific Coast Distributors*

United States Steel Products Company, New York, *Export Distributors*

UNITED STATES STEEL



WHERE HIGH PRESSURE MUST BE CONTROLLED

When oil drilling encounters pressures of several thousand pounds per square inch, there must be no "ifs" or "buts" about the blow-out preventer's capacity to hold things in check. In one particular type of blow-out preventer (6000-pound test), thrust screw and head bolts play a vital part in preventing accidents and waste. They are made of Chrome-Moly (S.A.E. 4140) steel because:

(a) Chrome-Moly bolt stock can be heat treated in tonnage lots with perfect assurance of consistently developing the combined strength and toughness to withstand heavy loads and to prevent threads from stripping.

(b) Chrome-Moly bolts are readily machinable after heat treating.

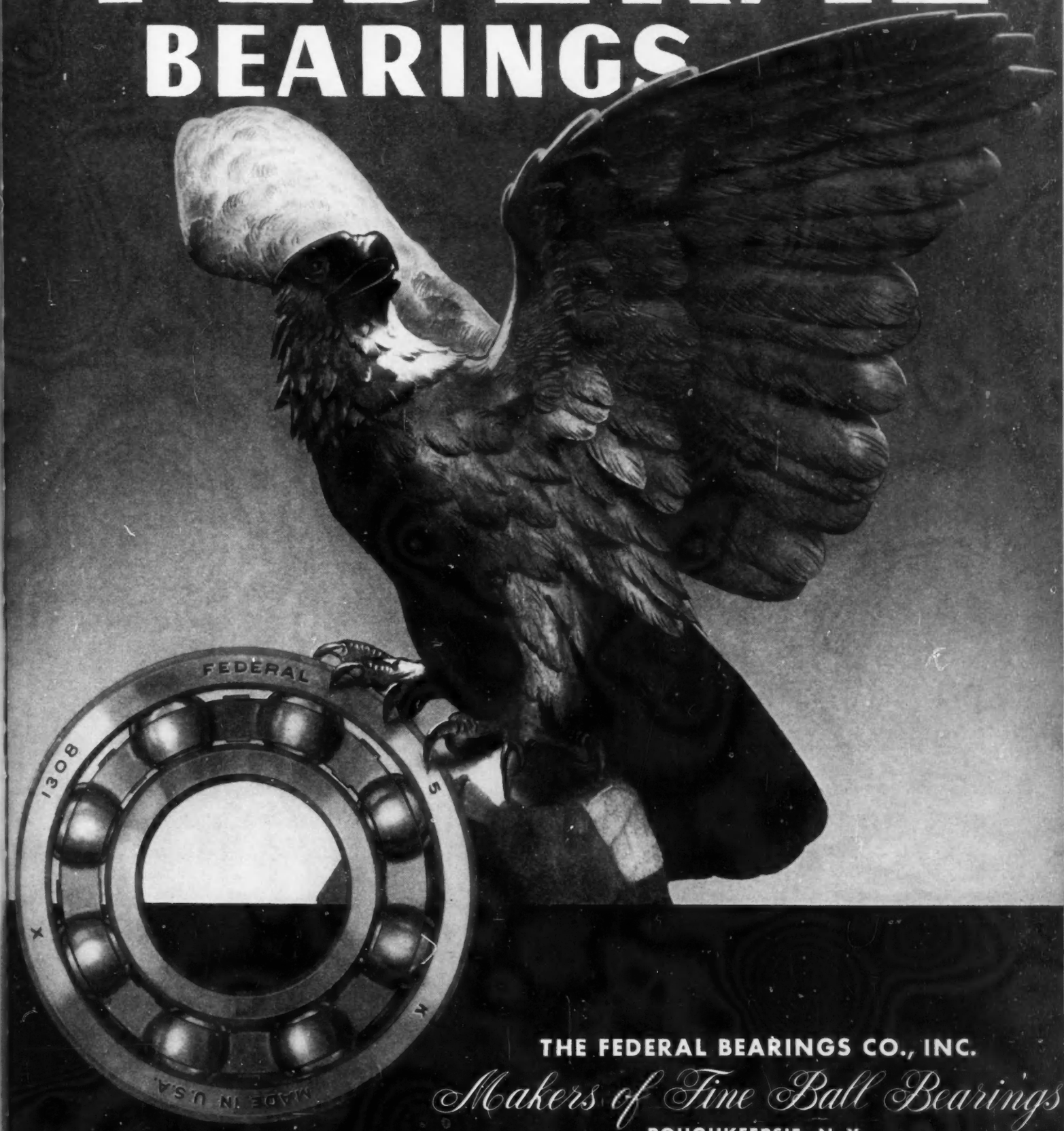
Bolt and parts manufacturers seeking to speed up and simplify production, reduce costs, and still give their customers a better product, should send for our free book, ***"Molybdenum in Steel."*** Climax Molybdenum Company, 500 Fifth Avenue, New York.

PRODUCERS OF FERRO-MOLYBDENUM, CALCIUM MOLYBDATE AND MOLYBDENUM TRIOXIDE

Climax Mo-lyb-den-um Company

MOLY

FEDERAL BEARINGS



THE FEDERAL BEARINGS CO., INC.

Makers of Fine Ball Bearings

POUGHKEEPSIE, N. Y.

Detroit Sales Office: 2608 Book Tower • Chicago Sales Office: 120 N. Peoria St.
Cleveland Sales Office: 402 Swetland Building

IS THIS CHILD

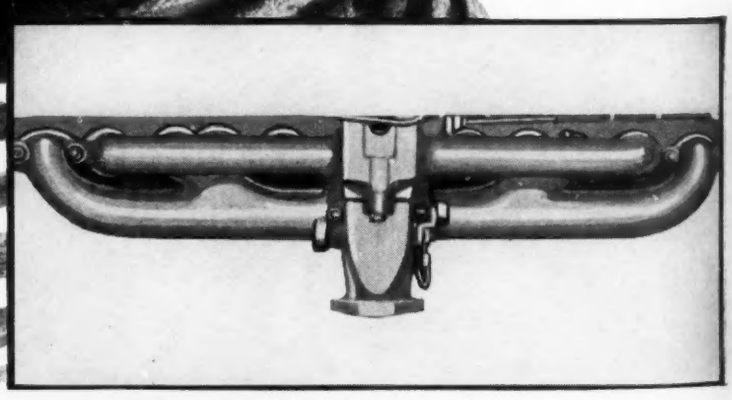


FIG. 1. **DIFFERENTIAL HEATING** and cooling along the length of the intake manifold contribute to variations in temperatures and air-fuel ratios of the mixtures entering the cylinders.

SICK OR WELL?

A doctor would take his temperature as the first step in a diagnosis. A similar procedure at several points in an automobile engine may reveal wide variations in the temperatures of individual cylinders and of the mixtures entering the cylinders. Such differences change the knocking tendencies of individual cylinders.

VARIATIONS in atmospheric temperature alone may cause a difference of several octane numbers in the anti-knock rating of a fuel in an automobile engine.

Other temperature variations may have just as marked an influence on the knocking tendency of any one cylinder. These occur in the engine itself. Since a single knocking cylinder may rate the anti-knock value of the fuel, these other temperature variations seem to deserve further study.

In the intake manifold, temperature variations are caused by the fact that the hot spot arrangement found on most automobile engines today applies more heat to the center of the manifold than to the ends. Also, the front of the manifold receives the greatest cooling effect from the fan blast and the central and rear parts less, as the air is warmed by the exhaust manifold in its journey to the rear of the engine. Such differential heating and cooling result in variations in the rate of vaporization along the length of the intake manifold, which cause differences in air-fuel ratios and temperatures of the mixtures entering the cylinders.

The cylinders themselves and the cylinder head are subject to further differences in temperature. Variations in coring in the foundry cause inequalities in cylinder wall thickness. *En bloc* design prevents each cylinder from being subjected to the same effect of the cooling liquid. Consequently, there are variations in cylinder and cylinder head temperatures which cause differences in the knocking tendency of individual cylinders and in the ratings of fuels.

The large differences in the temperatures of cylinder, cylinder head and piston head surfaces and the relative inaccessibility of these parts preclude a direct determination of the integrated temperature conditions within the cylinder. These *integrated* mixture temperatures may, however, be reproduced approximately in the cylinder of the laboratory knock testing engine by adjusting the temperature of the cylinder or mixture to give the same knock ratings on a group of widely different fuels, other conditions of operation being the same as in the cylinder of the multicylinder engine. When this condition has been obtained, the temperature of the combustion chamber may be more uniform than the multicylinder cylinder under study, but the integrated effect is the same. The same over-all temperature effect may be obtained in a single-cylinder engine with a cooling liquid temperature much different from that measured in the multicylinder engine.

The laboratories of the Ethyl Gasoline Corporation have had the problem of mixture distribution temperature and spark advance and their effect on the anti-knock value of gasolines under study for some years. These problems will be discussed further in future advertisements in this series.

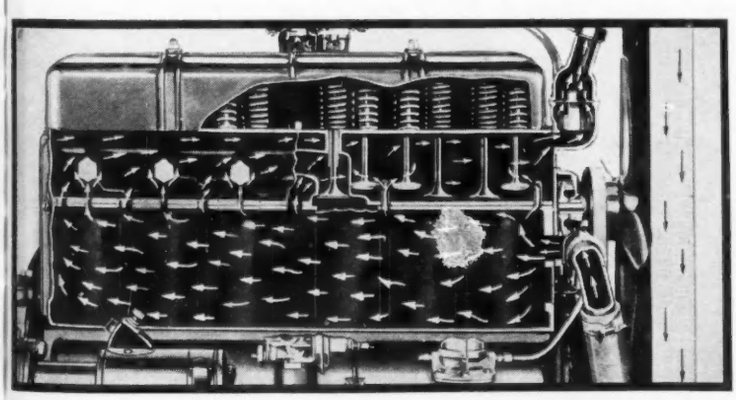


FIG. 2. **THE ARROWS** in the above diagram indicate the direction of flow of the cooling liquid in an automotive engine. All of the cylinders are not subjected to exactly the same effect of the cooling liquid even in the best of modern *en bloc* designs.

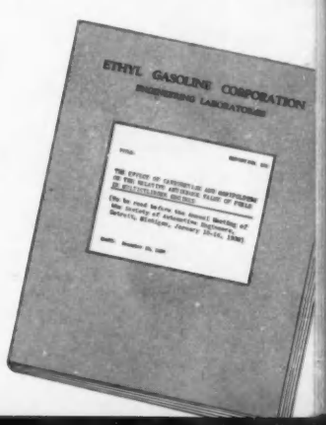
REPORT AVAILABLE—The results of this investigation to date have been published in a report entitled "The Effect of Carburetion and Manifolding on the Relative Anti-Knock Value of Fuels in Multicylinder Engines." If you will fill out this coupon or address a request to the Ethyl Gasoline Corporation, 405 Lexington Avenue, New York, New York, a copy of the report will be mailed to you.

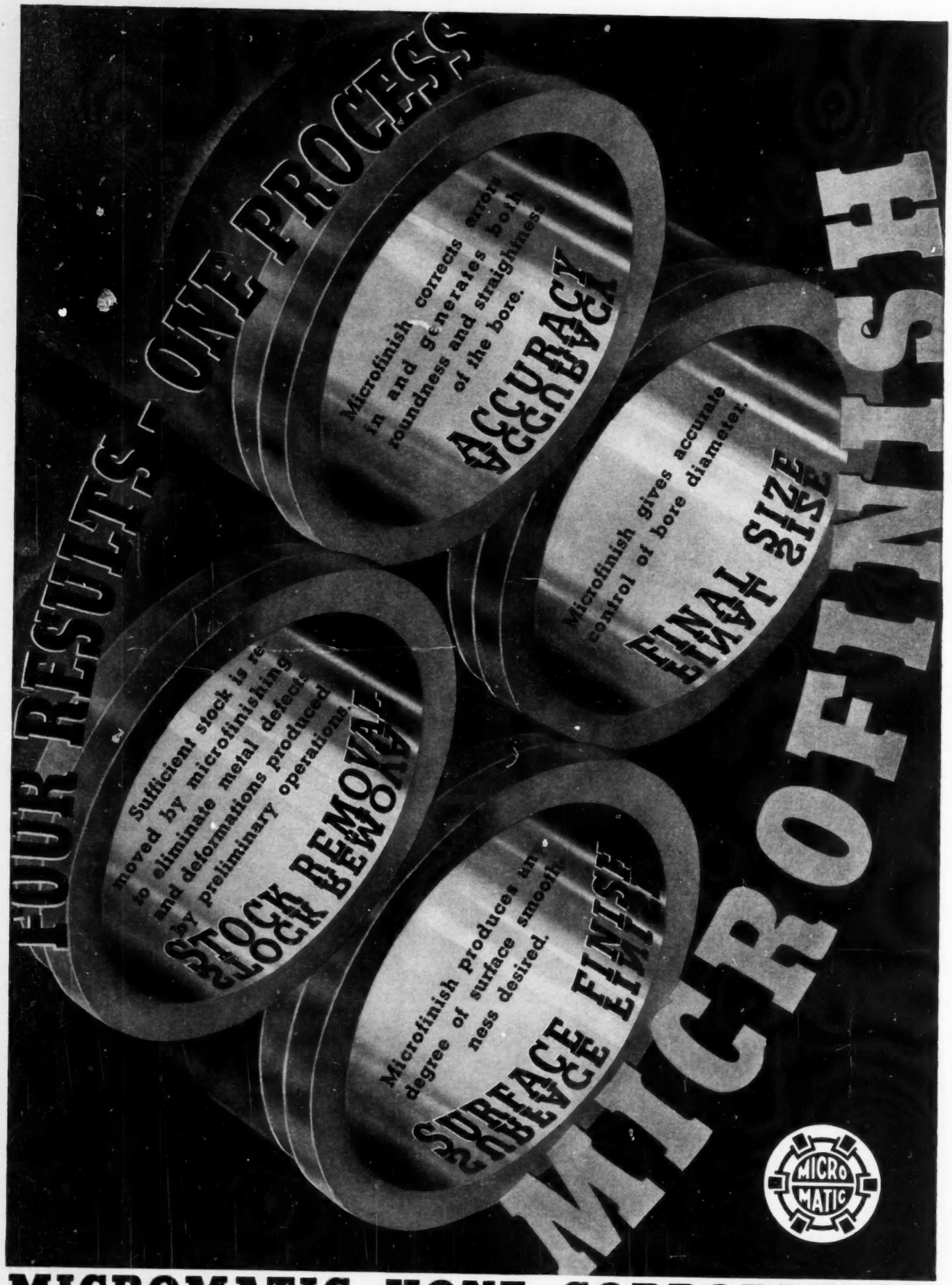
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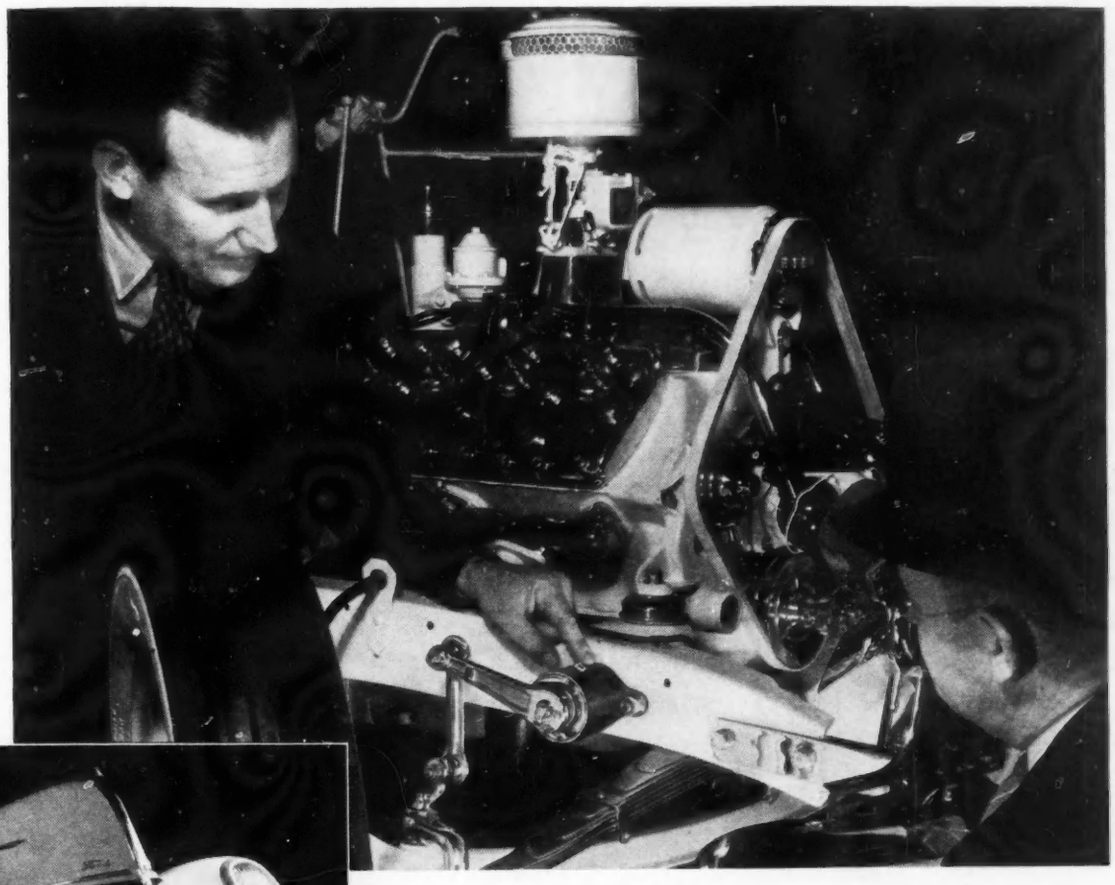




MICROMATIC HONE CORPORATION

7401 DUBOIS STREET - - - DETROIT, MICHIGAN

FOR 1939





The finest of all shock absorbers for *the finest of all Fords*

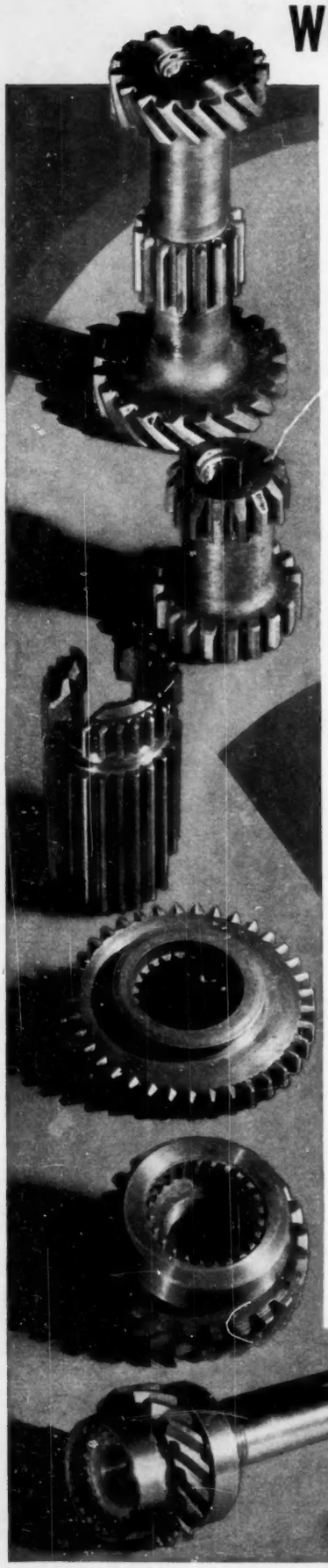
New highs in beauty, economy, comfort and quality are set unmistakably by the new Ford and Lincoln cars for 1939—the Ford V-8, the DeLuxe Ford V-8, the new Mercury 8, the new Lincoln-Zephyr and the Lincoln.

And one of the almost countless evidences of this fact is the Houdaille Hydraulic Suspension which again is an important feature of Ford and Lincoln mechanical design. Houdaille frequently proves to be as important to the salesman as to the engineer—for it helps sales just as emphatically as it helps riding performance.

HOUDAILLE HYDRAULIC SUSPENSION

HOUDE ENGINEERING CORPORATION, BUFFALO, N. Y. • A DIVISION OF HOUDAILLE-HERSHEY CORPORATION

BETTER MACHINABILITY MEANS BETTER GEARS when TIMKEN Alloy Steels are used



GOOD WILL BASED ON GOOD STEEL

Through the excellent machinability and uniform quality of TIMKEN Alloy Steels many manufacturers are producing better gears at lower finished cost. Rejects are fewer; production is increased.

TIMKEN Alloy Steels machine smoothly and accurately in any type of gear cutting machine because (1) physical and metallographic standards are consistently high and uniform; (2) grain size can be closely controlled to meet any requirement.

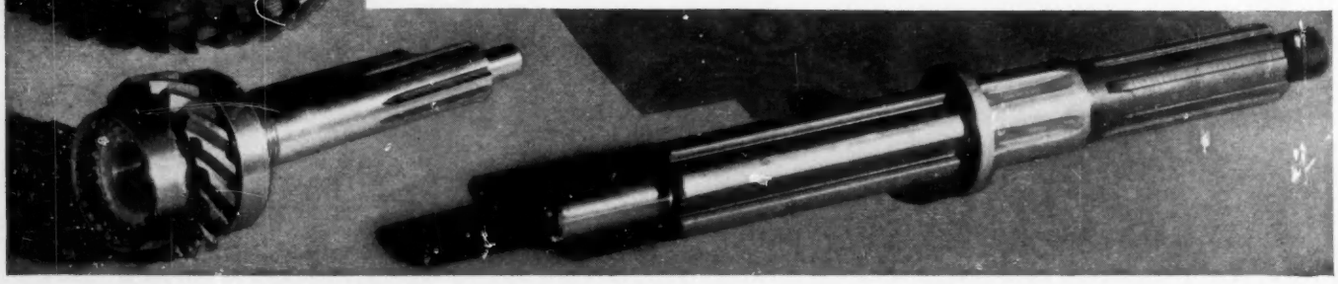
TIMKEN Alloy Steels possess every other desirable quality for gear manufacture. Response to heat treatment is consistently satisfactory. Distortion is held to a minimum and is so uniform that piloting, lapping and grinding are greatly reduced.

For transmission gears, ring gears, pinions and all other types of highly stressed gears, use TIMKEN Alloy Steels —carburizing or oil hardening—and combine quality with economy.

THE TIMKEN ROLLER BEARING COMPANY, CANTON, OHIO
STEEL AND TUBE DIVISION

TIMKEN
ALLOY STEELS

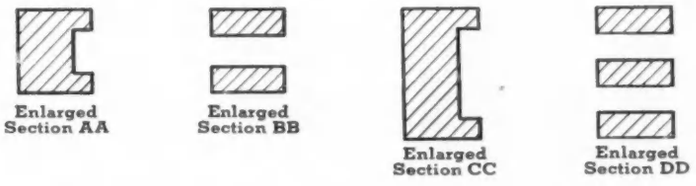
Manufacturers of TIMKEN Tapered Roller Bearings for automobiles, motor trucks, railroad cars and locomotives and all kinds of industrial machinery; TIMKEN Alloy Steels and Carbon and Alloy Seamless Tubing; TIMKEN Rock Bits; and TIMKEN Fuel Injection Equipment.



By every standard a great ring!



a rugged oil control piston ring . . . entirely new in design . . . entirely superior in performance



SECTIONS AA and CC—Bridge (1) constructed opposite gap for increased strength at point of greatest stress.

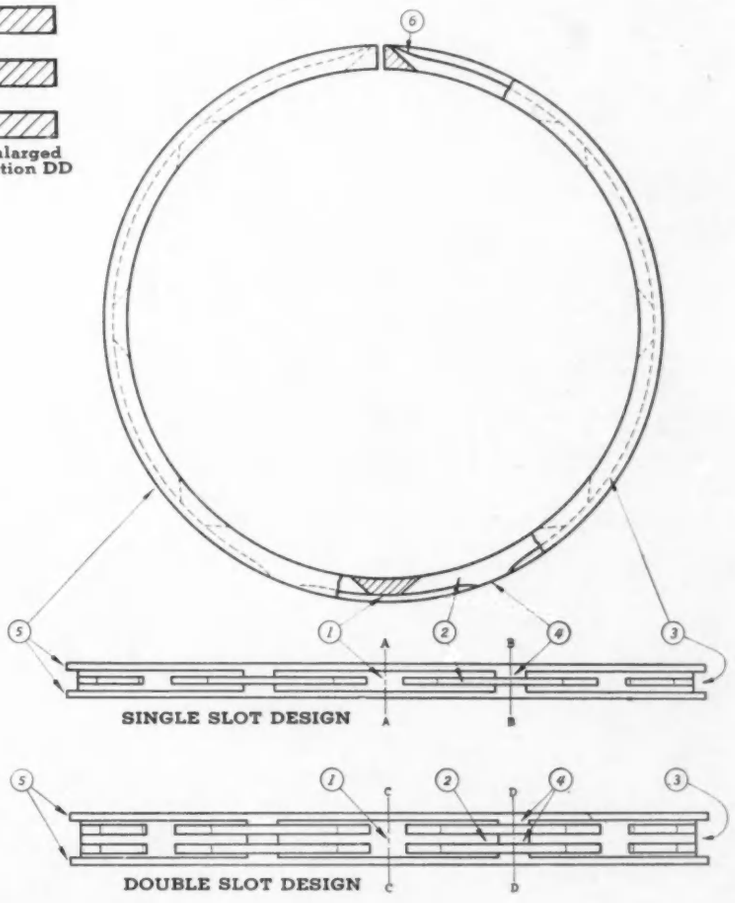
Sections BB and DD—Ring further strengthened at the back by indexing wide channel groove (3) so that it runs out forming reinforcing lands (4). Reinforcing lands (4) compensate for material removed by milling vent slots (2).

For more uniform distribution of pressures the wide channel groove (3) is machined eccentric to the peripheral surface and is shallowest at the back of the ring opposite the gap, gradually increasing in depth in the direction of the ends or gap.

Wide channel groove (3) is run out of the gap to give additional strength to contact lands (5) at points (6).

Single slot design furnished in all ring widths up to and including 3/16".

Double slot design for all rings 1/4" wide and over.



SEALED POWER PISTON RINGS

For additional facts on the Hi F Oil Ring address:

SEALED POWER CORPORATION, *Muskegon, Michigan*

Canadian Factory at Windsor, Ontario • Detroit Offices: 3-266 General Motors Building, Detroit, Michigan

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Coils
American Bosch Corp.

Condensers
American Bosch Corp.

Connections, Tire-Pump
Schrader's Son, A., Division of Scovill Mfg. Co., Inc.

Couplings, Flexible
Spicer Mfg. Corp.

Couplings, Tubing
Dole Valve Co.

Discs, Plow
Ingersoll Steel & Disc Division, Borg-Warner Corp.

Drop Forgings
Spicer Mfg. Corp.

Engines
Waukesha Motor Co.

Engines, Diesel
Waukesha Motor Co.

Engines, Industrial
Waukesha Motor Co.

Filters, Oil
AC Spark Plug Division, General Motors Corp.

Flashlights
Burgess Battery Co.

Fluxes, Soldering
American Chemical Paint Co.

Gages, Gasoline
AC Spark Plug Division, General Motors Corp.

Gages, Oil
AC Spark Plug Division, General Motors Corp.

Gages, Thermo
AC Spark Plug Division, General Motors Corp.

Gasoline, Motor Vehicle
Ethyl Gasoline Corp.

Gear, Starter
Eclipse Machine Co.

Generators (Standard Mountings)
American Bosch Corp.
Electric Auto-Lite Co.

Heaters, Hot Air
Houde Engineering Corp.

Hones, Automatic and Semi-Automatic
Micromatic Hone Corp.

Honing Machines
Micromatic Hone Corp.

Honing Tools
Micromatic Hone Corp.

Horns, Electric
American Bosch Corp.

Injection Equipment, Diesel
American Bosch Corp.

Injection Equipment, Fuel
Timken Roller Bearing Co., Fuel Injection Equipment Division

Joints, Universal
Bendix Aviation Corp
Spicer Mfg. Corp.

Lamps, Fog
American Bosch Corp.

Lathes, Turret
Bullard Co.

Machines, Automatic Chucking
Bullard Co.

Machines, Automatic Multiple Spindle
Bullard Co.

Machines, Boring (Vertical)
Bullard Co.

Machines, Boring, Turning and Facing (Vertical)
Bullard Co.

Machines, Chucking
Bullard Co.

Machines, Chucking and Turning
Bullard Co.

Machines, Honing
Micromatic Hone Corp.

Machines, Multiple Spindle
Bullard Co.

Magnetos
American Bosch Corp.

Motors, Electric
Wagner Electric Corp.

Mufflers
Burgess Battery Co.

Nickel, Cast Iron
International Nickel Co., Inc.

Nickel, Non-Ferrous Alloys
International Nickel Co., Inc.

Nickel Plating, Bright
Bright Nickel Corp.

Nickel, Steel Alloys
International Nickel Co., Inc.

Oil Rings, Piston
Sealed Power Corp.

Paints, Heat-Resisting
American Chemical Paint Co.

Panels, Instrument
AC Spark Plug Division, General Motors Corp.

Pipe, Welded Steel
Republic Steel Corp.

Piston Rings
Houde Engineering Corp.
Sealed Power Corp.

Pistons, Aluminum
Zollner Machine Works

Pistons, Heavy Duty
Zollner Machine Works

Plates, Clutch
Ingersoll Steel & Disc Division, Borg-Warner Corp.

Plates, Stainless-Clad Steel
Ingersoll Steel & Disc Division, Borg-Warner Corp.

Plating, Bright Nickel
Bright Nickel Corp.

Powerplants, Industrial
Waukesha Motor Co.

Propeller-Shafts
Spicer Mfg. Corp.

Pumps, Fuel
AC Spark Plug Division, General Motors Corp.

Pumps, Hydraulic
Houde Engineering Corp.

Pumps, Water
Houde Engineering Corp.

Radio
American Bosch Corp.

Refrigerators
Waukesha Motor Co.

Regulators, Temperature
Dole Valve Co.

Retainers, Oil
Chicago Rawhide Mfg. Co.

Rings, Piston
Sealed Power Corp.

Rust Proofing Processes
American Chemical Paint Co.

Screw-Machine Products
Spicer Mfg. Corp.

Screws, Tapping
Shakeproof Lock Washer Co.

Seals, Oil
Chicago Rawhide Mfg. Co.

Sheets, Cold Rolled Steel
Republic Steel Corp.

Sheets, Hot Rolled Steel
Republic Steel Corp.

Shock Absorbers
Houde Engineering Corp.

Silencers, Carburetor Intake
AC Spark Plug Division, General Motors Corp.

Spark Plugs
AC Spark Plug Division, General Motors Corp.
American Bosch Corp.

Speedometers
AC Spark Plug Division, General Motors Corp.

Springs, Coiled
Barnes-Gibson-Raymond, Division of Associated Spring Corp.
Cook Spring, Division of Associated Spring Corp.

Springs, Flat
Barnes-Gibson-Raymond, Division of Associated Spring Corp.
Cook Spring, Division of Associated Spring Corp.

Stampings
Cook Spring, Division of Associated Spring Corp.
Spicer Mfg. Corp.

Starter Drive
Eclipse Machine Co.

Steels, Alloy
Timken Roller Bearing Co., Steel and Tube Division

Steels, Automotive
Ingersoll Steel & Disc Division, Borg-Warner Corp.

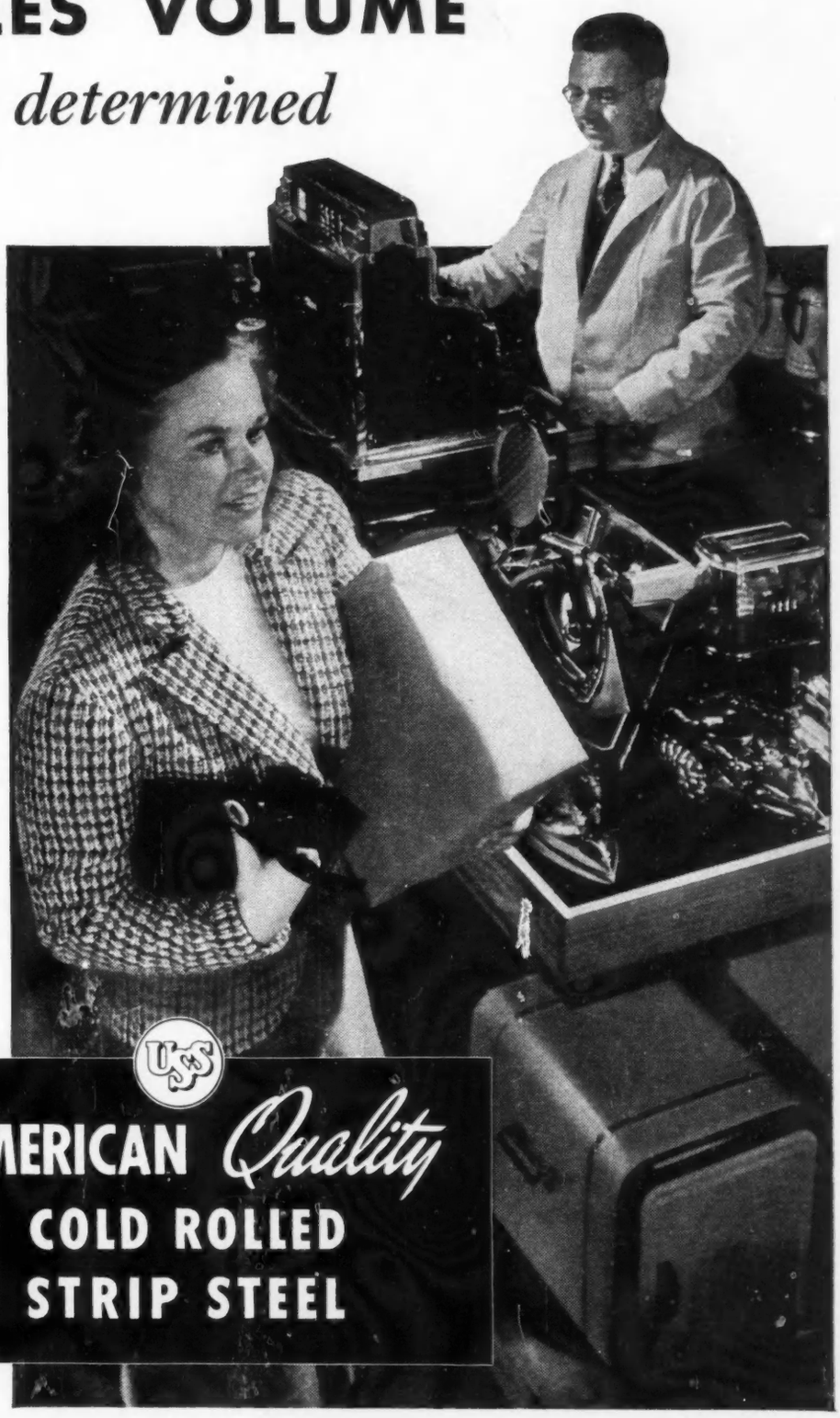
Steels, Carbon
Timken Roller Bearing Co., Steel and Tube Division

Steels, Die Rolled
Republic Steel Corp.

Steels, Electric Furnace
Timken Roller Bearing Co., Steel and Tube Division

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Here's where
YOUR SALES VOLUME
is finally determined

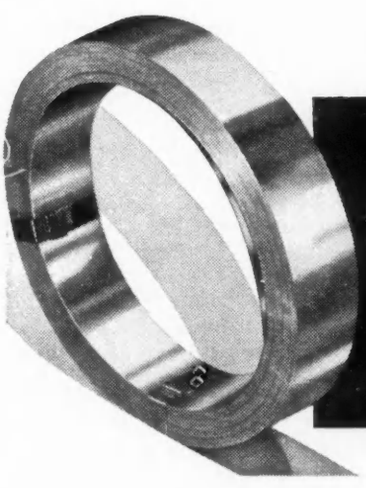


WHETHER you make complete products or parts it is the individual customer's preference that regulates your sales volume—and profits.

More and more manufacturers are turning to American Quality Cold Rolled Strip Steel because they have discovered that products made from this versatile material have the attractive, eye-catching gleam that makes sales. And because they have found, also, that the use of this material cuts costs.

American Quality Cold Rolled Strip Steel comes in a wide range of finishes, edges, tempers and widths. It is ideal for products or parts that must be light in weight, accurate in size and have a smooth finish.

We can help you. It has been our privilege to assist many customers in planning the economical application of Cold Rolled Strip Steel to their requirements. So why not put your problems up to our engineers and metallurgists?



USS

AMERICAN *Quality*
COLD ROLLED
STRIP STEEL

AMERICAN STEEL & WIRE COMPANY

Cleveland, Chicago and New York

Columbia Steel Company, San Francisco, *Pacific Coast Distributors* · United States Steel Products Company, New York, *Export Distributors*

UNITED STATES STEEL



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Timken Roller Bearing Co., Steel and Tube Division

Steels, Plow
Ingersoll Steel & Disc Division, Borg-Warner Corp.

Steels, Saw
Ingersoll Steel & Disc Division, Borg-Warner Corp.

Steels, Special Analysis
Carnegie-Illinois Steel Corp.
Columbia Steel Co.
Timken Roller Bearing Co., Steel and Tube Division
United States Steel Corp.

Steels, Stainless
Republic Steel Corp.

Steels, Tool
Timken Roller Bearing Co., Steel and Tube Division

Strainers, Gasoline
AC Spark Plug Division, General Motors Corp.

Strips, Cold Rolled Steel
Republic Steel Corp.

Strips, Hot Rolled Steel
Republic Steel Corp.

Testers, Headlamp
Thompson-Bremer & Co.

Thermostats
Dole Valve Co.

Transformers
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Tubing, Brass
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Tubing, Copper
Bundy Tubing Co.

Tubing, Steel
Bundy Tubing Co.

Tubing, Steel, Seamless
Timken Roller Bearing Co., Steel and Tube Division

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Bundy Tubing Co.

Valves, Hydraulic
Houde Engineering Corp.

Valves, Tire
Schrader's Son, A., Division of Scovill Mfg. Co., Inc.

Washers, Bronze
Bunting Brass & Bronze Co.

Washers, Lock
American Nut & Bolt Fastener Co.
Beall Tool Co.
Butcher & Hart Mfg. Co.
Eaton Mfg. Co.
National Lock Washer Co.
Philadelphia Steel & Wire Corp.
Positive Lock Washer Co.
Shakeproof Lock Washer Co.
Thompson-Bremer & Co.
Washburn Co.

Washers, Spring
American Nut & Bolt Fastener Co.
Beall Tool Co.
Butcher & Hart Mfg. Co.
Eaton Mfg. Co.
National Lock Washer Co.
Philadelphia Steel & Wire Corp.
Positive Lock Washer Co.
Shakeproof Lock Washer Co.
Washburn Co.

Zinc, Alloys
New Jersey Zinc Co.

Zinc, Sheet
New Jersey Zinc Co.

Zinc, Slab
New Jersey Zinc Co.

Zinc, Stick
New Jersey Zinc Co.

The addresses of companies listed in this index can be obtained from their current advertisements indexed on page 46

ACP

Products for the Automotive Industry

DEOXIDINE
For preparing steel and aluminum for painting. Removes rust, oil, acid runs, handmarks and other rust-accelerators, leaving an etched passive surface to which paint adheres perfectly.

RODINE
The inhibitor that revolutionized the pickling of steel, and is now used universally to prevent acid from dissolving metals.

ELECTRO-GRANODINE
The best rust-proof—paint receptive coating that can be applied directly to steel.

SPRAY-GRANODINE
A quick, effective and inexpensive process for developing, on cleaned steel, a phosphate coating that forms an excellent foundation for paint and that greatly increases the life of the applied finish.

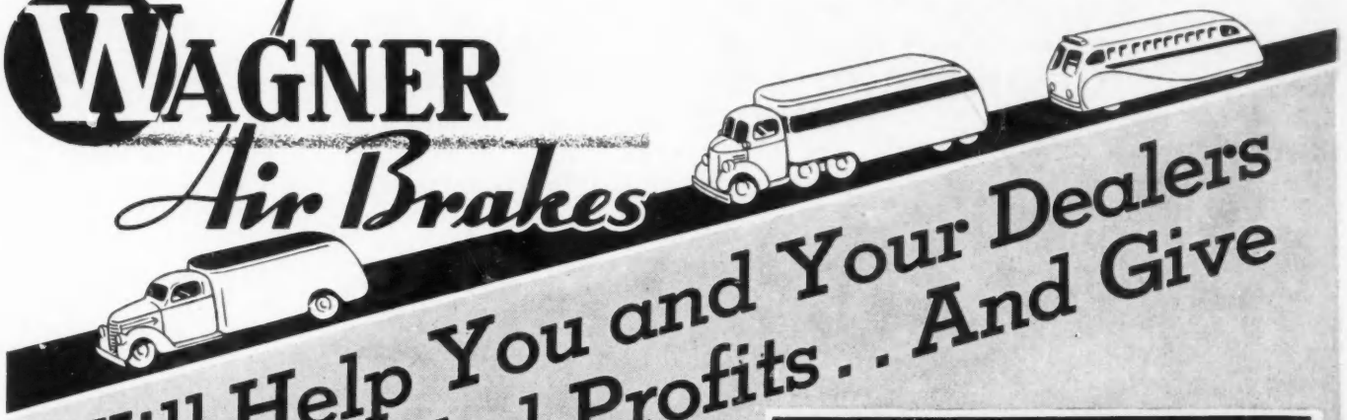
CROMODINE
The chemical used in a simple but effective process for increasing the life and adhesion of paint, lacquer or enamel finishes.

LITHOFORM
The coat-forming chemical that solves the old problem of making paint adhere to electro and hot dip galvanizing, zinc die castings and cadmium plate.

ALSO
Other famous ACP names are Flosol, Trisila, Murodine, Peroline, Ridoline, Tarnavoid, Kemick, Stripple and many more . . . each fitting into its specific link in the ACP chain of complete service.

AMERICAN CHEMICAL PAINT CO.
AMBLER PENNSYLVANIA
Detroit
6339 Palmer Ave. E.
Canadian Branch,
Walkerville, Ont.

WAGNER Air Brakes



Will Help You and Your Dealers Make Added Profits . . And Give Truck, Tractor, Bus and Trailer Users Better Service . . . .



THOUSANDS NOW BEING USED BY TRUCK OPERATORS AND FLEET OWNERS —

The Wagner Air Brake is a product of Wagner's recognized automotive engineering ability. It is the simplest and most advanced type of air brake on the market today. It has fewer parts than ordinary air brakes, making installation easier and quicker, and will give positive performance under the most severe operating conditions.

Truck operators and fleet owners, now using Wagner Air Brakes, have favorably commented on their performance, and a few excerpts quoted here are typical of thousands—

"They are a real investment for any truck operator." . . . "They have played a major part in our safety campaign." . . . "They have paid for themselves in brake lining savings alone." . . . "All our new equipment will have Wagner Air Brakes." . . . "We cannot recommend them too highly to those who want positive brake action."

Wagner Air Brakes will add to the safety and efficiency of the vehicles you are manufacturing or selling. We invite you to get complete information. Just fill in and mail the coupon at right.

Another WAGNER *contribution to driving* SAFETY and EFFICIENCY

NoRol . . . positively one of the greatest automobile safety achievements of all time. Can be quickly installed on any Hydraulic Brake equipped car to prevent back-roll while stopping in traffic or on inclines. 100% automatic. It is another Wagner item that will give you added profit.

CHECK the coupon on the right, for complete information. No obligation.

AUTOMOTIVE EQUIPMENT DIVISION

S. A. E. JL. K 38-9

Wagner Electric Corporation

6400 Plymouth Avenue, Saint Louis, U.S.A.

I am interested in receiving full information on the extra profit items checked.

- ☐ Wagner Auto-Motive-Air Power Brakes
- ☐ Wagner Automatic NoRol

NAME.....

COMPANY.....

ADDRESS.....

CITY..... STATE.....

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S.A.E.

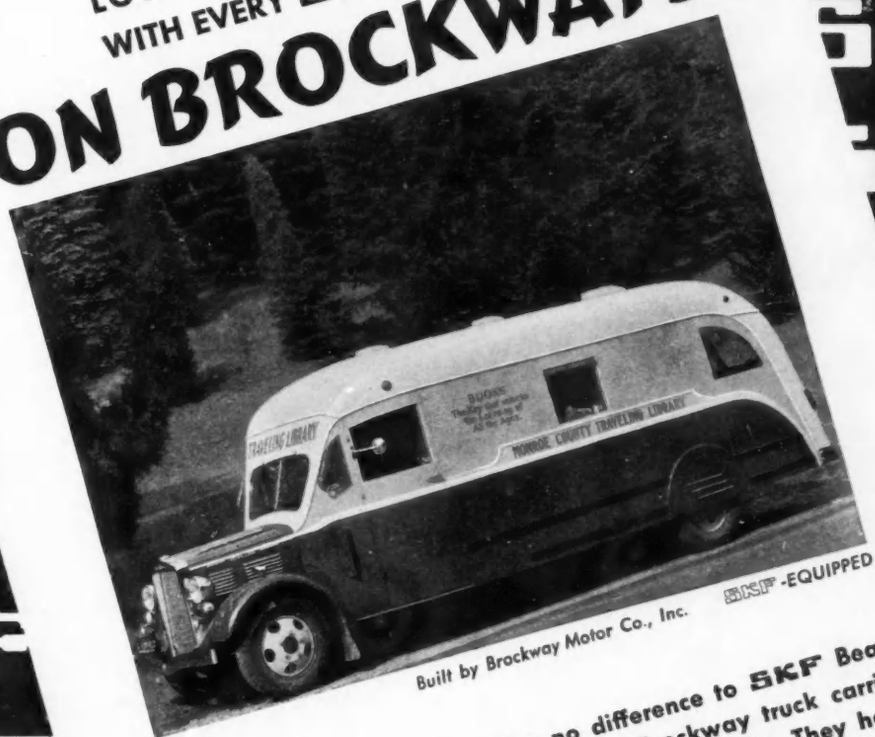
S.A.E. Annual Meeting

Book-Cadillac Hotel

Detroit + Michigan

January 9-13, 1939

LOW MILEAGE COSTS GO
WITH EVERY SKF BEARING
ON BROCKWAYS



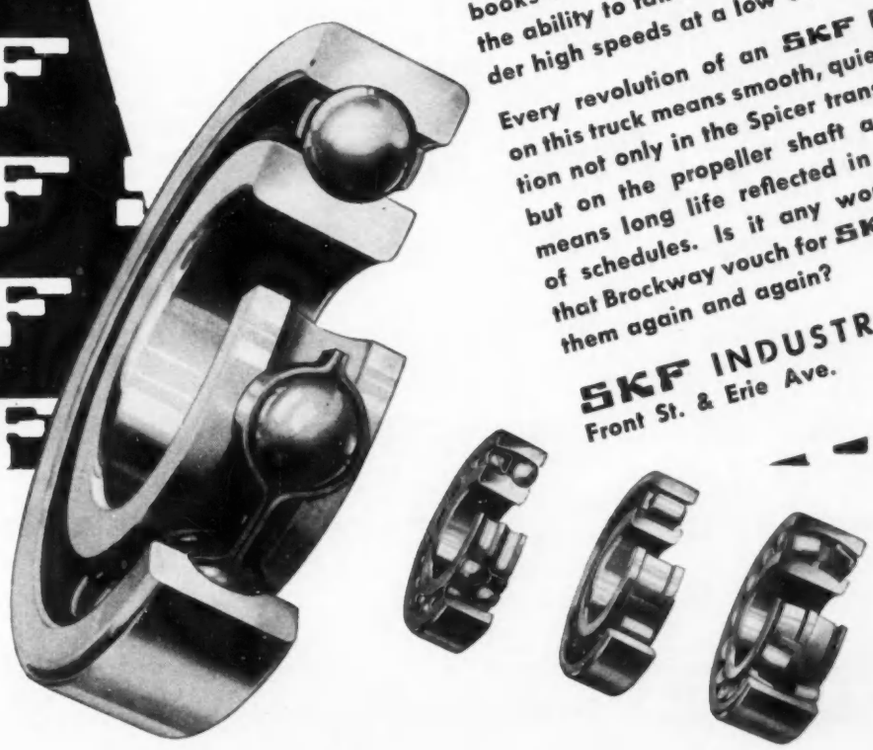
Built by Brockway Motor Co., Inc. SKF-EQUIPPED

It makes no difference to SKF Bearings whether a Brockway truck carries books or heavier materials. They have the ability to take all kinds of loads under high speeds at a low cost per mile.

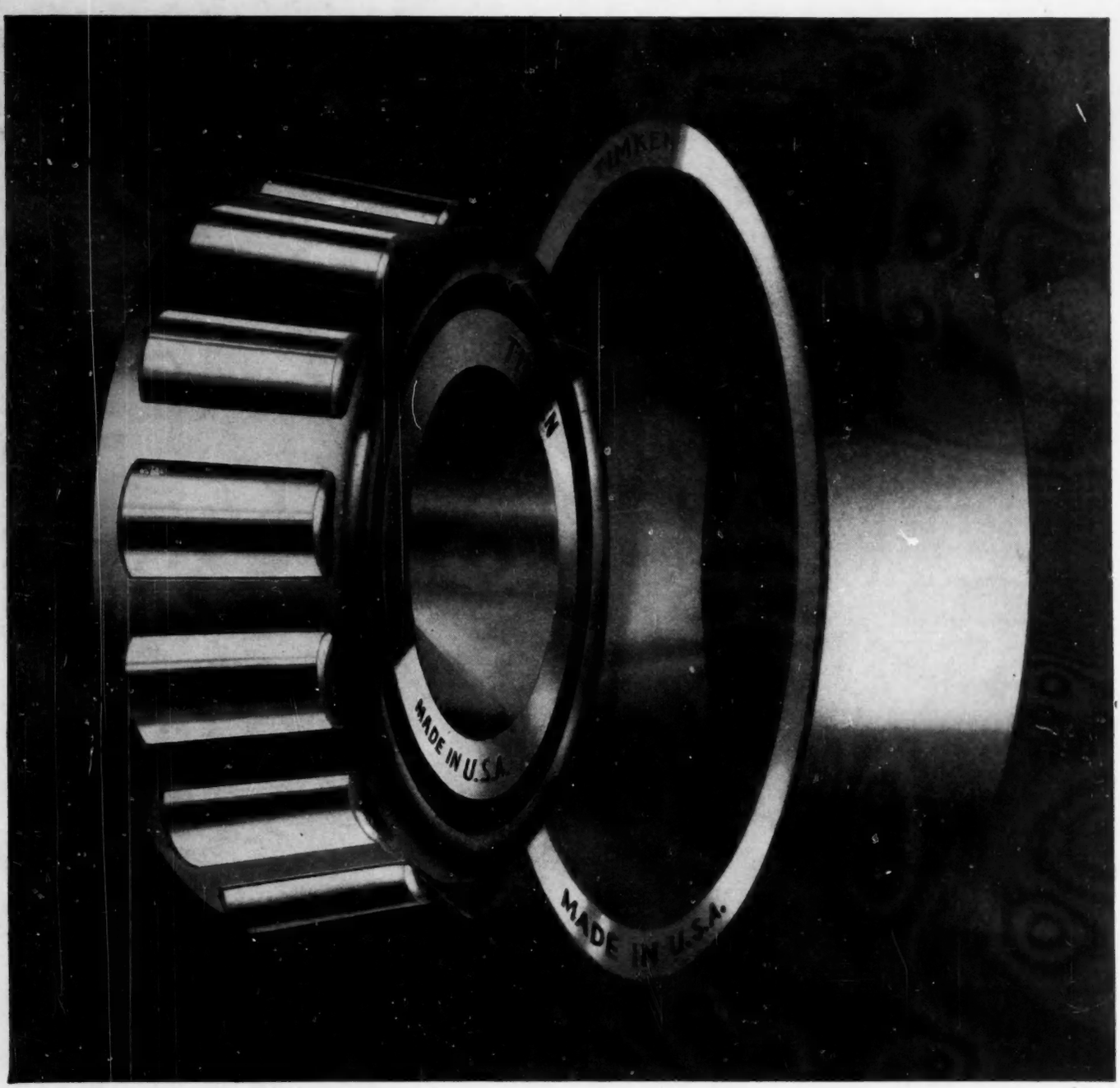
Every revolution of an SKF Bearing on this truck means smooth, quiet operation not only in the Spicer transmission, but on the propeller shaft as well. It means long life reflected in regularity of schedules. Is it any wonder, then, that Brockway vouch for SKF by using them again and again?

4220

SKF INDUSTRIES, INC.
Front St. & Erie Ave. Phila., Pa.



SKF makes more types and sizes of ball and roller bearings than any other manufacturer in the world.



THE ultimate in tapered roller bearings with 40 years of continuous development behind it, plus the strong financial support of one of the world's largest and most famous engineering-manufacturing organizations.

THE TIMKEN ROLLER BEARING COMPANY, CANTON, OHIO